COMBUSTION SIMULATION IN A SPARK IGNITION ENGINE CYLINDER: EFFECTS OF AIR-FUEL RATIO ON THE COMBUSTION DURATION

by

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Combustion is an important subject of internal combustion engine studies. To reduce the air pollution from internal combustion engines and to increase the engine performance, it is required to increase combustion efficiency. In this study, effects of air/fuel ratio were investigated numerically. An axisymmetrical internal combustion engine was modeled in order to simulate in-cylinder engine flow and combustion. Two dimensional transient continuity, momentum, turbulence, energy, and combustion equations were solved. The k-ε turbulence model was employed. The fuel mass fraction transport equation was used for modeling of the combustion. For this purpose a computational fluid dynamics code was developed by using the finite volume method with FORTRAN programming code. The moving mesh was utilized to simulate the piston motion. The developed code simulates four strokes of engine continuously. In the case of laminar flow combustion, Arrhenius type combustion equations were employed. In the case of turbulent flow combustion, eddy break-up model was employed. Results were given for rich, stoichiometric, and lean mixtures in contour graphs. Contour graphs showed that lean mixture ($\lambda = 1.1$) has longer combustion duration.

Key words: *internal combustion engines, finite volume method, combustion modeling*

Introduction

World's energy demand is met mostly with combustion technologies. In spite of alternative technologies of energy production for power plants, there is no alternative for transportation. Researchers also work on alternative technologies, like fuel cell, for transportation sector.

Internal combustion engines are major sources of urban air pollution [1, 2]. Combustion phenomena in internal combustion engines are to be investigated in detail. The most important process on internal combustion engines is combustion process, and one of the important parts of engine modeling is combustion modeling [3]. Numerical methods are cheaper than experimental methods. Numerical methods are used to estimate the parameters, that cannot be

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measured or hard to measure, easily [4]. Internal combustion engine modeling is one of the key elements of designing new engines. Internal combustion engine concept was first designed by J. de Hautefeuille in 1676, and then developed by Ch. Huygens and M. Pupin [5]; furthermore, developed by other researchers and development of internal combustion engines still continues.

Numerous studies could be found in the literature. Researchers studied combustion using the models in a range of simple thermodynamic models to complicated chemical kinetic models. Few of these studies are given below. Ahmadi-Befrui *et al*. [6] studied two-dimensional internal combustion engine cylinder. The authors investigated in-cylinder flow and combustion; in addition, they compared some parameters of in-cylinder flow with experimental results. Bilgin [7] studied intake stroke flow of a two-dimensional internal combustion engine cylinder. Bilgin used k-ε model for turbulent flow and compared his results with other researchers' studies. Abd-Alla [8] studied spark ignition engine cycle; in fact, Abd-Alla investigated the effects of compression ratio, equivalence ratio, spark ignition timing, heat release rate, combustion duration, combustion efficiency using experimental results from other researchers, and thermodynamic expressions for ideal gas, and friction work formulas. Abu-Orf *et al.* [9] developed a new reaction model for premixed turbulent combustion in spark ignition (SI) engines. They compared their engine cylinder pressure with other researchers' experimental and numerical data; furthermore, effects of fuel type such as propane, isooctane, and methane, equivalence ratio, and engine speed on turbulent combustion in a SI engine cylinder was investigated by researchers. Kodah *et al*. [3] modeled combustion using Wiebe function. Their model was built on thermodynamic properties. They investigated the change of in-cylinder parameters with respect to in-cylinder pressure. El Tahry [10] adopted laminar flamelet model of other researchers'. He compared his computational results with other researchers' experimental results. There were disparities between results due to assumption of one-dimensional flamelet assumption.

Kong *et al.* [11] compared experimental and numerical results of HCCI engine using KIVA-3V and CHEMKIN implemented in KIVA-3V. They compared numerical results of cylinder pressure, HC and CO emissions with measured data. According to their calculations numerical model combustion results are not affected with grid resolution; *i. e*., structure of coarse mesh and fine mesh calculated combustion with same level of accuracy. Emission calculation in piston-ring crevice is required to be determined. They concluded that combustion and emission levels of HCCI are more sensitive to initial mixture temperature than wall temperatures. The computational fluid dynamics (CFD) model results are consistent with cylinder pressure, heat release rate, and HC emissions rates; however, CO emission level was under predicted.

