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[1.] Mariusz CYGNAR, [2.] Bogusław CIEŚLIKOWSKI, [3.] Tomasz MACHOWSKI

MODELING OF THE INJECTION AND COMBUSTION PROCESSES IN THE TSI ENGINE

[1-2.] State Higher Vocational School in Nowy Sacz, Technical Institute, Zamenhofa 1A, Nowy Sacz, POLAND
[3.] Center of Practical and Continual Education in Gorlice, 11 Listopada 43, Gorlice, POLAND

Abstract: The paper presents the modeling of the injection and combustion processes done by means of KIVA 3V program in an engine with gasoline direct injection. Complicated mathematical formulas used in the calculations which describe the processes occurring inside the engine cylinder make it possible to create complicated virtual models reflecting the real conditions in a satisfactory way. The analysis was carried out for the optimum operating conditions of the engine. For a specific initial condition, the absolute velocity of charge, the course of temperature, and the content of toxic component in the exhaust gases were determined. Based on the modeling of the injection and combustion processes in the TSI engine, which is presented in the work, the basis for a broader analysis of the gas-dynamic phenomena was obtained.
Keywords: TSI engine, stratified charge, modeling of gas-dynamic

1. INTRODUCTION

For the purpose of the analysis of the gas-dynamic phenomena occurring in the combustion engine cylinder, advanced numerical methods were applied, which enabled virtual modeling of complex construction of systems of power supply. To obtain the correctness of the working parameters of a combustion engine with the simultaneous meeting of the more rigorous criteria of cost-effectiveness and greenness, it is necessary to prepare and research new constructions of power supply systems. A numerical analysis conducted by means of KIVA 3V program consists in preparation of a model of the injection and combustion process for a stratified charge. On the basis of the simulation results, thermodynamic parameters were presented on the graphs, such as: velocity of charge, temperature and toxic components. Mathematical models applied in the calculations and description of the processes occurring inside the engine cylinder enable in a satisfying manner taking into account of their real conditions. The correctness of the simulated researches results is determined on the grounds of the input data, which is obtained by means of real researches. The analysis of the process of injection and combustion of fuel-air mixture requires the application of complex calculation methods used in specialist numerical programs [1]. The combustion of stratified mixtures in gasoline engines with direct fuel injection is essential for achieving an increase in their efficiency with a simultaneous decrease in the content of toxic components in the exhaust gases, and a decrease in fuel consumption. A thorough numerical analysis allows a precise description of thermodynamic parameters for a gasoline engine burning lean air fuel mixture. [2,3]

2. GEOMETRICAL MODEL OF THE TSI ENGINE

Simulation studies were carried out for one of the cylinders of a TSI engine (Volkswagen group) working at a speed of 2000 [rpm]. The simulation of the injection and combustion in the engine for different lean air mixtures was made possible by the use of the KIVA 3V program working in 64-bit Linux, which was developed by the National Laboratories in Los Alamos [4,5]. Figure 1 shows the general view of the calculation grid of the test engine including the intake and exhaust system, for TDC (Top Death Center). The computer simulation required maintaining the real geometrical dimensions of the engine in the calculation grid, namely: the shape of the piston head and combustion chamber, and the geometry of the intake and exhaust. The engine cylinder was designed with 47 layers of the x-axis, 40 horizontal y-axis layers and 36 layers of the z-axis. The total number of cell calculations for the engine cylinder is 67 680. Taking into account the geometry of the intake ports and valves, the total number of calculation cells is 104 000. The design of the calculation grid corresponds to the actual geometry of the test engine and is designed in such a way that near the head its density is high, in order to maximize the accuracy of the calculations. Figure 2 shows a cylinder grid in cross section for 180° crank angle (CA) after TDC. The grid was laid down in the Lagrangian [6], due to the possibility of obtaining nearly uniform calculation cell volume. In the case of complex spatial geometry, the calculation cell number is smaller than in the polar coordinate system, which directly affects the shortening of the numerical analysis.

