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Original Research Article

The Investigation of Velocity and Temperature Changes by CFD Analysis for Intake Stroke of LPG Fuel Stratified Charge Engine

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Abstract

This study numerically investigates the in-cylinder speed and temperature variations for fuel-air mixture intake at the induction stroke by the LPG cylinder of an engine running on the stratified charge principle. The present study is based on the CFD model created using experimental data. Examinations were performed between the top dead center and the bottom dead center during the induction stroke. In-cylinder speed and temperature variations were obtained by k-ε turbulence model depending on the crankshaft angle and revolutions per minute (900, 3000, 5500 rpm). The obtained results were compared with the known theoretical values for stratified charge engines, and consistent results were obtained.

Key Words: Stratified Charge Engine, Gasoline Engine, CFD Analysis, Engine Performance

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1. Introduction

Studies on stratified charge gasoline engines started in 1920s. Such engines based on the combination of high specific output gasoline engines and high performance of diesel engines at partial load. For this purpose, many types of engines suitable for stratified charge system were developed. Common objective of these types of engines was to start the gasoline engines with a poor mixture. Basic principle of these engines is to generate an ideal (stoichiometric) mixture around the spark to combust easily and to start the gasoline engine with poor mixture by initiating combustion in this site. In this way, it is intended to reduce the specific fuel consumption of an engine starting with poor mixture.

Although a variety of automobile manufacturers (e.g., GM, Ford, Texaco, Volkswagen, BMW, Fiat, Honda, Mitsubishi, Toyota etc.) have offered many examples of stratified charge engines to the present day, no mass-production has taken place as with the classical motors (Weaving, 1990). This is because of the failure to provide an easily combustible rich mixture around the spark plug at overall load and speed regimes, and to safely combust the poor mixture [1].

Currently, commonly known stratified charge engine type is the Direct Injection (DI) systems. In DI systems, the fuel is gradually entered in the cylinder. Initially, a little amount of fuel is injected into the cylinder at the induction stroke to generate poor mixture, and the main injection is made to the sides near spark plugs at compression stroke immediately before ignition. As depicted in Figure 1, the amount of fuel around the spark plug is concentrated, thus a good ignition can be achieved [2,3].

LPG, an economical, low-emission and sustainable source for gasoline and diesel engines, has been used for over 40 years as an alternative fuel in gasoline and diesel engine automobiles and in light commercial vehicles. The use of LPG in internal combustion engines is an interesting case.

Although many studies are available in the literature on different fuel injection systems and different filling strategies using gasoline in gasoline engines, there are scarcely any studies performed using LPG especially in stratified charge systems. The primary problem with using LPG in direct injection systems is to deliver the gas fuel.

Therefore, a homogeneous charge gasoline engine was transformed into a stratified charge engine by making several modifications to the cylinder head and suction manifold of a gasoline engine, and favorable results were obtained [1]. This study demonstrates that improvements on performance, fuel consumption and emission values could be achieved by making modifications to different types of engines.

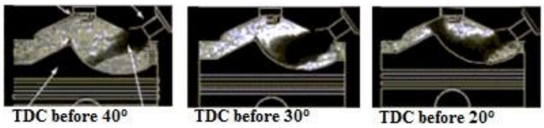


Figure 1. Spraying fuel into the cylinder compression time [8]

Besides DI systems, there are two-stage combustion mechanisms with significant advantages to reduce contaminant gas emission as well as to enhance combustion performance.

In this study, several modifications were made to a homogeneous charge gasoline engine and additional systems were used to obtain stratified charge engine which could be a good example for future two-stage combustions mechanics. In this experimental study, the engine was tested for performance and emission, and favorable results were obtained. In addition to performance testing, speed and temperature values of the mixture at charge was theoretically modelled by CFD (Computational Fluid Dynamics) packet program. Real values measured empirically were used when creating the model. Results from the theoretical and practical study were substantially consistent with each other.

In recent years, CFD techniques are used due to developing of numerical techniques and computer capabilities. The CFD is an indispensable tool for engine development with the increasing of computer capabilities. As is well known, the fluid motion inside a cylinder is turbulent, unsteady, cyclic and non-stationary both spatially and temporally [9, 10].

