Turbulent Premixed Combustion in SI Engine

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Abstract

The turbulent lean premixed combustion simulation is implemented in 4- stroke spark ignition (SI) engine. The Turbulent Flame speed Closure model (TFC) is used in different turbulent flow conditions. The model is tested for a variety of flame configurations such as turbulent flame speed, the heat release from the combustion and turbulent kinetic energy in the radial direction of the cylinder at 15.5 mm below the top dead center TDC point. The simulation performs in the three cases of the (intake / exhaust) valve timing. The exhaust valve case is an essential leverage on the turbulent flame specification. The combustion period is very important factor in SI engine which is controlled especially by the turbulent flame speed. The turbulent flame speed and heat transfer is ascendant less than 10 % and 3% in case of intake and exhaust valves are closed respectively. Moreover, the results show that the brake power enhances less than 4% and more than 40% with increase fuel temperature 60 K and engine speed 3000 rpm respectively.

1. Introduction

The main goal of the combustion researches currently is to reduce emissions and to ameliorate the fuel economy [1, 2 & 3]. Lean combustion is a promising ersatz to traditional combustion [4]. However, it demonstrate high challenges, especially for spark ignition engines (SI) [5, 6 &7]. Therefore, model utilized to predict such combustion has to be accurate, reliable and sever [1, 8, 9,10 &11]. When burning the fuel with an oxidizer the main of energy conversion for this transportation is domestic heating and power generation [8, 12, 13 & 14]. In addition, most combustion processes are used fossil fuels, which participate to more 80% proportion of the universal energy consumption in 2010 (IEA, 2011). Then Fossil fuels are especially catchy for the transportation section, with more 90 % of all fuels used for transport in 2010 existence on fossil fuels. Furthermore, it's predicted for more 80% proportion in 2035 (IEA, 2011). In addition, most of an engine in vehicles today is used in internal combustion (IC) engines, which burn fossil fuels to convert chemical energy into mechanical energy. The problems accompanied with the combustion of the fuels are pollutants e.g. (soot, unburned hydrocarbons, carbon monoxide, oxides of nitrogen and sulfur oxides). Moreover, combustion caused the most concentrations of (CO_2) in the atmosphere. These issues have grown from the fact that fossil fuel resources are limited. Therefore, they are an essential to reduce emissions and increase combustion efficiencies [1].

In premixed combustion the fuel and air are mixed homogeneously before entering the intake section [15, 16, 17, 18, 19, 20 &21].The simplest pattern of this combustion is the spark-ignition (SI) engines and the gas turbines for power generation. The ultimate challenge connected with turbulent combustion is to understand the interaction between combustion and turbulence over a broad range of length and time scales [22 &23]. One of the important magnitudes for turbulent premixed combustion is the turbulent flame speed [24, 25, 26, 27 & 28], which is indicated for the mixture will be burnt.

L. Liang and R. Reitz [2] used KIVA-3V modeling for (SI) engine simulation to predict the fuel oxidation and pollutants formation. They found the prediction of the cylinder pressure and engine out gases and compared with the experiments gives good agreement.

C. Hall, et al [3] used the generalize combustion phasing model for fuel flexible in SI engines. They found the efficiencies of combustion are perfect tied maximum CA50 (crank angle when 50 % of fuel is burned) for an engines.

The main objective of the present work is to study the demeanor and foretell trend of the turbulent flame speed, turbulent kinetic energy, the heat released and fuel temperature from the combustion in the cylinder spark ignition

engines at different valve timing in order to reduce exhaust emissions and increase engine performance.

2-Engine Specification and Computational Analysis

The two main valves, timing in SI engines, intake & exhaust valves are depicted in Figure 1. This study is performed in the line below TDC point at 15.5 mm. The parameters of the turbulent specification are implemented in the TFC model. The two sections are done within the combustor is depicts in Table 1. The two sections represent each part of the combustor which contains in the sections that is used in the simulation and explained in the results and in Figure 3.

The changes in intake & exhaust valves, timing periods are explained in Figure 2. Exhaust valve opening & closing times were fixed. The exhaust valve was opened in point (3) and closed in point (1). The intake valve was opened in point (4) and closed in point (2). The high compression ratio allowed alteration in valve activity during the overlap duration into TDC point to avert valve to contact the piston [4].The summary of valve timing within crankshaft angle CA is restricted in Table 2. Moreover, the initial and boundary conditions used within the TFC model, meanwhile the fuel types and specifications are listed in Table 3.

