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Theoretical and Experimental Analysis of the Fuel Stream in GDI Engine

Abstract

Abstract: The paper ranges over demonstration of increase in total efficiency of a GDI engine in dependence on the parameters of mixture formation and range of load at determined states of engine work in dependence on the rotational speed of the engine.

By use of the newest simulation programme KIVA 3V possibilities of up-to-date methods of calculation of changes of temperatures, pressures, formation of toxic components during the process of stratified charge combustion were presented. The elaborate mathematical models, used in calculations, describing the processes occurring inside the cylinder of the engine permit to create complicated virtual models reflecting the real conditions in a satisfactory way correctness of calculation results is determined by the quality of input data which can be obtained basing on model investigations.

In the experimental part investigations on test bed with use of specialist apparatus for visualizations of inside cylinder processes were discussed. By use of visualization observation was performed of the run of the fuel jet from the moment of injection, consequent by rebounding of the fuel from the piston head, on to reaching under the gap between the electrodes of the ignition plug and flame spreading from the moment of ignition till the end of the combustion process.

For the determination of the total efficiency of a Gasoline Direct Injection engine, test bed investigations were carried out with the aim to determine the speed and load characteristics of the investigated engine. On this basis, the total efficiency of a GDI engine can be determined.

Keywords: GDI Engine, Stratified Charge, Visualization, Combustion process

1. Introduction

Constructors of gasoline engines are faced with increasingly higher requirements with regard to ecology, increasing engine efficiency, and decreasing fuel consumption at the same time. The satisfaction of these requirements is possible through recognizing the phenomena that occur inside the engine cylinder, choosing suitable optimal parameters of the fuel injection process, and determining the geometrical shapes of the combustion chamber and the piston head.

All these parameters significantly influence the performance of gasoline engines and improve their efficiency. An increase in efficiency is, first of all, connected with a change in fuel supply, i.e. a proper regulation of fuel-air mixture depending on the rotational speed and the load. Hence, the combustion of stratified mixtures in gasoline engines with direct fuel injection is essential for achieving an increase in their efficiency with a simultaneous decrease in the content of toxic components in exhaust gases, and a decrease in fuel consumption.

2. Modelling of the injection and combustion process by means of *Kiva 3v* program

2.1 Geometry of the calculation model

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Unlike KIVA 3V programme, developed for computer modelling and simulation of combustion engines, possesses a large, developed, graphic interface which may additionally consider the inflow and outflow systems and create complicated curved surfaces depicting, as in our case, the piston head.

Although the combustion chamber of a Mitsubishi GDI engine is a complicated system, the commercial KIVA 3V programme (created on the basis of KIVA II in the National Laboratory at Los Alamos) describes fully the physical and thermodynamical processes inside the cylinder. Fig.1 shows a geometrical model of a piston of the Mitsubishi 4G93GDI gasoline engine.

The cylinder was divided into 20 400 cells (30x34x20) and each of the four pipes into 1900 cells. The total number of cells of the entire system (when the piston is in BDC) amounted to 29680 cells. The mesh of one cylinder with two inlet and two exhaust pipes and the pent-roof combustion chamber is shown in Fig.2.
In order to determine the engine's thermodynamic parameters, a special division of cylinder layers was needed along the wall and a more complex grid inside the combustion chamber. The piston head was created by the CAD system and transformed to a pre-processor file. For this reason, a special algorithm of interpolation was written to adopt Lagrange coordinates. The cylinder mesh changes while the piston is moving and the valve movement also affects the creation of the mesh at every time step. The combustion chamber mesh at 27 deg ATDC, when the inlet valves are open, is shown in Fig.3.