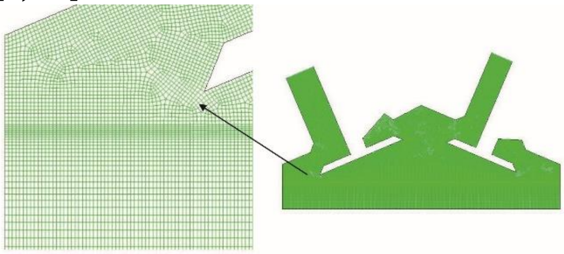


Figure 2. Numerical models are created by using the network structure for the Gambit program

The network topology created using the Gambit program for the numerical model is presented in Figure 2. Triangular and rectangular elements were selected for the network topology created using a dynamic mesh. The number of selected elements increases or reduces depending on the crank angle. The maximum number of meshes is 176484 and the minimum number of meshes is 87304.

Fluent was used for numerical analysis. Numerical analysis was performed considering that the flow was stable. Program analyzers in which pressure, pressure-speed relationship, momentum and turbulence kinetic energy variations were expressed by First Degree variables were used.

In this study the standard k-ε turbulence model is used. The major reason for using k-ε turbulence model is that this model has a large capability to respond more convergent and different problems compared to the other models.

A set of equations for momentum, energy and kinetic energy were used relating to temperature and flow variations of gases entered in the cylinder at the suction stroke.

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho V) = 0 \tag{1}$$

Momentum equations can be written in two directions as

$$\frac{\partial(\rho u)}{\partial t} + \frac{\partial(\rho u^2)}{\partial x} + \frac{\partial(\rho uv)}{\partial y} = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}\left(\lambda\left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right)V + 2\mu\frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left[\mu\left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right)\right] \tag{2}$$

$$\frac{\partial(\rho v)}{\partial t} + \frac{\partial(\rho v^2)}{\partial y} + \frac{\partial(\rho uv)}{\partial x} = -\frac{\partial p}{\partial y} + \frac{\partial}{\partial x}\left[\mu\left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right)\right] + \frac{\partial}{\partial y}\left(\lambda \nabla V + 2\mu\frac{\partial v}{\partial x}\right) \tag{3}$$

Energy equation

$$\frac{\partial(\rho e)}{\partial t} + \nabla(\rho e V) = \frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) \tag{4}$$

The turbulence kinetic energy, k, and its rate of dissipation, ε, are obtained from the following transport equations:

$$\rho\frac{Dk}{Dt} = \frac{\partial}{\partial x_i}\left[\left(\mu_{eff} + \frac{\mu_t}{\sigma k}\right)\frac{\partial k}{\partial x_i}\right] + Gk + Gb - \rho\varepsilon - Y_M \tag{5}$$

$$\rho\frac{De}{Dt} = \frac{\partial}{\partial x_i}\left[\left(\mu_{eff} + \frac{\mu_t}{\sigma\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_i}\right] + C_{\varepsilon 1}\frac{\varepsilon}{k}(Gk + C_{\varepsilon 3}Gb) - C_{\varepsilon 2}\rho\frac{\varepsilon^2}{k} \tag{6}$$

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{7}$$

$C\varepsilon_1$, $C\varepsilon_2$ and $C\mu$ are model constant. These coefficients are,

$C\varepsilon_1 = 1.44$; $C\varepsilon_2 = 1.92$; $C\mu = 0.09$; $\mu_k = 1.0$; $\sigma_\varepsilon = 1.3$

For mathematical model, a LPG converted, stratified charge, 4-stroke, 4-cylinder, water cooled, stratified injection and inline FORD Zetek engine was selected. The engine was loaded by Cussons P8602-brand hydraulic dynamometer, and performance tests were conducted in real operating conditions with throttle valve fully opened at full load. Modelling was made for three different ranges of revolutions (900, 3000 and 5500 rpm). Using data from the experimental measurement results, speed and heat transfer variations of different mixtures in the cylinder were modelled in two dimensions depending on the crankshaft angle. Table 1 shows data of the engine used.