Ogink *et al*. [12] studied gasoline homogeneous charge compression ignition engine modeling. They combined one dimensional gas dynamics program AVL-BOOST engine cycle simulation code with single-zone combustion model for further engine research. They compared their results with a single cylinder test engine. Their combined code predicts fuel consumption and indicated mean effective pressure over a range of 1-10% errors. There are many studies in the literature to solve the flow in the cylinder of an internal combustion engine like Haworth *et al*. [13] and Jasak *et al.* [14]. Akkerman *et al*. [15] studied axisymmetrical two dimensional SI engine to investigate turbulent flow produced by piston motion. Also some other studies deal with detailed combustion mechanisms like Soyhan *et al*. [16] and Kong [17].

In this study, effect of air-fuel ratio on premixed fuel air mixture combustion in an axisymmetrical SI engine cylinder was investigated numerically; moreover, effect of engine speed on combustion is studied.

Problem definition and formulation

The configuration of the cylinder with its ports is shown in fig. 1. The cylinder is axisymmetric and inlet and exhaust valves are located at the cylinder axis. In order to obtain flow field and combustion characteristics, the ensemble-averaged differential form of continuity, momentum, enthalpy, standard k-ε and chemical reaction equations are solved with appropriate boundary conditions. Turbulent combustion is simulated by eddy break-up model. All these equations were used to develop FORTRAN code to simulate the complete engine cycle. Code starts with intake stroke, then followed by compression, expansion (working), and exhaust strokes, respectively.

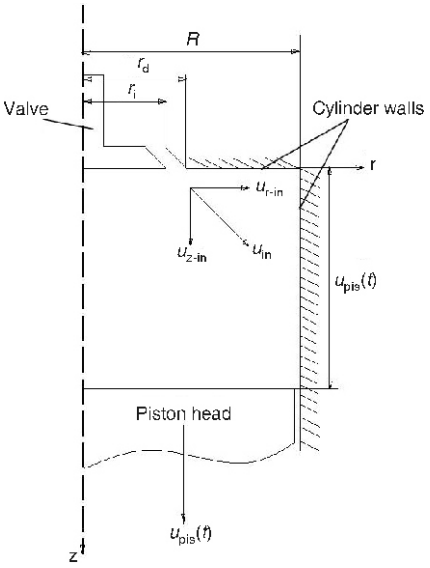

Figure 1. Schematically description of piston-cylinder assembly of the problem

Continuity equation

Continuity equation for 2-D unsteady flow in cylinderical co-ordinates are given as:

$$\frac{\partial \rho}{\partial t} \quad \frac{\partial (\rho u_z)}{\partial z} \quad \frac{1}{r}\frac{\partial (r\rho u_r)}{\partial r} \quad 0 \qquad (1)$$

Momentum equations

Momentum equations for two axes separately can be given as:

– r-axis radial momentum equation

$$\frac{\partial (\rho u_r)}{\partial t} \quad \frac{\partial (\rho u_z u_r)}{\partial z} \quad \frac{1}{r}\frac{\partial (r\rho u_r u_r)}{\partial r} \quad \frac{\partial}{\partial z}\mu_{\text{eff}}\frac{\partial u_r}{\partial z} \quad \frac{1}{r}\frac{\partial}{\partial r} r\mu_{\text{eff}}\frac{\partial u_r}{\partial r} \quad S \qquad (2)$$

$$S \quad \frac{\partial P}{\partial r} \quad \mu_{\text{eff}}\frac{u_r}{r^2} \quad S_t^r \qquad (3)$$

$$S_t^r \quad \frac{\partial}{\partial z}\mu_{\text{eff}}\frac{\partial u_z}{\partial r} \quad \frac{1}{r}\frac{\partial}{\partial r} r\mu_{\text{eff}}\frac{\partial u_r}{\partial r} \quad \mu_{\text{eff}}\frac{u_r}{r^2} \quad \frac{2}{3}\frac{\partial}{\partial r}(\mu_{\text{eff}} \quad \vec{\nabla} \quad \rho k) \qquad (4)$$

$$\vec{\nabla} \quad \frac{1}{r}\frac{\partial (r u_r)}{\partial r} \quad \frac{\partial u_z}{\partial z} \qquad (5)$$