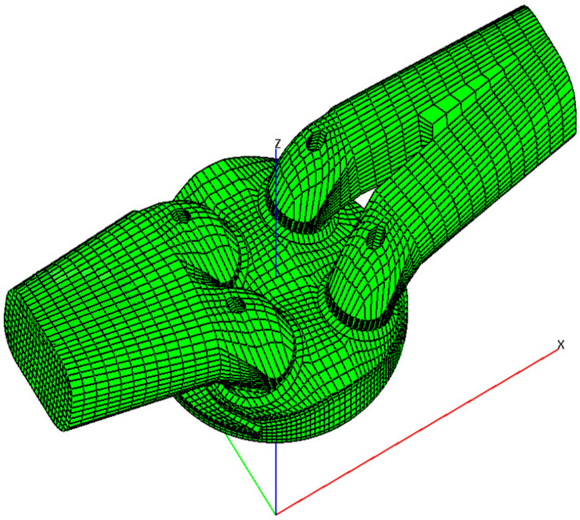

Figure 1 – Calculation grid of the TSI engine for TDC

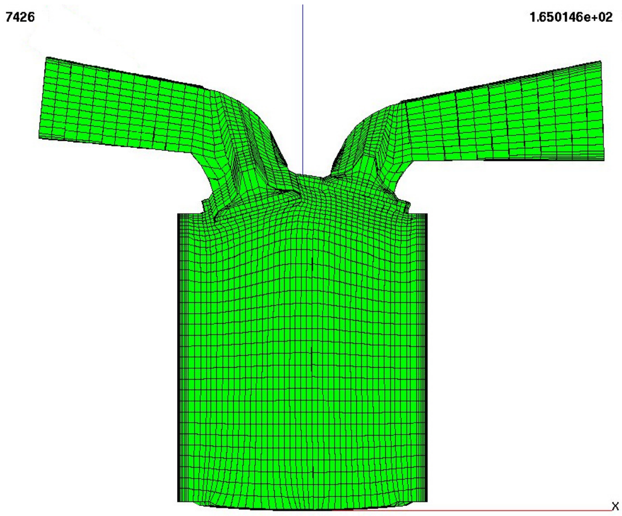

Figure 2 – Cross-section calculation grid of the cylinder engine TSI for 180° CA after TDC

3. PARAMETERS OF THE CALCULATION MODEL

During the compression stroke, the injector (located between the two intake pipes) delivers highly pressurized fuel. The fuel is injected towards the piston bowl and is directed by its wall to the spark plug. However, the injection time should be strictly defined depending on the engine speed and the ignition angle [7]. In the analysed mode, the injection angle was 70 deg BTDC and the duration of the process was 30 deg crank angle. The parameters of the fuel injection:

- rotational speed	- 2000 rpm
- direct fuel injection	- 70° before TDC
- total time of injection duration	- 30° (1.8 ms)
- mass of injected fuel per cycle	- 0.025 g
- injection path	- sinusoidal
- Sauter mean diameter SMD	- 50 μm
- injector position	- $\gamma = 65°$
- ignition moment	- 10° before TDC
- absolute pressure in the exhaust port	- 0.1 MPa
- absolute pressure in the intake port	- 0.13 MPa
- model of turbulence	- k-ε

The angle of intake valve opening at 4° CA before TDC and closing at 46° CA after BDC was adopted for calculations.

4. MODELLING OF THE INJECTION AND COMBUSTION PROCESS BY MEANS OF KIVA 3V PROGRAM

A computer simulation was carried out of the injection and combustion process for an engine working in the spark ignition mode, which was based on the PISA module (Piston Engine Simulator) dispensed in the PHOENICS program based on KIVA 3V. A simulation was carried out for the initial conditions determined in Chapter 3.

4.1. Modeling of the injection process in the engine with stratified mixture

Reitz [8] presented a fuel injection model adopted in the program KIVA for spark ignition engines with direct injection. The model takes into consideration fuel atomization, which means a close relation between drops movement and their disintegration with air movement. It is assumed that a conical fuel jet of length L and thickness h is formed at the outflow from the injector. The scheme of the atomization process of a fuel drop jet, was presented in Figure 3.