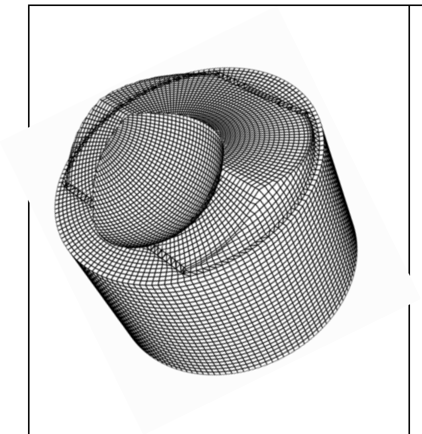

Fig.1 Geometrical model of a piston of a GDI engine

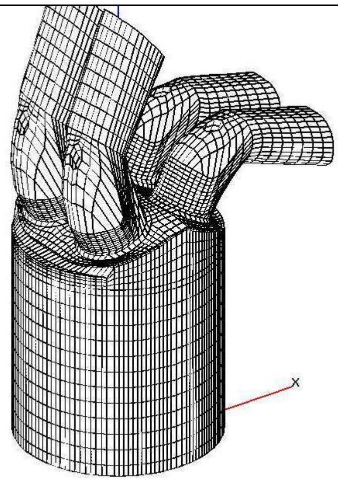

Fig.2 Complex mesh of a Mitsubishi GDI engine cylinder

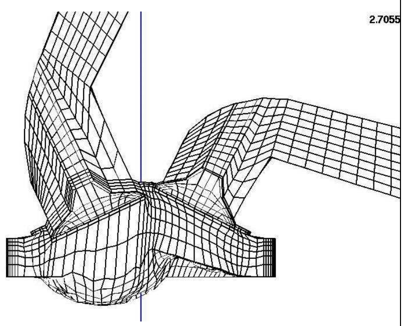

Fig.3 Engine mesh at 27 deg ATDC

2.2 Analysis of the gaseous phase participation
The following illustrations (Fig.4-6) show the gaseous phase participation changes inside the cylinder of a GDI engine from the moment of injection to the moment of ignition.

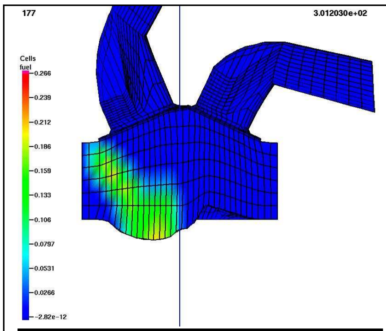

Fig.4 Participation of the gaseous phase in the combustion chamber in the stratified charge mode at 2400 rpm and 60^0 before TDC

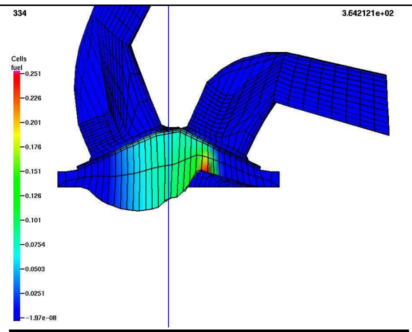

Fig.5 Participation of the gaseous phase in the combustion chamber in the stratified charge mode at 2400 rpm and4 ^{0} after TDC

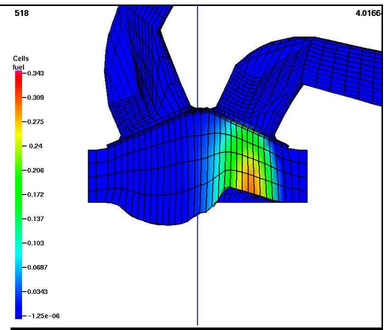

Fig.6 Participation of the gaseous phase in the combustion chamber in the stratified charge mode at 2400 rpm and40 ^{0} after TDC

The rotational speed of the engine was 2400 [rpm]. The amount of injected fuel corresponded to the global coefficient of air excess λ = 1,512. An equal temperature of the combustion chamber walls of approximately 500 ^{0}K, a lower temperature of the cylinder walls of approximately 480 ^{0}K, and the piston temperature of approximately 550 ^{0}K can be adopted.
A turning of the fuel jet to the spark plug is visible, but concentration of fuel on the piston bowl is also observed. Near TDC, some liquid fuel flows to the squish region and sometimes cannot be burned. When the jet is in motion, fuel vaporizes and there is more vapour at its boundary than inside the jet. Due to the limitations of this paper, the distribution of fuel-air equivalence ratio cannot be presented.

2.3 Analysis of temperature distribution in the cylinder

The following illustrations (Fig.7-9) show temperature changes inside the cylinder of a GDI engine from the moment of injection to the end of the combustion process.