– z-axis momentum equation

$$\frac{\partial (\rho u_z)}{\partial t} \quad \frac{\partial (\rho u_z u_z)}{\partial z} \quad \frac{1}{r}\frac{\partial (r\rho u_r u_z)}{\partial r} \quad \frac{\partial}{\partial z}\mu_{eff}\frac{\partial u_z}{\partial z} \quad \frac{1}{r}\frac{\partial}{\partial r} r\mu_{eff}\frac{\partial u_z}{\partial r} \quad S \qquad (6)$$

$$S \quad \frac{\partial P}{\partial z} \quad S_t^z \tag{7}$$

$$S_t^z \quad \frac{\partial}{\partial z} \mu_{\text{eff}} \frac{\partial u_z}{\partial z} \quad \frac{1}{r}\frac{\partial}{\partial r} r\mu_{\text{eff}} \frac{\partial u_r}{\partial z} \quad \frac{2}{3}\frac{\partial}{\partial z}(\mu_{\text{eff}} \quad \vec{\nabla} \quad \rho k) \tag{8}$$

$$\mu_{\text{eff}} \quad \mu_t \quad \mu \tag{9}$$

$$\mu_t \quad C_\mu \rho \frac{k^2}{\varepsilon} \tag{10}$$

Turbulence equations

For turbulence modeling, k-ε two equation turbulence model was chosen:

– transport equation of turbulent kinetic energy, k:

$$\frac{\partial(\rho k)}{\partial t} \quad \frac{\partial(\rho u_z k)}{\partial z} \quad \frac{1}{r}\frac{\partial(r\rho u_r k)}{\partial r} \quad \frac{\partial}{\partial z}\frac{\mu_{\text{eff}}}{\sigma_k}\frac{\partial k}{\partial z} \quad \frac{1}{r}\frac{\partial}{\partial r} r\frac{\mu_{\text{eff}}}{\sigma_k}\frac{\partial k}{\partial r} \quad S_k \tag{11}$$

$$S_k \quad G \quad \frac{2}{3}(\mu_{\text{eff}} \quad \vec{\nabla} \quad \rho k) \quad \vec{\nabla} \quad \rho\varepsilon \tag{12}$$

– transport equation of turbulence kinetic energy dissipation rate, ε:

$$\frac{\partial(\rho\varepsilon)}{\partial t} \quad \frac{\partial(\rho u_z \varepsilon)}{\partial z} \quad \frac{1}{r}\frac{\partial(r\rho u_r \varepsilon)}{\partial r} \quad \frac{\partial}{\partial z}\frac{\mu_{\text{eff}}}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial z} \quad \frac{1}{r}\frac{\partial}{\partial r} r\frac{\mu_{\text{eff}}}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial r} \quad S_\varepsilon \tag{13}$$

$$S_\varepsilon \quad \frac{\varepsilon}{k}(C_1 G \quad C_2 \rho\varepsilon) \tag{14}$$

$$G \quad \mu_t \quad \frac{\partial u_z}{\partial r} \quad \frac{\partial u_r}{\partial z}^2 \quad 2\mu_t \quad \frac{\partial u_z}{\partial z}^2 \quad \frac{\partial u_r}{\partial z}^2 \quad \frac{u_r}{r}^2 \tag{15}$$

The constants are: $C_1 = 1.44$, $C_2 = 1.92$, $C_m = 0.09$, $\sigma_k = 1.00$, and $\sigma_\varepsilon = 1.30$.

Wall functions applied to near wall flow are given in detail in Versteeg *et al*. [18].

Energy equation

Total enthalpy is used for calculating the energy equation. Energy equation is given as:

$$\frac{\partial(\rho h)}{\partial t} \quad \frac{\partial(\rho u_z h)}{\partial z} \quad \frac{1}{r}\frac{\partial(r\rho u_r h)}{\partial r} \quad \frac{\partial}{\partial z}\frac{\mu_t}{\sigma_{n,t}}\frac{\partial h}{\partial z} \quad \frac{1}{r}\frac{\partial}{\partial r} r\frac{\mu_t}{\sigma_{n,t}}\frac{\partial h}{\partial r} \quad \frac{\partial P}{\partial t} \tag{16}$$

– total enthalpy

$$h \quad C_p T \quad \frac{1}{2}(u_r^2 \quad u_z^2) \quad k \quad \Sigma m_j h_j \tag{17}$$

$$\sum_{\text{all } j} m_j \quad 1 \tag{18}$$

Chemical reaction equation

Chemical equation is given for hydrocarbon fuels:

$$\mathrm{C_nH_m} \quad n \quad \frac{m}{4} \quad \mathrm{O_2} \quad 3.76 \quad n \quad \frac{m}{4} \quad N_2 \quad n\mathrm{CO_2} \quad \frac{m}{2}\mathrm{H_2O} \quad 3.76 \quad n \quad \frac{m}{4} \quad \mathrm{N_2} \tag{19}$$