In this study, we use 3-D simulation of turbulent combustion modeling in SI engine. This modeling becomes an essential tool in the analysis and design of the high efficiencies and low emissions from the engines. The grid is unstructured polyhedral mesh with total number of cells and faces of each part as the follow: Body 1 : 26242 cells, 150951 faces, Body 2 : 942 cells, 4060 faces, Body 3 : 932 cells, 4071 faces and Body 4 : 768 cells, 3311 faces depicted in Figure 3. The simulations are carried out by the STAR CCM+ (v10.02) CFD software [30]. The realizable k-ε two layer model is used for turbulence modeling. The boundary conditions are included: velocity inlet condition in the injector of the fuel, pressure outlet in the valves when was opened and the wall when was closed.

The turbulent flame speed (S_T) utilized in the simulation can be calculated from the formulation of TFC reaction with the CFD software as below [30].

$$S_T = AG\left(S_L^{0.5}\alpha_u^{-0.25}\acute{u}^{0.75}l_t^{0.25}\right) \qquad (1)$$

Where, A is a TFC model constant, G is the flame stretch factor, α_u is the thermal diffusivity of the mixture, S_L is the laminar flame speed, and l_t turbulent length scale. Then the turbulent kinetic energy k which is represented the turbulence level within the combustor can be found from this equation [31].

$$k = 1.5(u')^2 \qquad (2)$$

where, u', is the root mean square of the velocity.

The heat release from burning gasoline can be determined from energy equation as [30].

$$HR = \sum_{i=1}^{n} h_i \, \rho_u S_T \, |\nabla Y_f| \qquad (3)$$

where, h_i is the heating value of gasoline, ρ_u is the unburnt density and Y_f is fuel mass fraction as listed in Table 3.
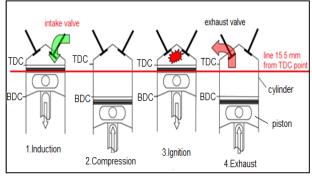

Figure 1: 4- stroke engine stages with valves and the 15.5mm line location

Table 1 Sections of the combustor and its parts.

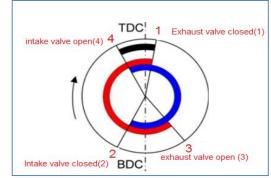| Section	Fuel injector	Intake valve	Exhaust valve	Cylinder
1	✓			✓
2		✓	✓	✓

Figure 2: Valve timing diagram [29]

Figure 3: the mesh of the cylinder and valves

Table 2 valves timing specification.

valve	Open	Closed
Intake	19° CA	38° CA
Exhaust	59° CA	8° CA

Table 3 Initial and boundary conditions and type of fuel and its specification used in the TFC model.

Equivalence ratio	Initial temperature[K]	Fuel mass fraction	Oxygen mass fraction	Nitrogen mass fraction
0.6	300	0.038	0.235	0.726
Fuel type	heating value of the fuel [kJ/g]	TFC model constant(A)	Turbulent kinetic energy k [J/kg]	Turbulent dissipation rate [m2/s3]
Gasoline C8H18	47.8	0.37	0.0324	0.04

3. Results and Discussion

The turbulent flame speed (S_T) utilized in the TFC model for a better description of the ignition flame evolution. The distribution of S_T in the line 15.5 mm below TDC point in the radial direction of the cylinder within valves timing i.e. when the exhaust valve is closed and both valves (intake / exhaust) are closed is depicts in Figure 4. It's clear that S_T is high values at the middle region and low quantity at regions near the wall within the cylinder due to the stagnation of the flow. The lower S_T occurs when the exhaust valve is closed case only.

Figure 5 shows the heat release from the combustion at two cases of the valve timing. The heat released when the exhaust valve is closed at the 8° CA is lower heat released than both valves are closed case in the radial-direction within the cylinder at 15.5 mm line below TDC point. While Figure 6 shows

distribution of the temperature in the 15.5 mm line below the TDC point within valves timing. The minimum temperature occurs when the exhaust valve is closed only case and maximum temperature reached 1600K. In addition, the heat transfer is enhancement by 2.94%.

The turbulent kinetic energy (k) is very important parameter in turbulent flame in SI engine. Figure 7 shows the k contour in sections 1 and 2 within the cylinder at the exhaust valve closed. While the inverse case happened when close the intake valve and the exhaust valve is open as depicted in Figure 8. The S_T used in the model for high description of the ignition flame development. The contours of the, S_T are depicted in Figures 9 and 10 with different cases of valve timing which are listed in Table 2. The exhaust valve case has important effect on S_T parameter on the flame spread.