Table 1. Test Engine Specifications

Specification	Description
Engine type	4 Stroke, Stratified Charge Gasoline Engine
Number of cylinders	4
Cylinder Diameter x Bore (mm)	89 x 95
Cylinder Volume (cm^3)	1600
Compression Ratio	11:1
Maximum Power 6000 rpm	kW
Maximum Moment 3300 rpm	Nm
Fuel system	Multi Point Injection
LPG System	Sequence Multi Point Injection
Fuel	Gasoline/LPG
Injector Spray Pressure (bar)	5
Ignition System	Electronic

The end-suction temperature of the mixture entered in the cylinder is higher than the ambient temperature due to waste exhaust gases and heat transfer from the mixing and walls. The end-suction temperature can only be calculated approximately from thermal balance equations. The suction temperature generated by mixing with waste gases is described as follows according to thermal balance condition.

$$T_e = \left(\frac{m_o(T_o \Delta T) + \varphi m_r T_r}{m_o + m_r}\right) \quad (8)$$

A set of equations for momentum, energy and turbulence kinetic energy was used in modelling temperature and speed variations of the mixture entered in the cylinder at suction and compression strokes depending on the value of crankshaft angle.

The engine used in the study had two intake valves. The diameter of the valves was different. Table 2 shows the diameter of the valves. The redirectors were placed on the front face of valves in order to control the charge flow profile entered in the cylinder.

The stoichiometric mixture came from the small valve, whereas the excessive poor mixture came from the large valve. The speed of air sucked into the cylinder is considered to be equal to the mean piston speed, and the mean piston speed and valve opening were calculated by the equation of continuity depending on the section area. The physical values used in the mathematical model are presented in Table 3.

Table 2. The intake valves dimension

Valves	Dimension (mm)
Small Intake Valve Diameter	22
Large Intake Valve Diameter	26

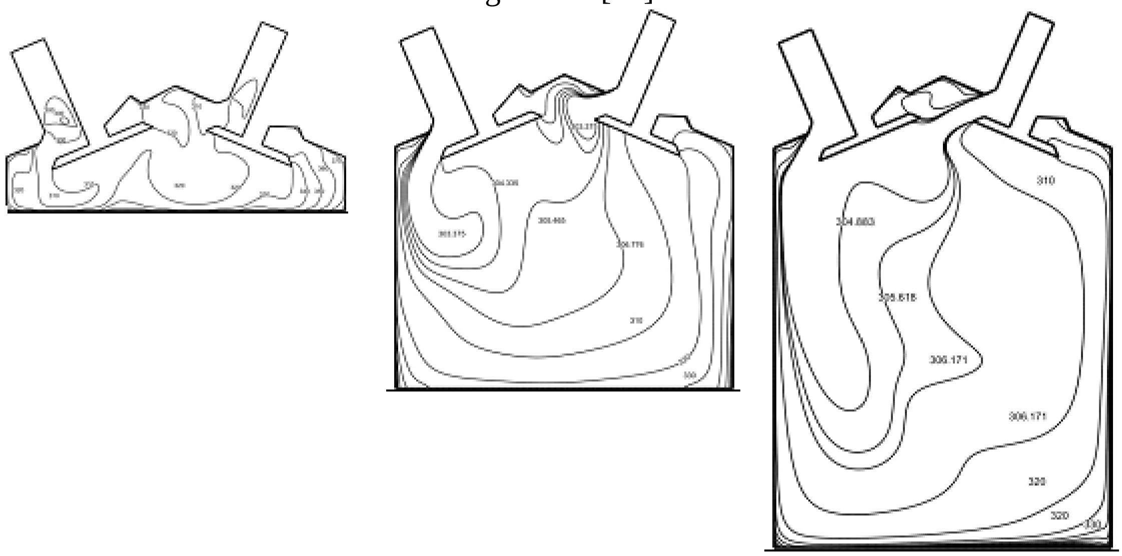
Table3. Boundary conditions

	900 rpm	3000 rpm	5000 rpm
Engine Cooling Water Temperature	362 K	362 K	362 K
Exhaust Gas Temperature	538 K	870 K	1054 K
Small Valve Mixture Velocity	98,8 m/s	139,8 m/s	214,6 m/s
Small Valve Mixture Velocity	29 m/s	41,73 m/s	63,15 m/s

