Simulation a Natural Gas Direct Injection Stratified Charge with Spark Ignition Engine

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Abstract

The purpose of present paper is simulation a direct injection stratified charge natural gas engine. The KIVA-3V code was used for gaseous fuel injection simulation. Compression and expansion stroke of engine cycle is simulated using KIVA-3V code. In cylinder fuel equivalence ratio distribution criterion is used for studying mesh independency. The results show that 550000 cells number is sufficient. The amount of NO emission in the end of closed cycle simulation was found equal 674.875 ppm and In cylinder pressure versus engine crank angle degree was simulated that maximum value found in 366 oCA that equal to 27.3222 bar.

***Keywords**: Simulation, Engine, Natural Gas, Direct Injection, Stratified Charge.*

1. Introduction

With increasing concerns about the harmful effects of conventional fossil fuel emissions, such as those from gasoline and diesel fuel, natural gas has become a very attractive alternative fuel to power prime movers and stationary energy conversion devices [1]. For reciprocating engines, natural gas has emerged to be a promising alternative fuel due to its lower cost, clean burning quality. The emissions of particulate (PM), nitrogen oxides (NOx), and carbon oxides could be significantly reduced for natural gas compared to traditional hydrocarbon fuels [2].

Technologies for natural gas applied in homogeneously premixed spark ignition (SI) engines are more mature than those in direct injection engines. But premixed operation has several drawbacks in terms of engine performance. Premixed SI engines generally have lower power output than the same size DI engine due to detonation-limited brake mean effective pressure (bmep) capability. Premixed SI engines also suffer lower thermal efficiency than the traditional DI engine due to detonation-limited lower compression ratio and high intake air pumping losses resulting from the need to throttle the intake air pressure at the part load conditions [3].

Simulation is an important tool for engine research work. It not only helps to understand the physics for the processes, but also provides the means for predicting performance that can be used in hardware development optimization, saving cost and time compared to experimental optimization. Much simulation work has been conducted on DING engines [4; 2; 5].

The future gas engines will be equipped with high pressure direct injection systems. Like the gasoline direct injected engine also the gas engine can operate mainly in two modes, which can create: a homogenous charge for full loads and a stratified charge for part loads. Injection of the gas fuel for full load takes place during the induction stroke after opening of inlet valves and this operation do not require so high injection pressure. For part load the fuel is injected during the compression stroke forming a bigger concentration of fuel near spark plug located in the central axis of the cylinder head. The timing of injection should correlate with the piston position BTDC and engine speed in order to enable the adequate stoichiometric mixture near the spark plug during the ignition. This mode requires higher injection pressure than in the first one. The stratification of the charge depends on the location of the injector and the angles of injection nozzles. The most important problem is to fulfill the dose of fuel especially for the gas injection during compression stroke for high speed operation.

Some researchers are studying Feasibility, performance and combustion characteristic of CNG DI Stratified Combustion Using a Spark-Ignited

Rapid Compression Machine [6; 7; 8]. The compression ratio of this machine is 10:1 and its chamber is disc.

This studying results show that direct injection combustion realizes shorter burn durations in both initial and main stages of combustion, which must be due to both the stratified mixture formation and the turbulence generated by fuel jet. The main burn duration is decreased to half to one tenth of the case of homogeneous mixture combustion.

A single cylinder engine was modified into a natural gas direct-injection engine in [9]. the injector used in the study is a modified version from a gasoline direct-injection engine made by the manufacturer (Hitachi Co.). To increase the flow rate for natural gas application, the swirler near the tip of nozzle was taken off. This paper results show that fuel injection timing had a large influence on the engine performance, combustion and emissions and these influences became largely in the case of late injection cases. Over-late injection would supply insufficient time for the fuel–air mixing of the late part of the injected fuel, bringing poor quality of mixture formation and subsequently resulting in the slow combustion rate, the long combustion duration and high HC concentration. Also this paper results show that there existed an optimum fuel injection timing where the maximum cylinder pressure, the maximum rate of pressure rise and the maximum rate of heat release got their highest values along with the shortest combustion durations, the shortest heat release duration and more concentrated heat release process closing to the top-dead-centre while maintaining the low level of HC and CO emissions.