The influences of fuel temperature [K] and engine speed [rev/min] on the engine performance and brake mean specific fuel consumption (BSFC) with 2500 [rev/min] are depicted in Figures 11and 12 respectively. Higher values of fuel temperature are produce utmost conditions for both engine performance and exhaust emission. Hence, the brake power (rate of work of the engine) enhances by 3.22% for 2500 rpm with increase fuel temperature 60 K. Meanwhile, enhances 43% from change 2500 rpm to 5500 rpm. The minimal BSFC (252.1 g/kW-hr) was acquired at minimal temperature of 330 K, while the maximum BSFC (353.6 g/kW-hr) was acquired at temperature of 343 K. The higher brake mean specific fuel consumption is due to the lower energy content of the fuel. As the temperature increases, the energy content also increases, causing the lowest BSFC for a temperature of 330 K.
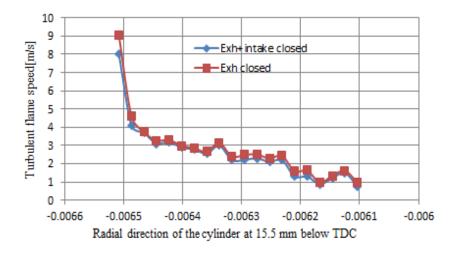
Figure 4: turbulent flame speed S_T distribution at the line 15.5 mm below TDC
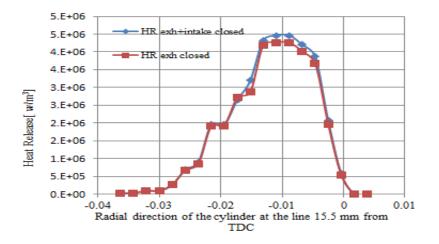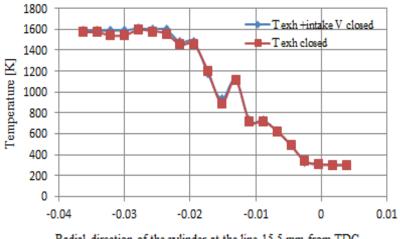

Figure 5: heat release at the line 15.5 mm below TDC point

Figure 6: temperature distribution at the line 15.5 mm below TDC point
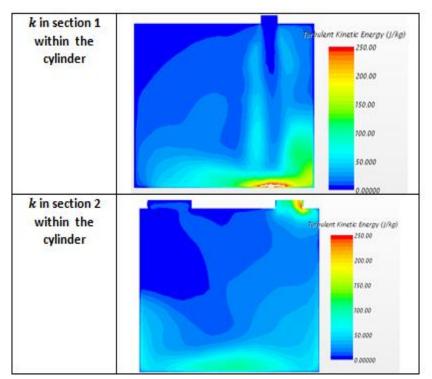
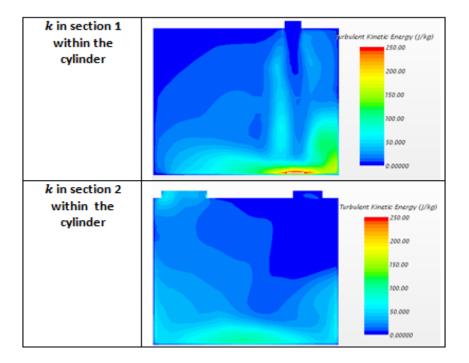**Figure 7:** Turbulent kinetic energy (k) contour in sections 1 and 2 in the cylinder at exhaust valve closed

Figure 8: Turbulent kinetic energy k contour in sections 1 and 2 in the cylinder at exhaust valve open
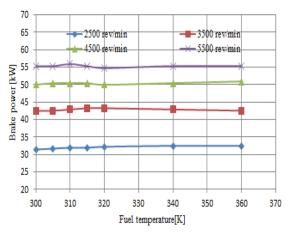
Figure 11: brake power of the engine at different fuel temperature
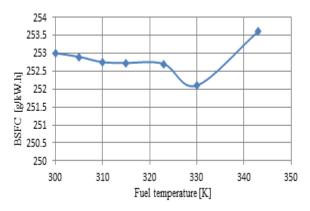and engine speed

Figure 12: the brake mean specific fuel consumption in different fuel
temperature at engine speed 2500 [rev/min]

4. Conclusions

The turbulent combustion simulation is performed in four stroke SI engine. The simulation performed by using STAR CCM+ v 10.02 software. TFC model used in different turbulent flow conditions. The turbulent flame speed, S_T turbulent kinetic energy k and the heat released from the combustion in the radial direction of the cylinder at 15.5 mm below the TDC point are studied with the influence of the valves timing. The simulation performs within cases of the (intake / exhaust) valve timing. We found that the exhaust valve case is a very important leverage on the turbulent flame specification. The turbulent flame speed is growing by 8.31% in case of intake and exhaust valves are closed. The heat transfer is enhancing by 2.94% when intake and exhaust

valves are closed. The results show that at higher values of fuel temperature produces maximum stipulation for both engine performance and exhaust emission.

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