Chemical reaction transport equation is used as:

$$\frac{\partial(\rho m_\mathrm{f})}{\partial t} \quad \frac{\partial(\rho u_z m_\mathrm{f})}{\partial z} \quad \frac{1}{r}\frac{\partial(r\rho u_\mathrm{r} m_\mathrm{f})}{\partial r} \quad \frac{\partial}{\partial z} \quad \frac{\mu_\mathrm{t}}{\sigma_{\mathrm{f,t}}}\frac{\partial m_\mathrm{f}}{\partial z} \quad \frac{1}{r}\frac{\partial}{\partial r} \quad r\frac{\mu_\mathrm{t}}{\sigma_{\mathrm{f,t}}}\frac{\partial m_\mathrm{f}}{\partial r} \quad R_\mathrm{f} \tag{20}$$

During the solution of chemical reaction equation the source term R_f is depend on flow conditions. For laminar flow combustion, the reaction rate is kinetically controlled and Arrhenius type source term is used:

$$R_\mathrm{f} \quad \rho^\mathrm{a} m_\mathrm{f}^\mathrm{b} m_\mathrm{ox}^\mathrm{c} \exp \quad \frac{E}{\mathrm{R}T} \tag{21}$$

where $E/\mathrm{R} = 1.84\cdot10^4$ K.

In this study the model constants are taken as:

$$\mathrm{a} = 1.59252 + 1.54482\exp(-0.0055N)$$
$$\mathrm{b} = 1.0$$
$$\mathrm{c} = 1.6$$

For turbulent flow combustion, eddy break-up model is used. The reaction rate of fuel is taken as the smallest value of turbulent dissipation rates of fuel, oxygen, and products:

$$R_\mathrm{fu} \quad C_\mathrm{R}\rho m_\mathrm{f}\frac{\varepsilon}{k} \tag{22}$$

$$R_\mathrm{ox} \quad C_\mathrm{R}\rho\frac{m_\mathrm{ox}}{s}\frac{\varepsilon}{k} \tag{23}$$

$$R_\mathrm{pr} \quad C_\mathrm{R}\rho\frac{m_\mathrm{pr}}{s \quad 1}\frac{\varepsilon}{k} \tag{24}$$

Equation of state

Equation of state is used as supplementary equation:

$$P = \rho\mathrm{R}T \tag{25}$$

Boundary and initial conditions

Boundary and initial conditions that used in order to solve the equations are given below. Boundary conditions during intake stroke are divided into parts. At the cylinder head has two parts; first one is the valve gap for inlet process, second one is the walls of cylinder head. Fuel mass fraction in the fuel/air mixture change according to the mixture is rich, stoichiometric,

and lean mixture. On the piston head, which limits combustion chamber, velocity of the working fluid is equal to velocity of the piston. On the walls of the combustion chamber wall boundary conditions are used, due to geometry on the symmery axis, symmetric boundary conditions are used.

At the cylinder head ($z = 0$) (during the intake stroke):

$$u_z \quad u_{z\ \mathrm{in}} \qquad u_r \quad u_{r\ \mathrm{in}}$$

$$T \quad T_{\mathrm{in}} \quad 300\,\mathrm{K} \qquad k_{\mathrm{in}} \quad 0.000 1 u_{z_{\mathrm{in}}}^2$$

$$\varepsilon_{\mathrm{in}} \quad 0.1643 \ \frac{k_{\mathrm{in}}^{1.5}}{0.035B} \qquad \text{at the valve inlet}$$

$$m_f$$

$m_f = 0.0608$ for $\lambda = 0.9$ rich mixture

$m_f = 0.055$ for $\lambda = 1.0$ stoichiometric mixture

$m_f = 0.0503$ for $\lambda = 1.1$ lean mixture

$$u_z \quad 0, \quad u_r \quad 0, \quad k \quad 0, \quad \varepsilon \quad 0$$

$$T \quad 450\,\mathrm{K}, \quad \frac{\partial m_f}{\partial z} \quad 0 \qquad \text{on the walls of the cylinder}$$

On the piston face ($z = z_{\mathrm{piston}}$)

$$u_z = U_{\mathrm{pis}}(t), \quad u_r = 0, \quad k = 0, \quad \varepsilon = 0, \quad T = T_{\mathrm{pis}} = 450\ \mathrm{K}, \quad \frac{\partial m_f}{\partial z} \quad 0$$

On the cylinder surface ($r = R$)

$$u_z = 0, \quad u_r = 0, \quad k = 0\,, \ \varepsilon = 0, \ T = 450\ \mathrm{K}, \quad \frac{\partial m_f}{\partial r} \quad 0$$