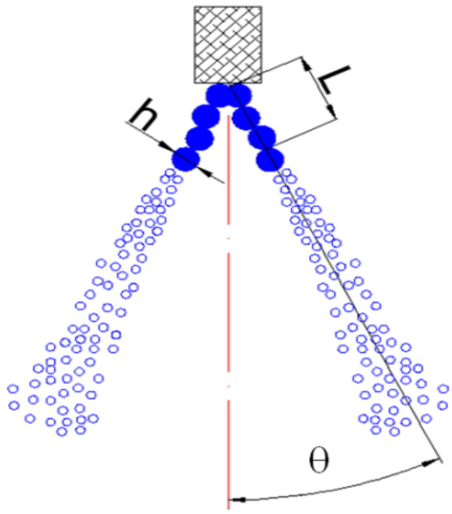

Figure 3 – Scheme of the atomization process of a fuel drop jet

The mean angle of the injection cone 2θ is determined by the geometry of the injector needle.

The dimension of the fuel drop portion corresponds with the jet thickness h. A stochastic injection model [7] is assumed. The speed of the fuel jet is determined by the difference between the injection pressure p_1 and the pressure of the surrounding p_2 and fuel density ρ_p:

$$v = K_v \left[\frac{2(p_1 - p_2)}{\rho_p} \right]^{0,5} \tag{1}$$

$$K_v = C \left(\frac{1-X}{1+X} \right)^{0,5} \frac{1}{\cos\theta} \tag{2}$$

$$X = \left(1 - \frac{2h}{d_0} \right)^2 \tag{3}$$

where: $d_0 -$ diameter of the injector nozzle outlet, $h-$ thickness of the jet

The above equations determine the initial stage of the beginning of jet breakup into smaller drops. The model of jet breakup created as a consequence of forces affecting a fuel drop moving at a speed ϑ, applies, in the KIVA 3V module, the Taylor Analogy of Breakup procedure (TAB):

$$\frac{d^2 y}{dt^2} + \frac{5\mu_p}{\rho_p r^2}\frac{dy}{dt} + \frac{8\sigma}{\rho_p r^3} y - \frac{2}{3}\frac{\rho_g \vartheta^2}{\rho_g r^2} = 0 \tag{4}$$

where: $\sigma -$ surface tension of the fuel drop, $r-$ radius of the drop before breakup, $y-$ standardized deformation of the drop (in relation to the drop radius).

The above equations were modified and applied to the program KIVA 3V.

4.2. Modeling of the combustion process in the engine with stratified mixture

In the combustion model shown in Figure 4, the flame front inside the combustion chamber consists of a thin Δx layer separating the unburned mixture from the burned mixture.

The combustion model used here is a coherent flame model, consisting of a thin flame sheet dividing the fresh gases and the burned gases. This flame propagates from the spark plug towards the piston. Its shape is modeled by turbulence [9]. The average fuel consumption rate is given by:

$$\bar{\omega}_F = \beta \cdot \tau \cdot Y \cdot \rho \cdot S \cdot \Omega \tag{5}$$

where: β - ratio between the specific heats at constant pressure and at constant volume, τ - rate of fuel in the thermal expansion function, Y - average mass fraction of fuel in the fresh charge, ρ - average local density of the fresh charge, S - laminar flame velocity, Ω - flame area volume density.

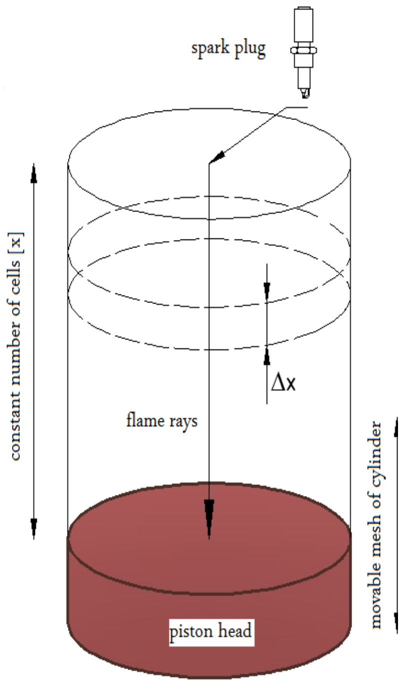

Figure 4 – Model of the mixture combustion in an engine with spark ignition

The mean motions of the swirl and the tumble are generally used in the engines to increase the turbulence levels which have a beneficial effect on the air-fuel mixture preparation. The velocity field of the one-dimensional model converges naturally to a linear solution between the zero velocity at the cylinder head and the piston velocity. However, their mean motion acts on the combustion by the turbulence. Therefore, to take into account the influence of turbulent kinetic energy generated by the swirl and tumble motion, their effect is introduced in the turbulence model by a source term obtained from the equation of the velocity:

$$\frac{d(I\omega)}{dt} = \dot{J}_i - \dot{J}_e - \tau_f \tag{6}$$

where: I - is the inertial momentum for the gas mass, ω - is the angular velocity of the fluid, $\dot{J}_i - \dot{J}_e$ - are the angular momentum flux at the intake and at the exhaust, τ_f - is the wall shear stress.

4. SIMULATED STUDIES FOR STRATIFIED MIXTURE IN TSI ENGINE

Simulation studies have been conducted for the TSI engine on the grounds of the data presented in Chapter 3 for the following parameters: absolute velocity of charge in the cylinder (Figures 5 and 6), temperature and toxic components of exhaust gases (Figures 7-10).

The movement of the air flow at the intake manifold and the cylinder is influenced by the geometry of the intake system. In this position, the inflow of the air into the cylinder is on the initial phase during the intake stroke, as presented in Figure 5. For the position 152° of the crank angle, the velocity of the air in the cylinder achieved the highest speed, which directly influences the turbulence inside the cylinder. It is important for the formation of the fuel injected directly into the engine cylinder. The distribution of the maximum speed of air is presented in Figure 6.

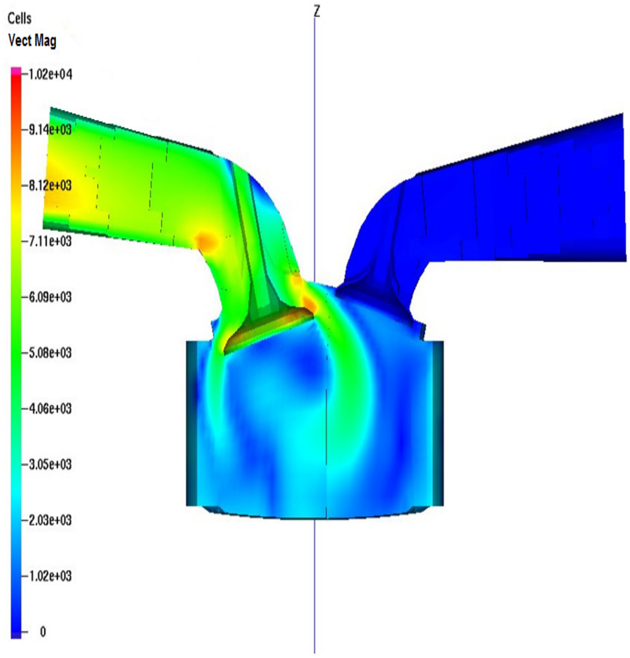

Figure 5 – Absolute velocity of charge in the cylinder for 66° CA after TDC

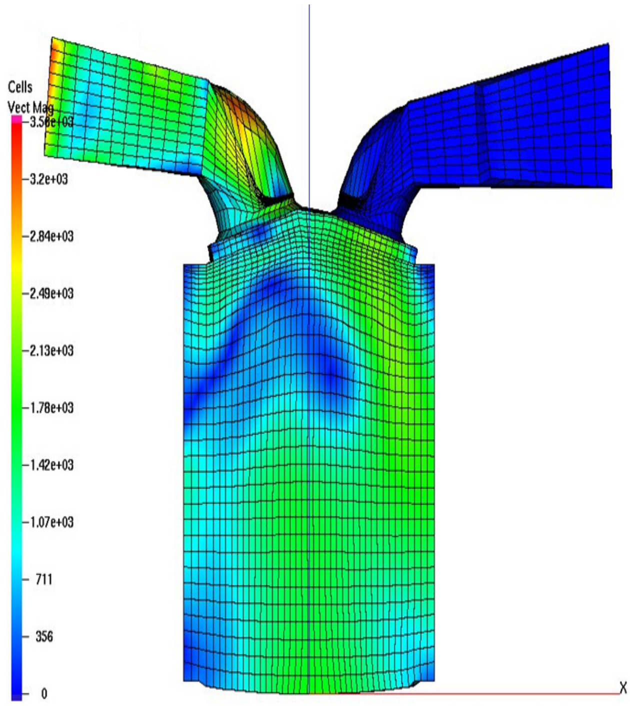

Figure 6 – Absolute velocity of charge in the cylinder for 152° CA after TDC