The Cracow University of Technology will provide tests with direct injection of CNG on one-cylinder motorcycle 4-stroke engines SUZUKI DRZ400S adopted for natural gas direct injection stratified charge engine. Metianiec in [10] use this engine for test and results show that Realization of CNG direct injection for high loads requires a significant technical development of a new type of gas injector and utilization of the initial charge tumble or swirl at stratified charge operation and The injection pressure should be higher than 35 bars for both injection modes.

In this paper Compression and expansion stroke of engine cycle is simulated using KIVA-3V code and the simulated results is calibrated using experimental data of paper [11] from politecnico di torino advanced internal combustion engine lab. Equivalence distribution criterion in combustion chamber is used for studying mesh independency. Engine combustion chamber pressure and NO emission is predicted using simulation.

2. The Gaseous Injection Model

The simulations will be performed using the KIVA-3V code [12] that has been modified for gaseous injection [13]. The gaseous injection has been simulated by specifying a time-varying mass flow, velocity, specific internal energy, turbulent kinetic energy and species concentration upon entry. This information is converted to appropriate source terms which are applied to the equations of mass, momentum, specific internal energy and turbulent kinetic energy. The source terms are explicit in nature and get updated each time step in order to simulate as continuous an injection profile as possible. The governing equations, modified for gaseous injection, are summarized below:

$$\frac{\partial \rho_m}{\partial t} + \nabla \cdot (\rho_m U) = \nabla \cdot \left[\rho D_t \nabla \left(\frac{\rho_m}{\rho} \right) \right] + \dot{\rho}_m^c + \dot{\rho}_m^g \tag{1}$$

$$\frac{\partial (\rho U)}{\partial t} + \nabla \cdot (\rho U U) = -\frac{1}{\alpha^2} \nabla P - A_0 \nabla (2/3 \rho k) + \nabla \cdot \sigma + F^g + \rho g \tag{2}$$

$$\sigma = \mu_{eff} [\nabla U + (\nabla U)^T] - \frac{2}{3} \mu_{eff} \nabla \cdot U \bar{I} \tag{3}$$

$$\mu_{eff} = \mu + \mu_t \tag{4}$$

$$\frac{\partial (\rho I_{\text{int}})}{\partial t} + \nabla \cdot (\rho U I_{\text{int}}) = -P \nabla \cdot U + (1 - A_0) \sigma \nabla U - \nabla \cdot J + A_0 \rho \varepsilon + \dot{Q}^c + \dot{Q}^g \tag{5}$$

$$J = -K \nabla T - \rho D_t \sum_m h_m \nabla \left(\frac{\rho_m}{\rho} \right) \tag{6}$$

3. Engine Specification

The engine was used for simulation in this paper is an optical-access single cylinder engine (SCE) from the paper [11], which was designed and The implementation of the gaseous source terms in the discredited governing equations was made possible by introducing the sources in the boundary cells of the computational domain. For each injection time step, the mass of the injection cells is updated using Eqs. (7)- (9),

$$\rho_{g,ic,n+1} = \rho_{g,ic,n} + \frac{\dot{m}_g dt}{vol_{ic,n}} \tag{7}$$

$$\dot{m}_{ic,n} = \rho_{g,n} vol_{ic,n} \tag{8}$$

$$\dot{m}_{ic,n+1} = \rho_{g,n+1} vol_{ic,n} \tag{9}$$

The velocity of the injection cells is kept constant using:

$$u_{ic,n+1} = u_{ic,n} = u_g \tag{10}$$

Where n and n+1 fall in the interval $\frac{t_{1,inj}}{dt}, \frac{t_{3,inj}}{dt}$, the specific internal energy is updated based on the enthalpy of the incoming jet using Eq. (11),

$$I_{\text{int},ic,n+1} = \frac{I_{\text{int},ic,n} m_{ic,n} + \dot{m}_g dt I_{\text{int},g}}{m_{ic,n+1}} \tag{11}$$

And the incoming turbulent kinetic energy (TKE) is imposed to be 10% of the incoming jet velocity squared, using Eq. (12).

$$K_{ic,n+1} = \frac{K_{ic,n} m_{ic,n} + \dot{m}_g dt 0.1 u_g^2}{m_{ic,n+1}} \tag{12}$$

Developed in order to experimentally investigate the jet evolution and the mixture formation in this type of engine, as well as to generate the experimental database for the validation of numerical simulation results.

