EXPERIMENTAL STUDY OF THE DIRECT METHANE INJECTION AND COMBUSTION IN SI ENGINE

Dariusz Klimkiewicz and Andrzej Teodorczyk

Warsaw University of Technology, ITC, Nowowiejska 25,00-665 Warsaw Poland
tel.:+48 22 6605277; fax: +48 22 8250565, e-mail: dklim@itc.pw.edu.pl, ateod@pw.edu.pl

Abstract

This paper reports on experimental study performed in a single cylinder SI research engine with the direct injection of methane to the cylinder. The methane injection was performed during compression stroke just before TDC – at the similar period as fuel injection occurs in Diesel engines. One of the main goals for direct injection of methane to the cylinder is the improvement of volumetric efficiency of the engine in comparison with standard method of gas mixture formation with the mixer. Direct methane injection creates many problems with proper mixing. The time available for mixing of the injected gas with the air is very short and the gaseous jet penetration in the combustion chamber volume and its ignition and mixing with the air is weak.

During the study several design parameters, such as, position of ignition, direction of methane jet, douse piloting, were tested on their influence on ignition and combustion process. The high pressure injector was used for methane injection with the pressure of injection equal to 10 MPa. As a result of the experimental investigations a number of pressure profiles and the corresponding series of the frame schlieren pictures of the injection and combustion processes were obtained.

Keywords: transport, SI combustion engines, combustion processes, methane injection

1. Introduction

Main cause of interest of research centers and vehicle industries are first of all increasing standards of emission level engines from, that they say [1][2]:

- The Kyoto Protocol have said, that by 2008, countries have to reduce their GHG (greenhouse gas) emissions to around 5.2% below their 1990 levels (for many countries such as the EU members that corresponds to some 15% below their expected GHG emissions in 2008)
- ACEA self commitment – reduction of emission CO_2, from 186 g/km level in 1995 year to 140 g/km level in 2008 year
- The purpose of EU have been reduced CO_2 emission to 90 g/km level in 2010 year (Fig.1).

The exhaust from engines are the highest emitter of pollution in city agglomerations. They are generating near 99% CO, 96% soot, 76% nitrous oxides (NO_x) and benzene, SO_2 and CO_2 [3]. So, they say that the CNG natural gas is most of all promising alternative fuels in future and they suggested the CNG is the fuel to join nowadays fuels (conventional-liquid fuels) with the future fuels (hydrogen fuels). If we use this fuel, we would reduce the pollution emissions nowadays and in the future. The main component of CNG is methane (near 98%). This fact causes the combustion of natural gas to benefit:

- Reduced particulate and NO_x emissions,
- Reduced greenhouse gas emissions,
- Widespread availability of natural gas,
- Lower cost,
- Can be derived from renewable sources (biogas),
- Technically proven,
- Available now,

- Can be used in all vehicles classes,
- Minimal processing or refining requirements,
- Safer than most liquid fuels,
- Can be refueled at home or workplace,
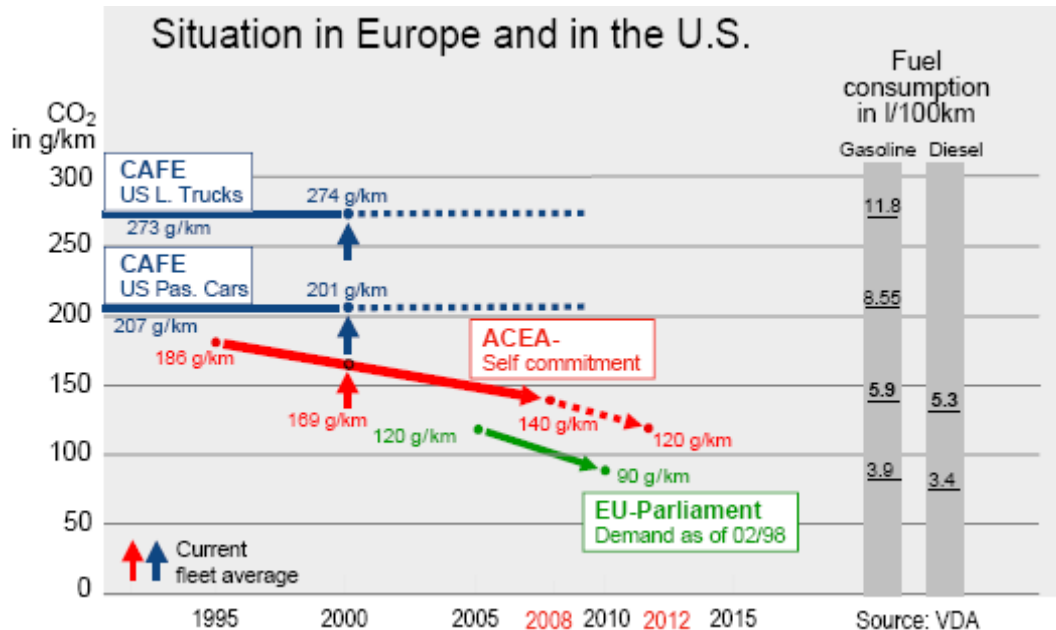- Noise reductions of as much as 50%,
- Reduced engine wear,
- and more…