The flow characteristics of engine models are considered as two dimensional. The dynamic models were investigated numerically by means of FLUENT commercial code [11]. It is a well-known Computational Fluid Dynamic program that is the science of predicting fluid flow, heat transfer, mass transfer, chemical reactions, and related phenomena by solving governing equations. Widely used in the field of internal combustion engine design, this program uses finite volume method in order to solve Navier-stokes and energy equations. The finite volume method can accommodate any type of grid. Thus, it is suitable for complex geometries, like present study [10]. The in-house CFD code developed incorporates the RNG k-ε turbulence model [12] with some slight modifications to introduce the compressibility of a fluid in generalized coordinates, as is described in Ref. [13]. It solves the transport equations for the conservation of mass, momentum, chemical species and energy [14].

4. Results and Discussion

A flow modelling was created for stratified charge engine at a volume where piston was replaced at crankshaft rotation angle A=30°, B=90°, C=180° after top dead center at 900, 3000 and 5500 rpm engine revolution. The temperature and speed values are presented in vectors. The initial modelling for stratified charge engine was made at induction stroke for variation in the temperature of the mixture taken into the engine. Temperature variation was studied in relation to the engine revolution and crankshaft angle parameters. The temperature of the intake air enables evaporation depending on the speed of the fuel in the mixture entered in engine cylinders or of the fuel injected by the injector into the cylinder. Each component of the fuel causes differences in cycles at different temperatures. On the other hand, the amount of unburned fuel will increase as a part of the fuel evaporates at a low temperature. In addition, the temperature of the sucked air or the mixture is known to influence the speed of the flame and thus engine power during combustion process [14].



A B C

Figure 3. Temperature distribution inside the cylinder at 900 rpm according to the change of crankshaft angle in the suction stroke

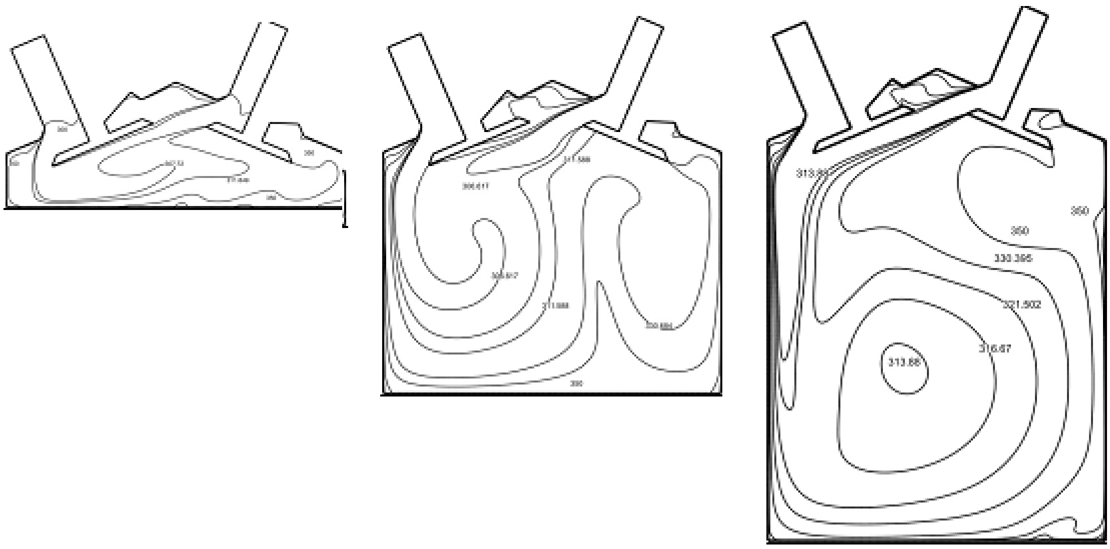


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Figure 4. Temperature distribution inside the cylinder at 3000 rpm according to the change of crankshaft angle in the suction stroke