4. Computing mesh generation

Along with KIVA-3(V), there is a pre-processor called K3PRP. K3PRP has been applied for simple or simplified engine geometry. In K3PRP, the input of engine geometry is in text format either in a form of function or sets of discredited points of the surfaces or curves. Time consumption is another concern of K3PRP. In order for 3-D simulation to be an engineering tool, the mesh must have high geometry fidelity, and the mesh generation process needs to be very efficient. In order to reduce the mesh preparation time for 3-D simulation of in-cylinder phenomena of internal combustion engines, a rapid mesh generation and dynamic mesh management methodology has been developed by ICEMCFD. With this new methodology, the mesh generation time for engines with moving Boundaries is within a week with ensured engine geometry fidelity. The current dynamic mesh management algorithm has been successfully applied for wide range of different engine geometry. It is universal or generalized now and we use that for mesh generation. Fig. 1 shows this engine mesh that is generated in ICEMCFD.

Table 1 –Engine characteristics [11].

Number of cylinders	1
Bore	82 mm
Stroke	85 mm
Con rod length	136.5 mm
Cyl. displacement	449 Cm3
Compression Ratio	8.7

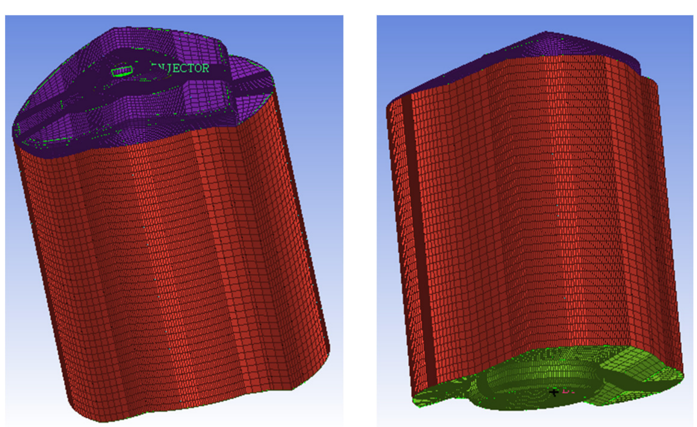

Fig1. Engine mesh generated in ICEM-CFD

Table2. Engine initial conditions

In cylinder initial pressure	bar 0.83
Intake valve close time	120 ^{oCA} BTDC
End of injection	50 ^{oCA} BTDC
In cylinder initial temperature	347 ^{K}
Exhaust valve open time	120 ^{oCA} ATDC
Injection duration	20 ^{oCA}

Table3. Engine boundary conditions

350 ^{K}	In cylinder wall temperature
370 ^{K}	Piston crown wall temperature
370 ^{K}	Cylinder head temperature

Table 4. Characteristics of used computer

CPU	RAM
Intel(R) Core(TM) i7	6 GB

5. Initial and boundary conditions

The initial condition of engine in 2000 rpm is presented in Table 2 and engine wall boundary conditions is presented in table 3. The initial turbulence kinetic energy in KIVA-3V code is calculated using deduction of kinetic energy of mean piston speed that was tacked to be tkei=0.6 in this simulation. The initial turbulence length scale () was tacked equal to minimum high of the computing cell in input file (scli=0.048). Injection velocity was input using injection velocity table with choosing pulse=3.

6. Mesh independency studying

The computer characteristics that used for this simulation is presented in table 4.

Mesh independency Simulation results was shown in Fig.2. We see that the equivalence ratio contours in 5oCA after start of injection for cells number of 550000 and 650000 is the same. Then we use cells number 550000 and continue simulation. The equivalence ratio contours in 5oCA after start of injection from simulation is shown in comparison to experimental results from paper in fig.3. There is a good agreement between numerical and experimental results.