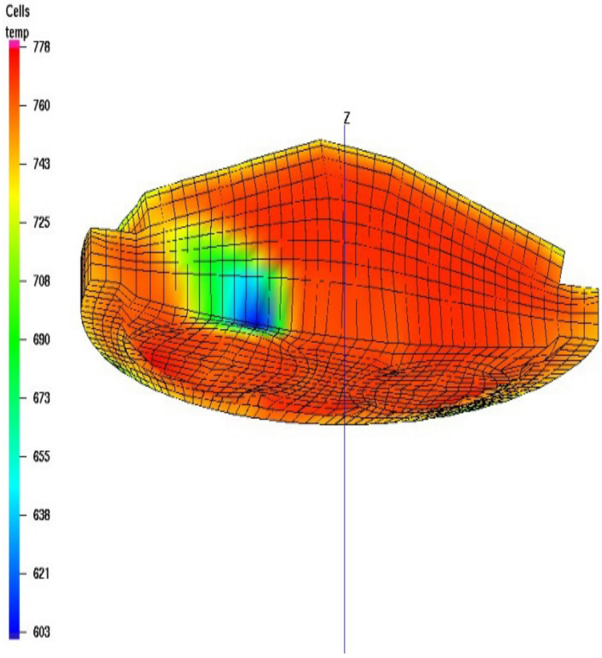

Figure 7 – Distribution of temperature in the cylinder at 20° CA before TDC

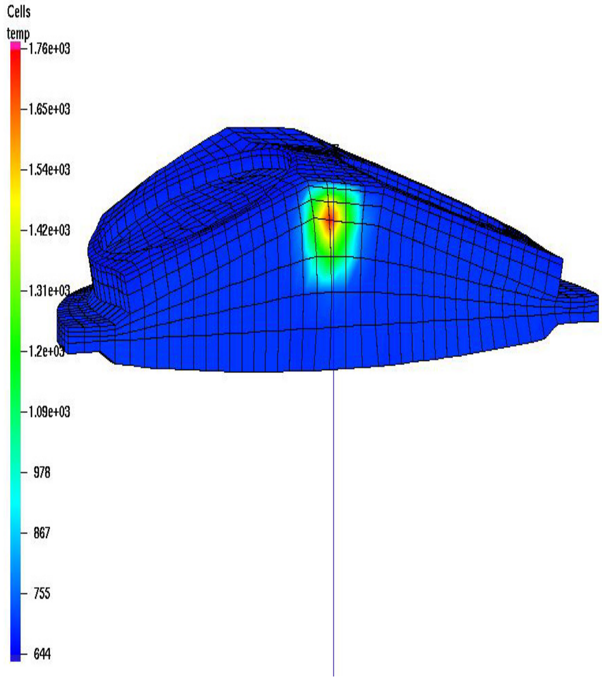

Figure 8 – Distribution of temperature in the cylinder at 8° CA before TDC

Temperature distribution in the engine cylinder is closely connected with the change of temperature of a fuel drop. Numerical analysis assumed a uniform wall temperature of the combustion chamber and a lower cylinder wall temperature. Figures 7-8 show the charge temperature in the cylinder for the selected angles of the crankshaft rotation. In Figure 7 disintegration of temperature was presented in combustion chamber at the 20° crank angle before TDC. In this section areas of different temperatures of the injected fuel are presented, and the beginning of the fuel injection took place for the position 28° of the crank angle before TDC.

Figure 8 shows the initial phase of the flame propagation for 8° CA before TDC and the ignition timing occurring at 15° CA angle before TDC. At the time of the initiation of the ignition, the temperature is approximately 1300 [K]. The initial mechanism is noted of the formation of the flame which propagates radially in all directions, and other areas are lit from the flame front.

Mass participation of the analyzed components of exhaust gases in the cylinder is calculated on the basis of the number of moles of these components.

