Two-Stage Combustion Mechanism in Gasoline Engine for CO_2 Emission Reduction

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1 Introduction

Although internal combustion (IC) engines are not the only sources for CO_2 production, their contribution is significant. There is an agreement between European car manufacturers and European Community to reduce CO_2 emissions from new passenger cars to 120 g/km by 2012, meaning a 30% drop relative to 2000 emission level. In 2012, manufacturers will have to pay a penalty of €20 for every g/km CO_2 emission exceeding 120 g/km, multiplied by the number of vehicles sold. This penalty rises to €95 as of 2015. Such a target creates a significant challenge to the automotive gasoline engine, which typically produces higher CO_2 emission than the diesel counterpart as a result of poor part-load efficiency and low compression ratio. When all of these facts are taken into consideration it is obvious that CO_2 emission reduction in spark ignition engines is a challenging research area and a significant issue for global warming.

Stratified charge engines have the characteristics of the two most popular forms of combustion used in IC engines. A non-homogenous mixture is formed in the combustion chamber, like in diesel engines. Conventional gasoline engines have stoichiometric mixture in every part of the cylinder. However, stratified charge engines have rich mixture near the spark plug and lean mixture in the cylinder, globally. This non-homogenous mixture is obtained usually with the modification of the piston geometry. The geometry of the intake manifold can also be modified. Since there is a lean mixture in the combustion chamber globally, stratified charge engines have a lower knock tendency than the conventional gasoline engines. Due to this fact, the compression ratio (ε) of a stratified charge engine can be higher than the compression ratio of a conventional gasoline engine; $\varepsilon \geq 12$ is possible. A higher compression ratio leads to a higher efficiency. Moreover, the existence of a lean mixture in the combustion chamber leads to lower fuel consumption. CO_2 emissions from automobiles are related to fuel consumption. Improving the fuel efficiency of a vehicle leads to a decrease in CO_2 emission. As it is known, burning of stratified mixture could be a very effective way to increase the fuel economy and decrease CO_2 emissions in gasoline spark ignition engines. It was expected that stratified charge engines have a potential to attain 20% reduction in fuel consumption [1].

2 Two-stage combustion mechanism

A stratified charge engine operated with "Two-Stage Combustion Mechanism" is proposed in Azerbaijan Technical University (AzTU) and has been developing in cooperation of Warsaw

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Technical University (WTU), ITU and METU. In Figure 1 are given scheme of the combustion chamber that allows two-stage combustion mechanism and example of photographs of combustion realized in a physical model [2,3]. As shown in this figure, this combustion chamber looks like “8” and is separated into two zones. Twin turbulence motion occurs in opposite directions during the intake and the compression cycle of the engine. While the spark plug mounted part of the combustion chamber contains fuel-rich mixture with A/F ratio (λ) of 0.6-0.8 values, the other part contains pure air. The equality of swirl moments and rotational speeds do not allow the mixing of two zones until ignition time and this preserves the stratification of air-fuel mixture at all loading regimes of the engine.

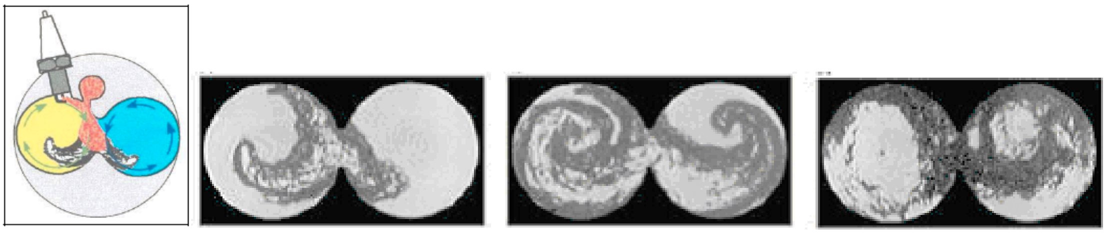

Figure 1. Two-stage combustion mechanism in twin swirl combustion

Since the swirl motion occurs with the start of the intake cycle, air-fuel mixture can be prepared in the intake manifold (outside of cylinders). Therefore current electronic injection systems or carburetor engines can be used with this method; in other words, special and expensive direct injection systems are not required like in gasoline direct injection (GDI) engines, where the injection of fuel into the cylinder reduces the time available for evaporation and mixing. Currently the most widely used injector for GDI applications is the single fluid, swirl-type unit, that needs to operate in a fuel pressure in the range of 7-10 MPa.