Figure 1 shows the modelling for temperature variation value in respect to crankshaft angle for 900 rev-1. No significant differences existed in-cylinder mean temperature with increasing the volume of the cylinder. The period when the temperature remained stable at induction stroke could be considered isotherm. The homogeneity of cylinder content makes heat distribution more homogeneous at a real induction stroke. However, the heat is transferred from cylinder walls and back gasses to the mixture entered in the cylinder. Therefore, the temperature of the mixture slightly increases in sections near the walls and even significantly increases in areas near the exhaust valve.

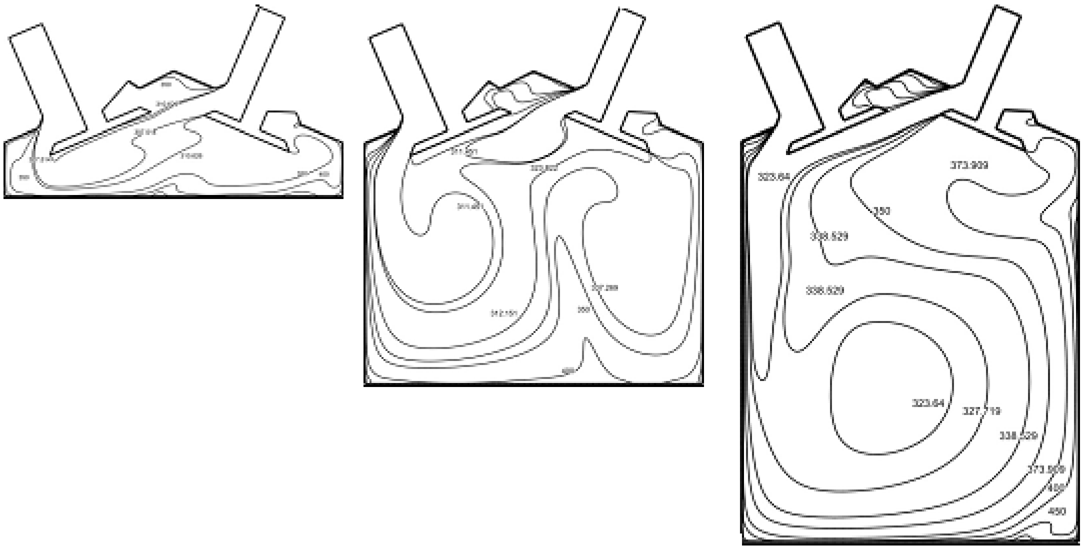

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Figure 5. Temperature distribution inside the cylinder at 5500 rpm according to the change of crankshaft angle in the suction stroke