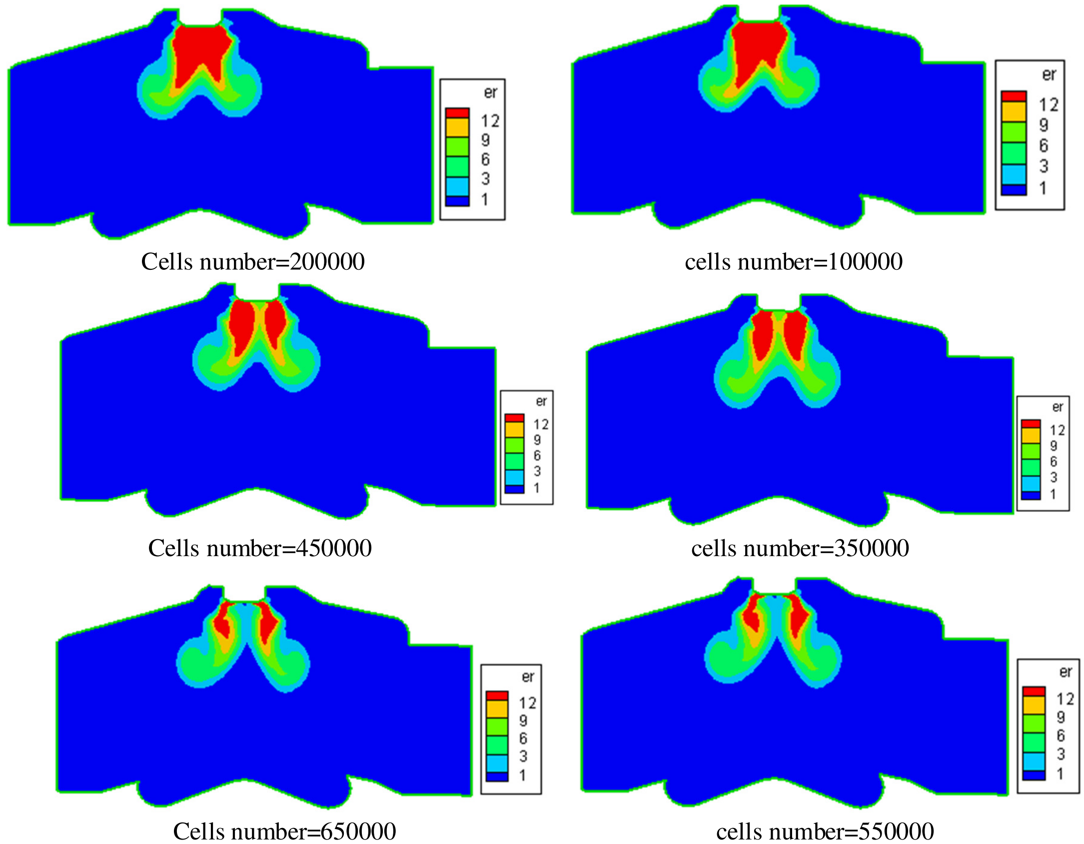

Fig2. Mesh independency results

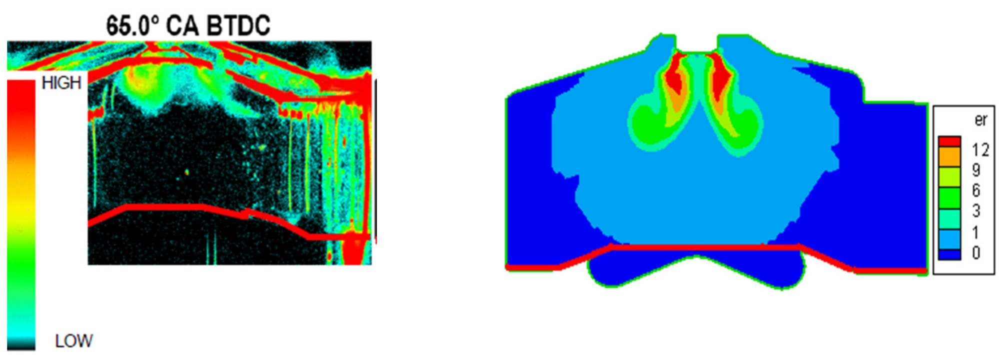

Fig3. equivalence ratio contours results in comparison to experimental

Now, we can study engine performance results using simulation. Simulated NO Emission is shown in Fig4 that show that the amount of NO in exhaust valve opening is less than 700ppm.

In cylinder temperature contours after start of combustion (Spark advance is 20 oCA) was shown in Fig 5. From this result we see that combustion propagate in chamber, but it cannot propagate to the cylinder wall exactly and it must be modify mixture preparation. We can modify injector or combustion chamber shape for overcoming.