At the symmetry axis ($r = 0$)

$$\frac{\partial u_z}{\partial r} \quad 0, \quad u_r \quad 0$$

$$\frac{\partial k}{\partial r} \quad 0, \quad \frac{\partial \varepsilon}{\partial r} \quad 0, \quad \frac{\partial T}{\partial r} \quad 0, \quad \frac{\partial m_f}{\partial r} \quad 0$$

Initial conditions ($t = 0$)

$$u_z = 0, \quad u_r = 0, \quad k = 0, \quad \varepsilon = 0, \quad T = T_{\mathrm{initial}} = 300\ \mathrm{K}$$

$$m_f = 0.0$$

Reciprocating motion of piston is modeled by using moving mesh technique. All equations were converted according to conversion equation of moving mesh [7, 19]:

$$\xi \quad \frac{z}{z_{\mathrm{pis}}(t)}$$

Numerical solution procedure

The governing equations subject to relevant boundary conditions were solved numerically using finite-volume method. The upwind technique was employed to discretize the con-

vective terms. All discretization steps were given in [20]. A CFD code has been developed by using the SIMPLE algorithm [21] in FORTRAN programming language. In order to obtain a solution independent of the grid distribution, grid sensitivity tests were performed by tracing the cylinder pressure against crank angle (CA) as it was done by Watkins *et al*. [22]. It is found that the solution becomes almost independent with 50 uniform grids in ξ-direction and 30 uniform grids in the r-direction. All momentum, energy and chemical reaction equations are coupled and solved together. The developed CFD code simulates four strokes of engine cycle including flow motion and combustion.

Results and discussion

In this study combustion in an idealized homogeneous charge SI engine is analyzed numerically. It is assumed that spark plug and inlet/exhaust valves are located at the centerline of cylinder. Computations are performed for three different air fuel ratios, for compression ratio of 10:1 and inlet valve angle of 60°. Engine speed is set to $N = 24$ rpm and $N = 2400$ rpm, which corresponds to laminar and turbulent flow conditions, respectively. Cylinder bore $D = 0.1$ m, stroke $L = 0.09$ m and diameter of inlet/exhaust valve $d = 0.05$ m are chosen. Methane is used as a fuel. In case of laminar combustion, Arrhenius type combustion equations were employed. In case of turbulent combustion, eddy break-up model was used.

Ignition timing is an important parameter for SI engine combustion. Ignition timing advance (ITA), is calculated by the following equation which is suggested by the authors:

$$ITA = 6.83635 + 12.282[1 - \exp(-0.00087\ N)]$$

Combustion is started assuming the 70 percent of fuel consumed at the ignition time in the region around the spark plug. This region corresponds to the computation cells of (1 to 3 1 to 10) (r ξ). The flame develops from the ignition points in a nearly hemispherical region and it reaches to the piston surface. Then the flame front propagates in the radial direction.

In order to investigate the effect of air/fuel ratio, excess air coefficient λ was chosen $\lambda = 0.9, 1.0, 1.1$, which represents the mixtures of rich, stoichiometric and lean, respectively. Air fuel mixture enters the cylinder with given values as boundary condition, then concentra-

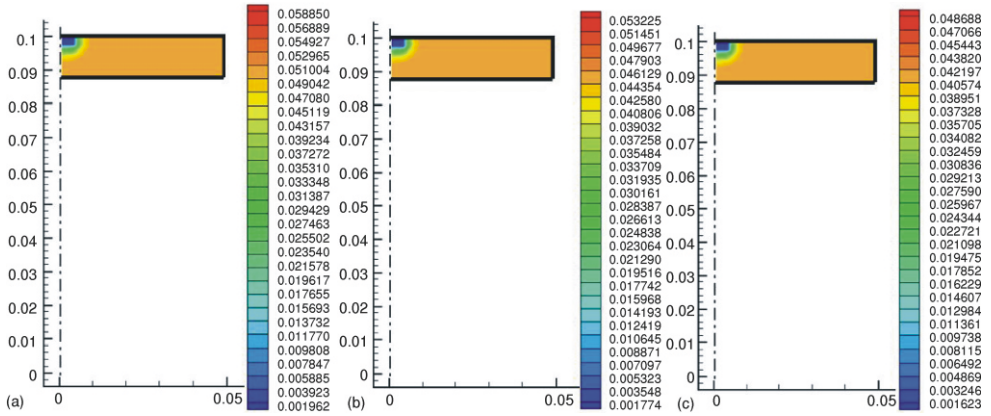

Figure 2. Mass fraction of fuel at 343 °CA. Effect of excess air coefficient λ: (a) $\lambda = 0.9$, (b) $\lambda = 1.0$, (c) $\lambda = 1.1$; (compression ratio $r = 10:1$; engine speed $N = 2400$ rpm (color image see on our web site)