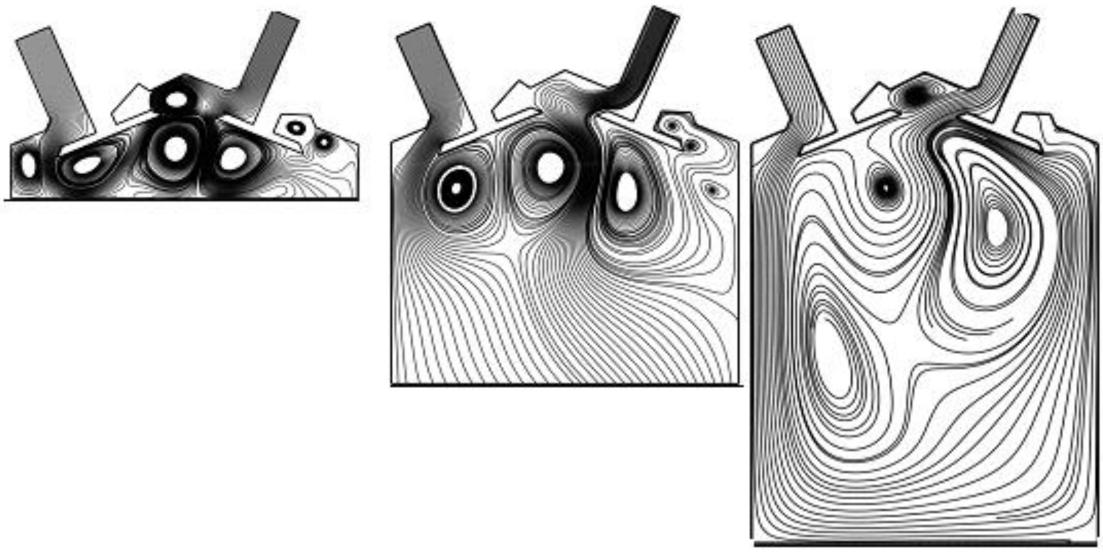

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Figure 6. In-cylinder speed distribution at induction stroke in respect to variations in crankshaft angle at 900 rpm

Figures 3, 4 and 5 show the heat distribution at different engine revolutions. No significant differences were present in the temperature variations at each of three revolutions. The temperature of cylinder walls increases depending on the increase in the number of cycles in unit time with increasing engine revolution. Heat transfer from the cylinder wall to suction charge is expected to increase, thus the charge temperature is expected to increase. However, it appears that revolution per minute doesn’t influence the charge temperature as a result of reduced time

needed for heat transfer from hot surfaces to the new charge with increasing revolution per minute. Increasing the temperature of charge at high revolutions is considered to result from back gasses.
Two different sections were created in the cylinder by modifications to the experiment engine. The section near the spark-plug was filled with the stoichiometric mixture in the combustion space and the other section was filled with excessively poor mixture in an attempt to run the engine with the poor mixture. In mathematical modelling, in-cylinder gas movements and resulting turbulence values are consistent with the results from the experimental study. Making the mixture poor yielded favorable performance results, particularly at low revolutions and partial engine loads.