Another advantage of the two-stage combustion mechanism, when compared to the other types of stratified charge engines, is the fact that it has the potential to reduce other exhaust gases emissions in addition to the reduction of CO_2 emission. In two-stage combustion mechanism, gasoline is injected into the intake manifold instead of cylinders, therefore liquid phase of gasoline does not contact with the cold walls of the cylinders. Due to this fact, stratified charge engines that have twin swirl combustion chamber produce lower HC emissions. Furthermore, incomplete combustion products (CO and H_2) produced during the combustion of the rich mixture (λ=0.6…0.8) at the first stage can be burned in the second stage of combustion with the effect of turbulence so detonation firmness is satisfied. Realization of combustion in two-stage also provides an advantage to decrease NO_x emissions. Lack of oxygen in rich mixture and low combustion temperature at the first stage of combustion do not allow NO_x formation. At the second stage, since incomplete combustion products (CO and H_2) have been burned quickly, nitrogen have not enough time to oxidize and NO_x formation decreases.

The spread out of the flame front for the two-stage combustion mechanism is observed by using high-speed photography methods in WTU and example of these photographs are given in Figure 1. As a result of series of experimental investigations it is found that swirl velocity must be between $\omega \approx 600$ s^{-1} and $\omega \approx 1500$ s^{-1}.

3 Theoretical calculations

The CO_2 emissions of conventional and stratified charge engines are compared using a real cycle gasoline internal combustion engine mathematical model. Details of this model are given in the reference [4]. This comparison is carried out for partial loads. The CO_2 emissions are calculated as kg/h emissions for 76.5 mm diameter, 86.9 mm stroke 1.6 liter four cylinder engine. The CO_2 emission for the conventional gasoline engine is taken as reference. The CO_2 emission reductions obtained with the stratified charge engines are given relative to this reference value. The simulation results are given in Table 1 for partial load operation. λ is the air-to-fuel ratio, which is the inverse of equivalence ratio ϕ. η_v is the volumetric efficiency, and is higher for stratified charge engines due to the absence of a throttle valve. The values for compression ratios and air-to-fuel ratios are chosen such that the power output calculated for each stratified charge engine simulation is equal to the power output calculated for the conventional gasoline engine. For the partial load operation, CO_2 emission reduction can go up to 19.2%. The reduction of CO_2 emission in partial loads is especially critical due to the fact that engines are usually operated in this mode, especially in urban traffic conditions. The reduction of CO_2 emissions is mainly due to three factors: 1) leaner fuel-air mixtures; stratified combustion, 2) higher volumetric efficiency due to the reduction of pumping losses at partial load conditions and 3) the improved thermal efficiency due to the usage of higher compression ratios. This reduction can be increased by using higher compression ratios and higher stratification rates (air-to-fuel ratios). In practice, the use of high compression ratios is limited by the occurrence of detonation, and the use of very lean mixtures can lead to inappropriate conditions for the flame propagation.