Figure 1. Local CO_2 emissions [5]

In this days, the CNG engines are under the development. There are more than 5.2 million natural gas vehicles (NGVs) in operation around the world today [5]; nearly 30% in Argentina and 7,5% in Italy alone (Table 1).

In majority they are the engines with mixer supply system, but similarly how in liquid fuels an engine, the development of those engines follows the supply system with direct CNG injection to cylinder. The centre which promotes effectively this kind system to the power supply of engines with CNG is the Westport Innovations Inc. in Canada. The direct injection of CNG is mainly steered on growth of efficiency of engine, the lowering the emission of toxic components of fumes, improvement of coefficient ratio in the cylinder and reduce engine wear. The main problems in this kind of supply system are: the obtainment recurrent and effective ignition mixture (in every cycle of work in engine) the produced in cylinder and the obtainment the more suitably mass speed of burning. It was can reach across selection of suitable shape of injected jet, choice of suitable way of ignition mixture and location the point of ignition so, to that was in the place, in which in moment of ignition the composition of mixture is close stoichiometric mixture. Because the methane which is main component of CNG characterizes with high temperature of spontaneous ignition it's necessary to use of external system of ignition. Ignition of mixture air-gas, got in result of direct injection CNG we could use three ways of ignition: with piloting dose of diesel fuel, with the heat candle [3] and from sparking plug [4]. All results of investigations of direct injection methane in article were introduced to produce mixture of burning in cylinder SI engine with ignition from sparking plug.

Table 1. Numbers of CNG vehicles on the world

Country	Vehicles*	Refuelling Stations	VRA**	Last Update
Argentina	1,459,236	1,400	32	XII 2005
Brazil	1,117,885	1227	-	IV 2006
Pakistan	1,000,000	930	-	V 2006
Italy	382,000	509	-	V 2005
India	248,000	198	-	III 2006
USA	130,000	1,340	3,331	I 2006
.	.	.	.	.
Spain	797	28	21	VI 2005
Polska	*771*	*28*	*18*	*IV 2005*
Unitek Kingom	543	31	115	XI 2004
.	.	.	.	.
Taiwan	4	1	-	VI 2005
North Korea	4	1	-	VIII 2005
Bosnia i Herzegovina	1	-	1	IV 2005
TOTAL	**5,145,449**	**9,114**	**9,158**	

* Includes both OEM and converted NGVs
** VRA = Number of Vehicle Refuelling Appliances
Update: 22.06.2006

2. Experimental setup

The objective of this paper is to present the results of the investigations of the combustion system with direct methane injection. The investigations were performed with the use of the one-piston SI engine, described elsewhere, which allows to visualize the in-cylinder phenomena. The specification of the engine is given in Table 2.

Table 2. Specification of the engine

Number of cylinders	1
Bore [mm]	79,55
Stroke [mm]	77,00
Capacity [cm^3]	431
Speed [rpm]	750
Compression Ratio	$\varepsilon = 7,07$
Inlet Valve Pens [BTDC]	5°
Inlet Valve Clone [ATDC]	44°
Fuel	CH_4
Injection pressure	~ 10MPa