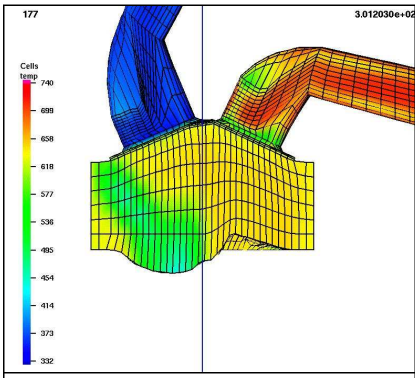

Fig.7 Temperature distribution in the cylinder for 59^0 before TDC

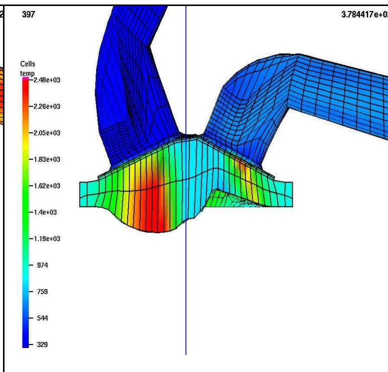

Fig.8 Temperature distribution in the cylinder for 11^0 after TDC

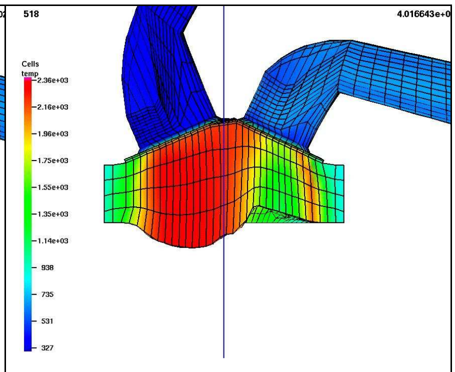

Fig.9 Temperature distribution in the cylinder for 41^0 after TDC

During the injection process, one can observe a decrease in the temperature of the charge where there is liquid fuel, which is caused by a vaporization process. When the piston is near TDC, the temperature of the charge in the squish region is higher than in the centre of the combustion chamber. The process of combustion during the stratified charge mode is irregular. As a result of fuel and gas conductivity, the regions with fuel vapour surrounding liquid fuel ignite first. This can also be observed during the visualization process. The distribution of temperature shows the entire process of combustion and proceeds in a different way than in conventional engines with a homogeneous charge. At the very end of this process the charge in the middle of the combustion chamber burns as a result of a higher temperature and vaporization of the fuel jet.

3. Visualization of the injection and combustion process in a GDI engine

3.1 Investigation object

The investigation object is a 1800-cm^3 Mitsubishi 4G93GDI engine, which is a 4-stroke, 4-cylinder, 16-valve engine with a double distribution shaft in the head, constructed on the basis of an engine with multi-point indirect fuel injection of the same volume.
For a visualization of this type of engine access of the measuring apparatus to the inside of the combustion chamber had to be made possible. This is why the engine head was subjected to some modifications: two openings of the same diameter $\varphi = 10[mm]$ were made at an angle measured from the bottom horizontal plane of the head. The angles between the two openings, measured in the cross plane of the head, were $\varphi = 5$ and $\varphi = 7^0$. The choice of the value of these angles is decisive for recording the processes occurring inside the cylinder since they determine the area of visual control of the recorded pictures. Two sleeves were fastened in these openings; one for an endoscope and another for a stroboscope lamp. After this adaptation of the engine head, visualization was performed on a chassis test bed.

3.2 Visualization of injection and combustion in engine working on stratified mixture

This visualization concerned the injection and combustion process when the engine worked on a stratified mixture. The recording was carried out at the engine's rotational speed of 2400 [rpm] with a partial load. The value of fuel consumption per unit was 238 [g/kWh]. Fuel injection took place at 78^0 before TDC.

The film frames presented below (Fig.10-12) are selected photographs showing fuel being injected into the cylinder of a GDI engine.

Fig. 13-15 show selected film frames from the visualization concerning the combustion way in the GDI engine when the engine worked on a stratified mixture. The ignition occurred at 10 deg CA before TDC.