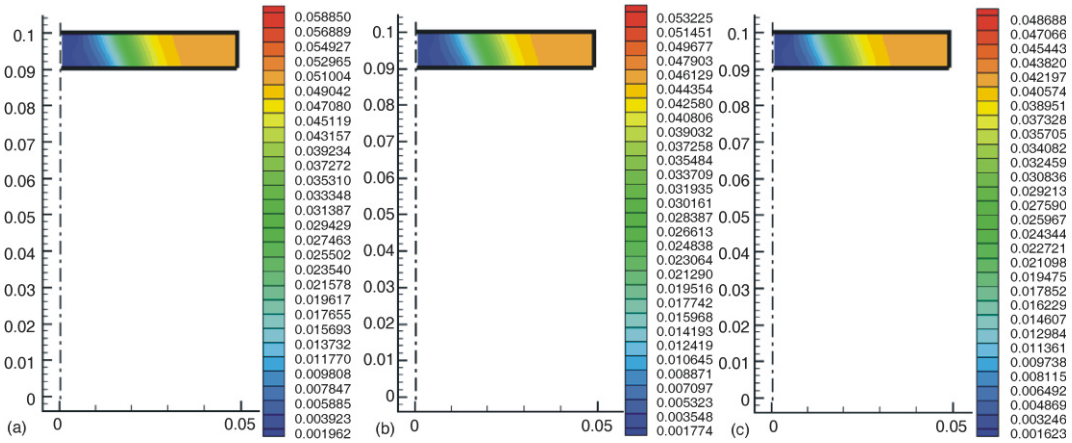

Figure 3. Mass fraction of fuel at 360 °CA. Effect of excess air coefficient λ: (a) λ = 0.9, (b) λ = 1.0, (c) λ = 1.1; (compression ratio r = 10:1; engine speed N = 2400 rpm (color image see on our web site)

tion of fuel changes in the cylinder. For the turbulent combustion modeling, engine speed was chosen 2400 rpm (figs. 2 and 3). During the intake stroke, contours of mass fraction of fuel look similar but have different values due to mixture composition. After piston reaches BDC (bottom dead center), valve opening closes and the air-fuel mixture compressed during the compression stroke and homogeneous mixture fraction obtained (fig. 2). After the ignition, the flame develops from the ignition points in a nearly hemispherical region and it reaches to the piston surface. Then the flame front propagates in the radial direction. For the rich and stoichiometric mixtures combustion duration is the same because the maximum amount of fuel that can be burned is the stoichiometric ratio of that fuel. Excess fuel does not involve in the combustion. For lean mixture, flame velocity is lower than stoichiometric and rich mixture, combustion duration is longer (figs. 3 and 4). The same trend was observed in the recent

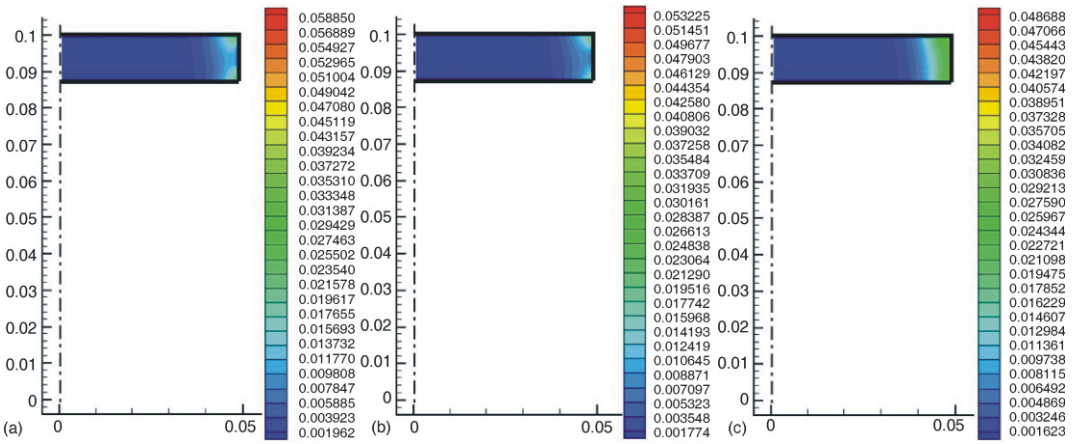

Figure 4. Mass fraction of fuel at 379 °CA. Effect of excess air coefficient λ: (a) λ = 0.9, (b) λ = 1.0, (c) λ = 1.1; (compression ratio r = 10:1; engine speed N = 2400 rpm (color image see on our web site)

study of Perini *et al*. [23] about combustion process in SI engines fueled with hydrogen, methane, and hydrogen-methane blends.