The original head of engine geometry is shown in figure 2. During the study several design parameters, such as, position of ignition, direction of methane jet, dose piloting, were tested on their influence on ignition and combustion process.
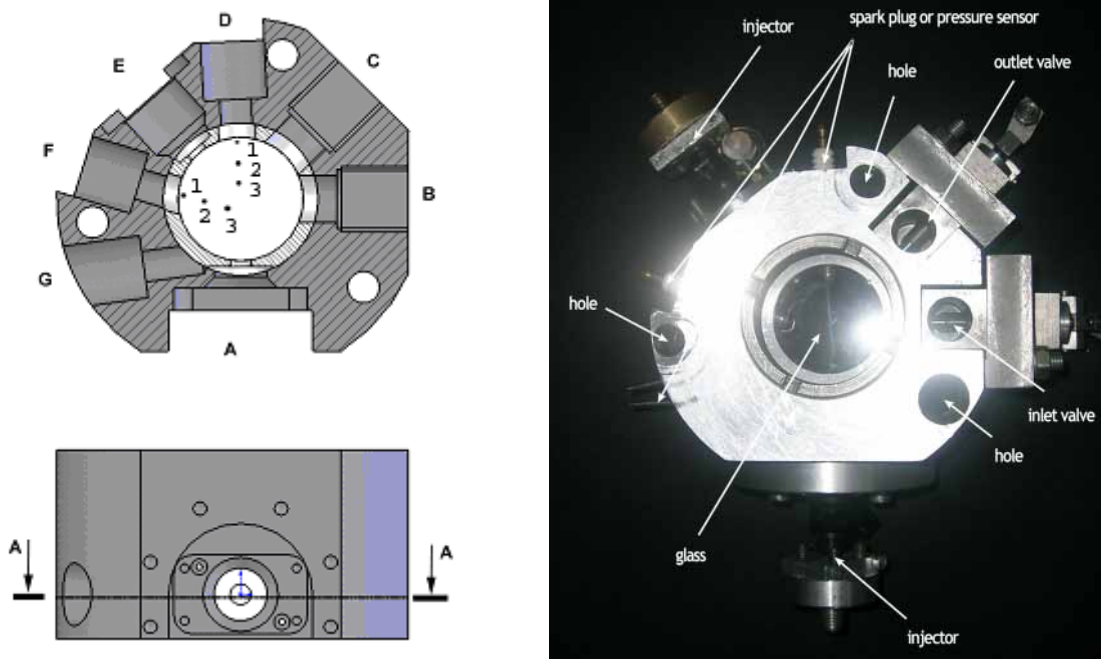
Figure 2. The head of one-piston SI engine
(**A, E**-injectors; **B**-inlet valve; **C**-outlet valve; **D**,**F**,**G** – sparking plug or pressure sensor, **1,2,3** – ignition area)

The compression ratio was 7.1 and the injector was used for methane injection with the pressure of injection equal to 10 MPa. The reactions of the system investigated on the changes of following parameters were: start of the injection – (20-180 deg BTDC); injection duration – (16-100 CA deg); ignition timing – (2-40 deg BTDC). As a result of the experimental investigations a number of pressure profiles and the corresponding series of the frame schlieren pictures of the injection and combustion processes were obtained. The schematic diagram of the experimental apparatus is shown in figure 3.
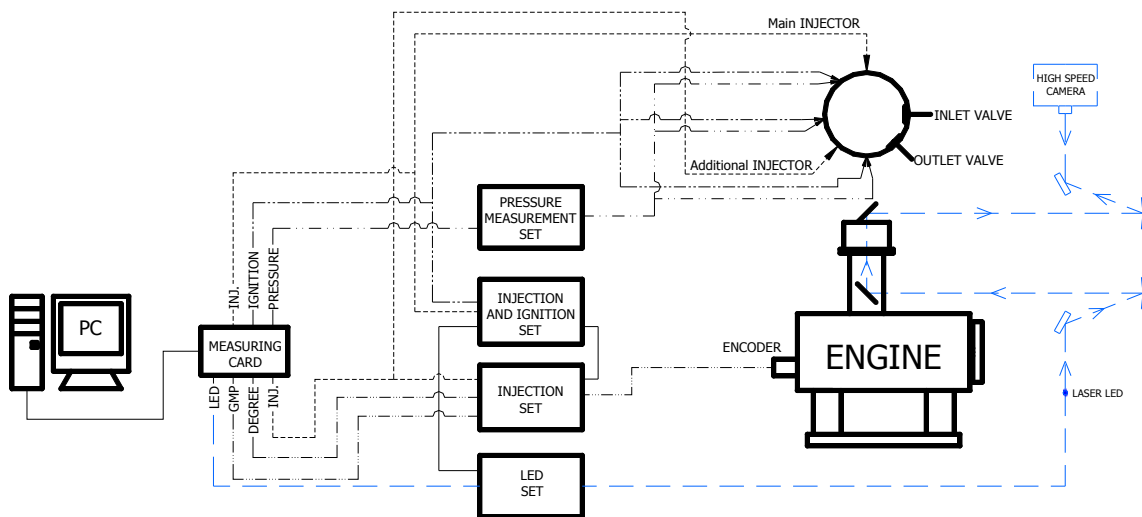