		
Fig.10 Photograph of fuel jet injected at 55 deg CA before TDC. Due to turbulence, considerable evaporation of fuel has taken place, while a portion of the part that has not evaporated reaches the inclination of the piston head.	Fig.11 Photograph of fuel jet injected at 39 deg CA before TDC. The fuel jet is directed by the curvature in the piston head towards the ignition plug	Fig.12 Photograph of fuel jet injected at 35 deg CA before TDC. The fuel jet is at the exit from the piston head curvature. Further evaporation is taking place.
		
Fig.13 Photograph of further flame development at 20 deg CA after TDC. The flame spreads over the entire combustion chamber and the flame front moves towards the zone of expression.	Fig.14 Photograph of a consecutive phase of stratified mixture combustion at 30 deg CA after TDC. High whirling inside the cylinder allows the flame to penetrate the entire area of the engine cylinder.	Fig.16 Photograph of the after-burning phase at 37 deg CA after TDC. After-burning concerns the residue of lean mixture occurring on the edges of the combustion engine.

4. Test bed investigations of the 4g93gdi engine

4.1. Test bedA roller chassis test bed, equipped with an electrically controlled water brake, whose maximal moment was 180 [Nm], was adopted for the determination of speed and load characteristics of an 1834-cm^3 Mitsubishi 4G93GDI engine.

The system was equipped with meters of vehicle speed V [km/h] and power on wheels [kW].
The system for fuel consumption measurement was equipped with a Flowtronic apparatus, measuring fuel consumption per hour Ge [l/h], connected to the fuel pump located in the fuel tank.
The system for the measurement of the engine's rotational speed was equipped with a crankshaft angle encoder (Angle Encoder 364 made by AVL) on a pulley.

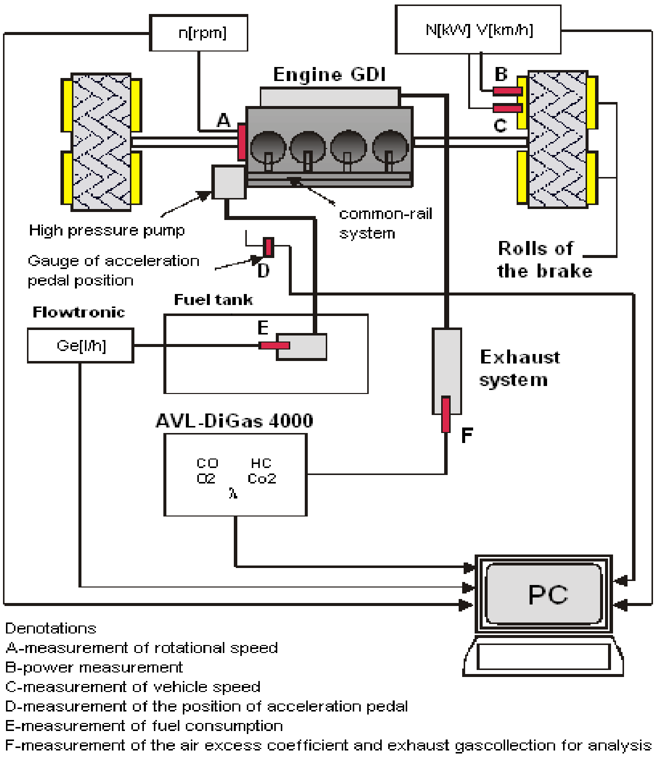

Fig.17 Scheme of the test bed

All the measurement systems were integrated with the central measurement computer mounted on the test bed to allow a precise determination of all possible data for a given rotational speed and load of the investigated engine.
The scheme of the measurement test bed used for the determination of speed and load characteristics of the investigated engine is provided in Fig.17.

4.2. Determination of the total efficiency of a GDI engine based on test bed investigation results

In order to determine the total efficiency of the 1834-cm^3 Mitsubishi 4G93GDI engine, use was made of the speed characteristics obtained during the test bed investigations of the relation of fuel consumption per unit in function of the engine's rotational speed.
With regard to a considerable decrease in fuel consumption per hour and per unit between the rotational speeds of 750 and 2700 [rpm], which was caused by the engine working in the mode of stratified fuel-air mixture ($\lambda \cong 1{,}5$-$2{,}1$) depending on the engine's rotational speed and load), the diagrams had to be complemented by additional characteristics of fuel consumption per unit. With this aim in mind, diagrams of fuel consumption per unit within the same range of rotational speed (750–2700 [rpm]) were drawn in the same way as for the engine working on a homogeneous mixture ($\lambda \cong 1$). In consequence, the value of fuel consumption per unit does not show the characteristic jump from one mode of work to the other.
Fig.18-21 show some traces of changes in the GDI engine's total efficiency in function of its rotational speed.