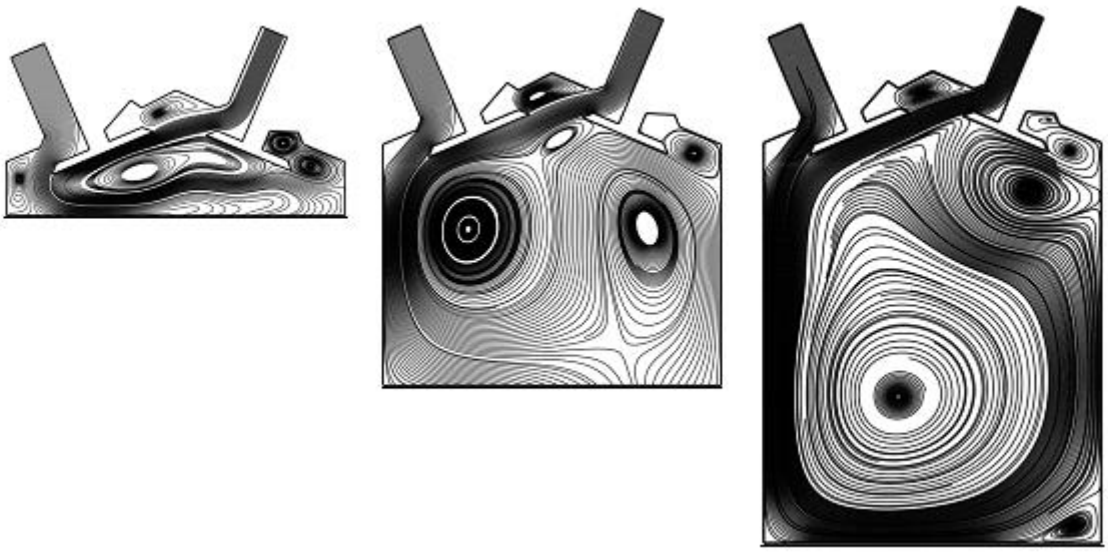

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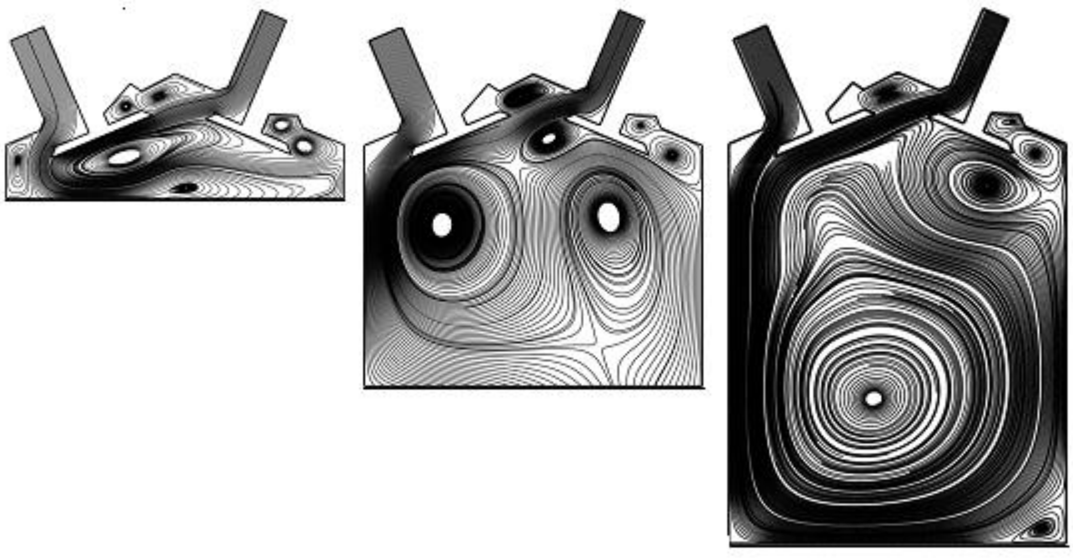
Figure 7. In-cylinder speed distribution at induction stroke in respect to variations in crankshaft angle at 3000 rpm



A B C

Figure 8. In-cylinder speed distribution at induction stroke in respect to variations in crankshaft angle at 5500 rpm

As depicted in Figures 6, 7 and 8, distribution of in-cylinder speed vectors for each of three revolutions was achieved as desired in different sections for different mixtures. Tendency to deterioration began due to accelerated gas movements for 5500 revolutions as seen in the figure. Even though obtained turbulence and distribution structure deteriorated, it is apparent that different mixtures were generated in the cylinder in the desired direction. Fresh gasses with different mix ratios taken in the cylinder at induction stroke were successfully forced to move in the cylinder as desired by means of redirectors located at the back of valves so as to create turbulence.

As depicted in all three figures, the section near the spark-plug was filled with the stoichiometric mixture in the combustion space and the remaining section of the cylinder was filled with the poor mixture. In this way, one of the major problems has been overcome, that is, slow rate of feed for the flame when gasoline engines are run with the poor mixture. As noted in the model, the stoichiometric mixture filling the combustion space easily ignites to generate the flame face, and the poor mixture filling the remaining section of the cylinder is ignited by this flame and the combustion is completed. The completed combustion is evidenced by experimentally measuring the resulting combustion products. In modelling figures, the presence of occurrence of turbulences in the cylinder at different engine revolutions and at different crankshaft positions made favorable contribution to the combustion.

Results

We investigated the variations in temperature and speed of the mixture sucked at induction stroke of a stratified charge engine using a model made by fluent program. The results from the study are provided below:

- Different mixtures entered in the cylinder through different valves and the distribution in the cylinder occurred as desired.
- Heat distribution in the cylinder was at an expected level and the results are consistent with the literature.
- Speed distribution and turbulence formation in the cylinder were in accordance with experimental results.
- Severity of turbulence varied at different engine revolutions.
- Heat distribution and section with turbulence varied at different positions of the crankshaft.
- Stoichiometric and poor mixture sections were generated at each engine revolution and at each position of the crankshaft.

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