In fig. 5, mass fraction of fuel contours is given for the SI timing. Combustion duration in laminar combustion is shorter in terms of the CA, because engine speed is slower. When λ increases combustion duration also increases in laminar combustion. The longest duration is obtained for $\lambda = 1.1$ (figs. 6 and 7). Experimental investigation on flame stability of hydrogenated mixture was done by Halter *et al*. [24], it is observed that for laminar combustion, burned gas Markstein length, that is related to flame velocity, increases with decreasing air/fuel ratio. This observation is in agreement with observation of the numerical results of this study.

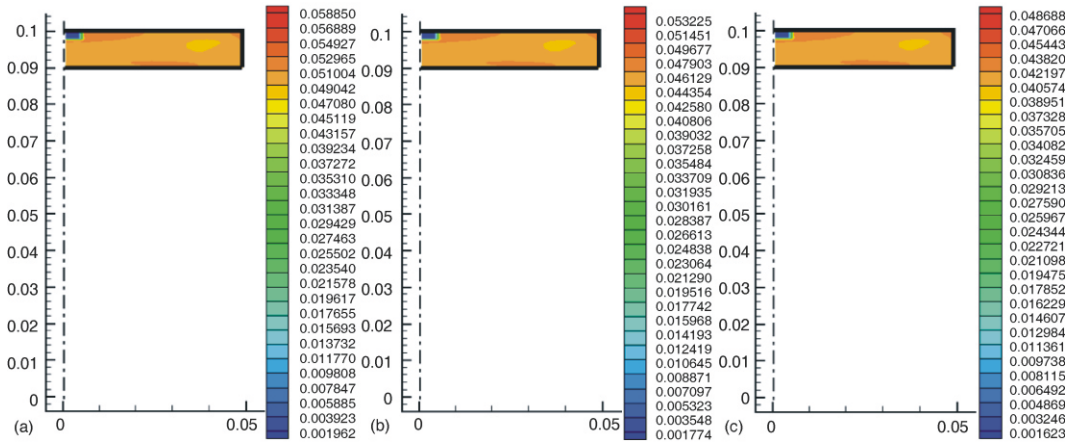

Figure 5. Mass fraction of fuel at 354.5 °CA. Effect of excess air coefficient λ: (a) $\lambda = 0.9$, (b) $\lambda = 1.0$, (c) $\lambda = 1.1$; (compression ratio $r = 10:1$; engine speed $N = 24$ rpm (color image see on our web site)

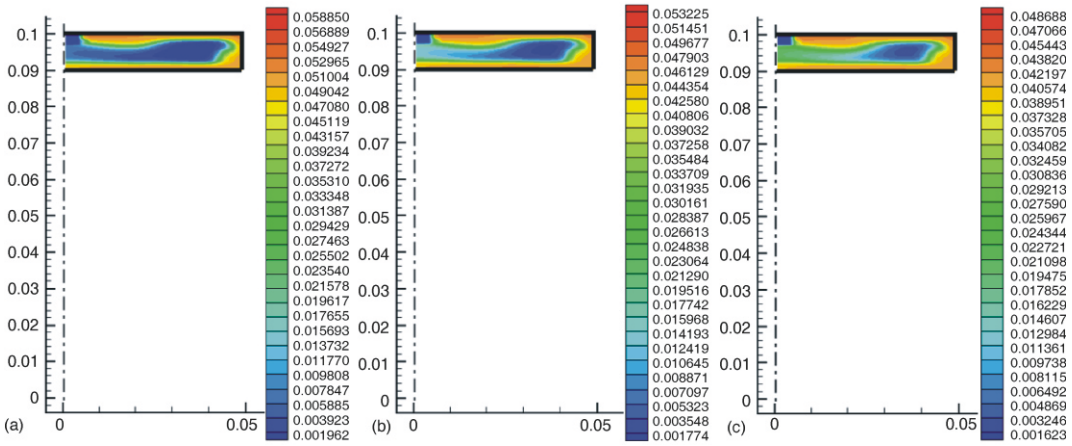

Figure 6. Mass fraction of fuel at 355.5 °CA. Effect of excess air coefficient λ: (a) $\lambda = 0.9$, (b) $\lambda = 1.0$, (c) $\lambda = 1.1$; (compression ratio $r = 10:1$; engine speed $N = 24$ rpm (color image see on our web site)