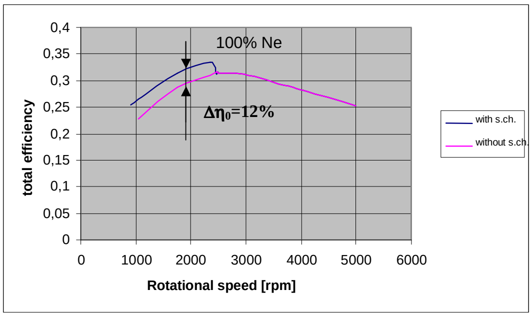

Fig.18 Relation of total efficiency η_0 in function of rotational speed at full power

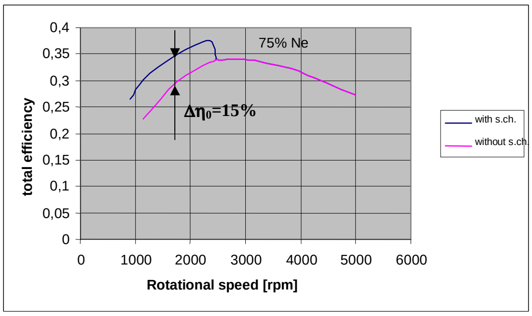

Fig.19 Relation of total efficiency η_0 in function of rotational speed at 3/4 of rated power

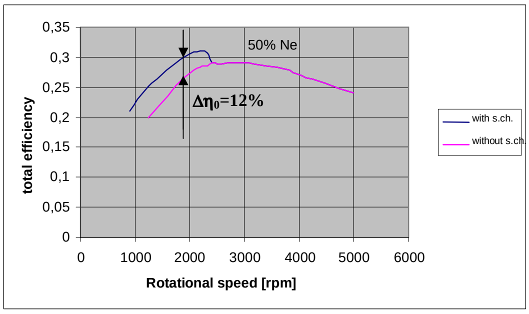

Fig.18 Relation of total efficiency η_0 in function of rotational speed at ½ of rated power

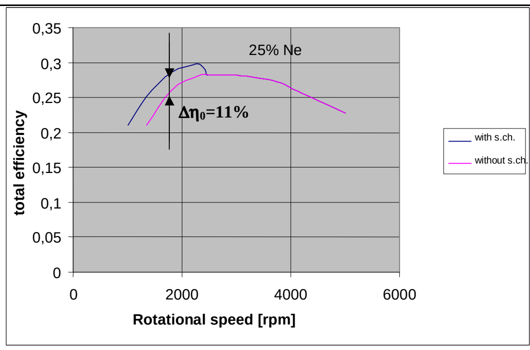

Fig.19 Relation of total efficiency η_0 in function of rotational speed at 1/4 of rated power

5. Conclusion

1. A considerable increase of approximately 17% has been noticed in the total efficiency of the investigated gasoline direct injection engine at the determined rotational speed when the engine worked on stratified mixtures (injection during the compression stroke). As a result, fuel consumption per unit and per hour decreases by approximately 17%.
2. The air excess coefficient value during work on a heterogeneous mixture increases to $\lambda \cong 2{,}2$.
3. Stratification of the charge depends, first of all, on the engine's rotational speed and load, and remains at this level to approximately 2700 [rpm].
4. A characteristic moment of transition from the engine working on a heterogeneous mixture to it working on a homogeneous mixture is noticeable in the form of a rapid jump of fuel consumption per unit by approximately 60 [g/kWh] on all characteristics of partial powers (i.e. for the full value, 3/4, 1/2 and 1/4 of the rated power).
5. The range of rotational speeds where the total efficiency increases is 600-2700 [rpm] depending on the engine's working conditions.

6. References

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