In Figures 9-10 a simulation of the toxic components of exhaust gases was presented for 4° CA after TDC and for 8° CA after TDC, respectively for carbon dioxide and nitrogen oxide.

During the injection process, one can observe a decrease in the temperature of the charge where there is liquid fuel, which is caused by a vaporization process. The combustion process during the stratified charge mode is irregular.

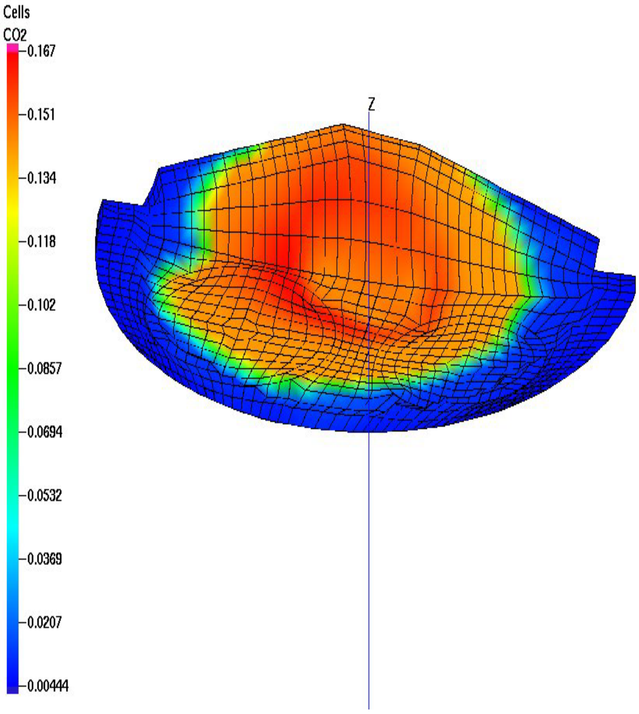

Figure 9 – Mass participation of carbon dioxide in cylinder at 4° CA after GMP

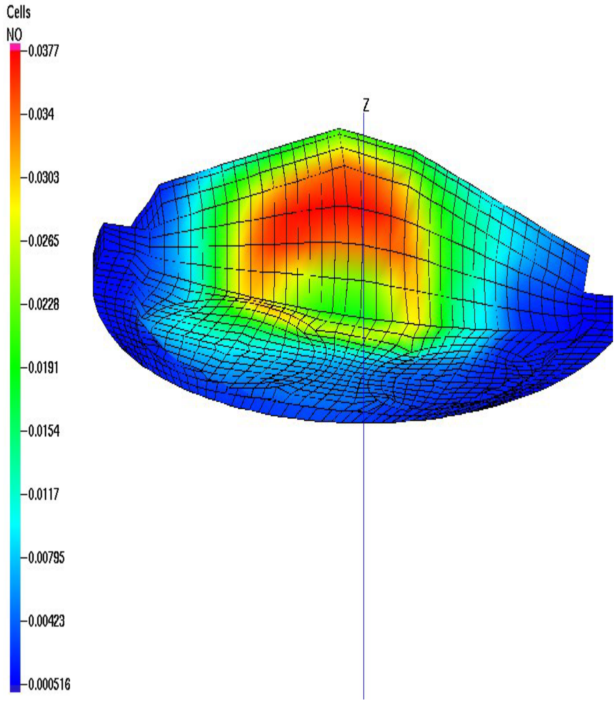

Figure 10 – Mass participation of nitrogen oxide in cylinder at 8° CA after GMP

The distribution of temperature shows the process of combustion which proceeds in a different way than in conventional engines with a homogeneous charge. However, near the spark plug, the air excess coefficient is sufficient to begin the combustion process.

5. CONCLUSIONS

The analysis of the results of the combustion process simulation showed the following results:

1. Presented numerical model applied to KIVA 3V allows to describe realistic conditions prevailing in the cylinder TSI engine.
2. Grid computing reflects the actual dimensions of the engine which is essential to obtain correct simulation results and the geometry of the power supply system allows for proper air turbulence and forming stratified mixtures on TSI engine.
3. High velocity of absolute charge allows correct formation of air - fuel mixture which directly affects the combustion process and content of toxic components..
4. The obtained results of simulation will be used for further analysis in order to carry out the optimization of injection and combustion in the engine TSI.

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