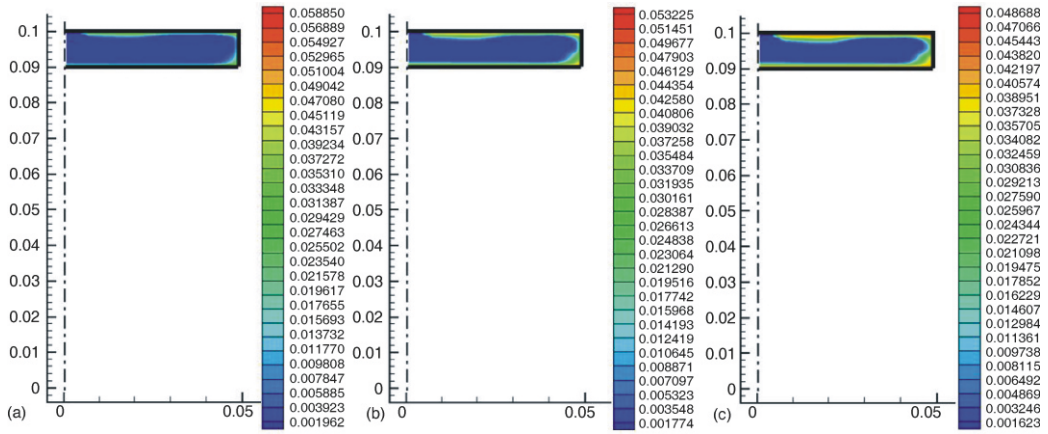

Figure 7. Mass fraction of fuel at 356 °CA. Effect of excess air coefficient λ: (a) λ = 0.9, (b) λ = 1.0, (c) λ = 1.1; (compression ratio r = 10:1; engine speed N = 24 rpm (color image see on our web site)

Conclusions

The purpose of this study is to investigate the effect of air/fuel ratio on combustion duration. A 2-D CFD code was developed for this purpose. This code simulates overall engine cycle – intake, compression, expansion, and exhaust strokes – with combustion. The authors suggest ignition timing advance (ITA) formula to determine the ITA according to the code simulations. In turbulent combustion eddy break-up model was used. Combustion durations of the rich and stoichiometric mixtures are the same due to the maximum amount of fuel that can be burned is the stoichiometric ratio of that fuel. However, combustion duration of the lean mixture is longer than stoichiometric mixture combustion duration. In laminar flow conditions Arrhenius type combustion model was used to model the combustion.

Acknowledgment

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Nomenclature

a – model constant, [–]
B – bore, [m]
b – model constant, [–]
C_1 – turbulence model constant, [–]
C_2 – turbulence model constant, [–]
C_R – model constant, [–]
C_R – model constant, [–]
C_μ – turbulence model constant, [–]
c – model constant, [–]
E – activation energy, [Jmol^{-1}]
H_j – formation enthalpy of j^{th} component, [kJkg^{-1}]
h – enthalpy [kJkg^{-1}]
k – turbulence kinetic energy, [m^{2}s^{-2}]
M – molecular weight of gas, [kgkmol^{-1}]
m_j – mass fraction of j^{th} component of mixture, [kgkg^{-1}]
m_f – mass fraction of fuel in air/fuel mixture, [kgkg^{-1}]
m_{ox} – mass fraction of oxygen in air/fuel mixture, [kgkg^{-1}]
m_{pr} – mass fraction of products in mixture, [kgkg^{-1}]
P – pressure, [kPa]
R – gas constant, [Jmol^{-1}K^{-1}]

R_f – source term due to reaction, [kgm^{-3}s^{-1}]
R_{fu} – turbulent dissipation rate of fuel, [kgm^{-3}s^{-1}]
R_{ox} – turbulent dissipation rate of oxygen, [kgm^{-3}s^{-1}]
R_{pr} – turbulent dissipation rates of products, [kgm^{-3}s^{-1}]
S_t – turbulent source term, [kgm^{-2}s^{-2}]
s – stoichiometric oxygen value for unit mass of fuel, [–]
T – temperature, [K]
u_z – z-axis (axial) velocity component, [ms^{-1}]
u_r – r-axis (radial) velocity component, [ms^{-1}]

Greek letters

ε – turbulent kinetic energy dissipation rate, [m^{2}s^{-3}]
λ – excess air coefficient
μ_{eff} – effective viscosity of the fluid, [kgm^{-1}s^{-1}]
μ_t – turbulent viscosity, [kgm^{-1}s^{-1}]
ρ – density, [kgm^{-3}]
σ_k – turbulence model constant
$\sigma_{n,t}$ – turbulent Prandtl number
σ_ε – turbulence model constant

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