Table 1. Reduction of CO_2 emission for partial load operation

Partial load	INPUTS				OUTPUTS	
n = 2000 min^{-1}	η_v	ε	λ	P_e, kW	b_e, g/kWh	Relative CO_2 reduction
Conventional	0.60	9.0	1.00	17.9	264	0 (reference)
Stratified charge	0.95	10.0	1.70		239	9.8 %
		11.5	1.80		224	14.8 %
		13.0	1.90		212	19.2 %

4 Experiments and validations

A 1.6 liters 8v single point injection (SPI) test engine has been analyzed at full load to compare the power output and specific fuel consumption values of a conventional gasoline engine with a stratified charge engine having a twin swirl combustion chamber [5]. The results of the experiments are given in Figure 2, which shows the changes of power and specific fuel consumption values versus engine speed when conventional and twin swirl combustion chambers have been used. Table 2 compares the results of these experiments with the real cycle IC engine mathematical model, for full load. The theoretical calculations are very close to the experimental results (with an error < 2%). Therefore the real cycle IC mathematical engine model is validated. When the experimental results are analyzed, it is seen that a 17% reduction in specific fuel consumption is satisfied with the use of twin swirl stratified charge engine, and the effective power is increased by 7%. Moreover, with the aid of this model it is calculated that the CO_2 emission is reduced by 9%. Note that in the previous example the percentage of CO_2 emission reduction was calculated very close to the reduction percentage of the specific fuel consumption (Table 1). That is because the engine powers for these simulations were the same. However for the case of the experiment, the power output of the stratified engine (62 kW) is higher than the power output of the conventional gasoline engine (58 kW). Therefore the reduction percentage of the CO2 emission (9%) is below the reduction percentage of the specific fuel consumption (17%). According to the results of the theoretical calculations and the engine tests, with the use of stratified charge engines having twin swirl combustion chamber, engine performance can be increased and fuel consumption, therefore CO_2 emission, can be significantly decreased.

5 Conclusion

Two-stage combustion mechanism mixture is a very effective way to increase the fuel economy and decrease CO_2 emissions up to 19% in gasoline ignition engines by preserving its power.

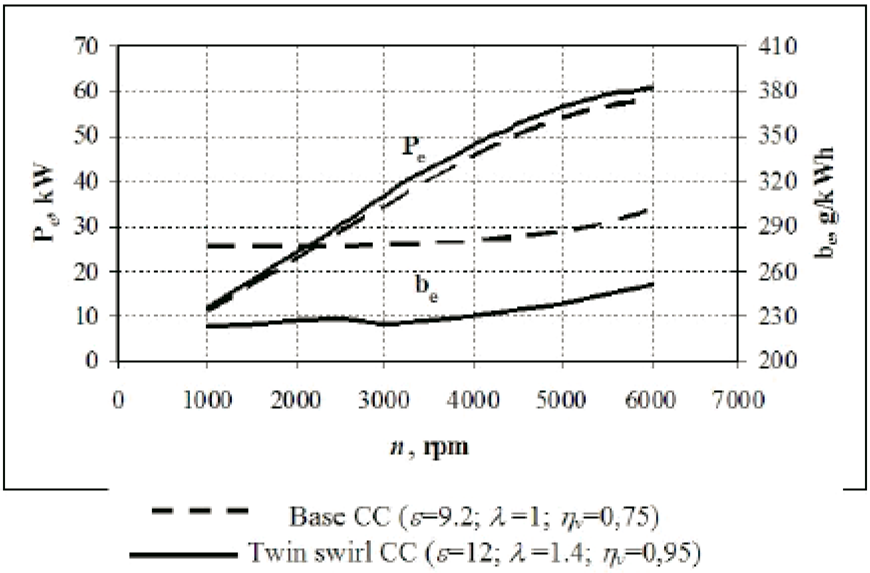

Figure 4.1. Power and specific fuel consumption changes versus engine speed of 1.6 8v SPI engine at full throttle loading.

Table 2 Power and specific fuel consumption changes versus engine speed of 1.6 8v SPI engine at full throttle loading

Full load n = 6000 min^{-1}	INPUTS			THEORETICAL			EXPERIMENTAL	
	η_v	ε	λ	P_e, kW	b_e, g/kWh	Relative CO_2 reduction	P_e, kW	b_e, g/kWh
Conventional	0.75	9.2	1.00	57	302	0 (reference)	58	300
Stratified charge	0.95	12.0	1.40	62	249	9 %	62	250

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