In cylinder pressure versus crank angle was shown in Fig 6. we see that in cylinder maximum is 27.322 bar at 366 oCA.

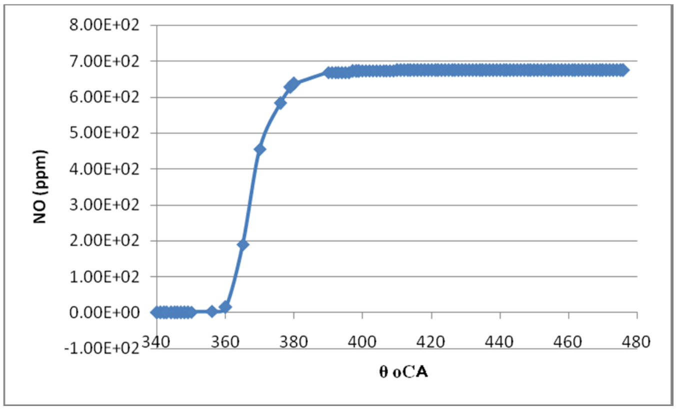

Fig4. NO emission from simulation results

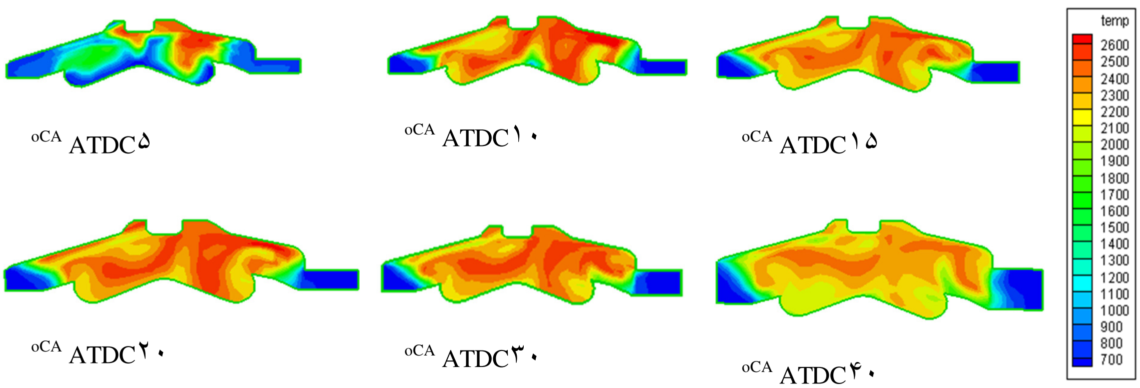

Fig5. Incylinder temperature contours

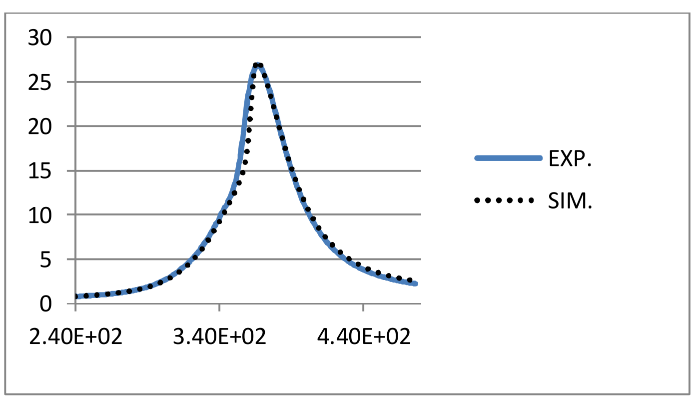

Fig6. In cylinder pressure versus crank angle

Conclution

The present work was focused on the numerical investigation performance of a research transparent single-cylinder engine. The simulation results can be concluded as below:

- In this paper natural gas direct injection stratified charge engine in closed cycle was simulated using KIVA-3V code.
- In cylinder fuel equivalence ratio was used for mesh independency studying and results show that cells number equal to 500000 is sufficient for this case simulation.
- The amount of NO emission in the end of closed cycle simulation was found equal 674.875 ppm.
- In cylinder pressure versus engine crank angle degree was simulated that maximum value found in 366 oCA and equal to 27.3222 bar.

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