Figure 3. Schematic diagram of experimental apparatuses

3. Results and discussions

3.1. Time of ignition

First series of experiments were made for different time of ignition. Studies involved following cases: different ignition advance (16, 14, 12, 10, 8, 6 and 4 degrees before TDC), injection beginning: 60 degrees before TDC, equivalence ratio: 0.35, area ignition 1D and place of fix injector A (Fig.2). Results are shown in Table 3 and Figures 4.

Table 3. Research results for different ignition advance (equivalence ratio 0.35)

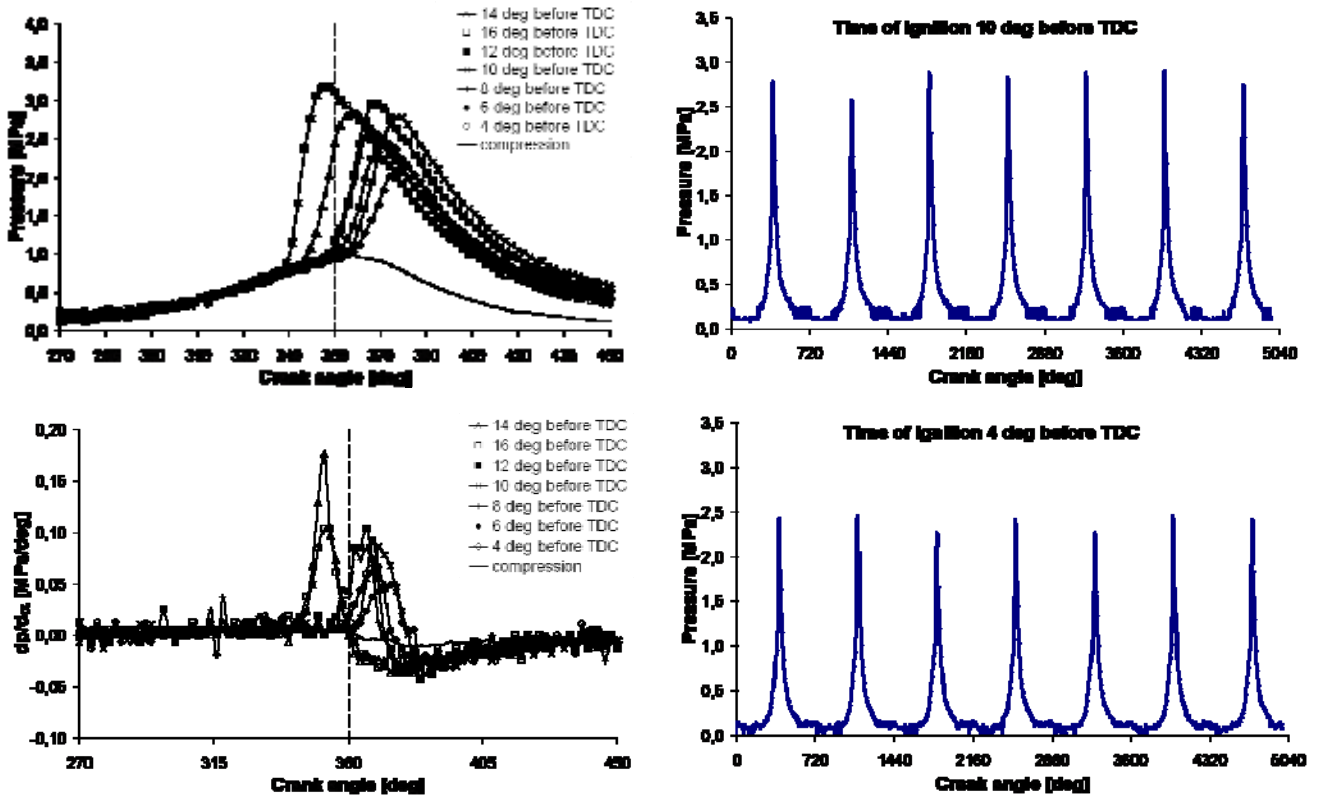| Ignition advance	Maximum of pressure	Degree of maximum of pressure performance	Combustion duration	$(dp/d\alpha)_{max}$
[deg before TDC]	[MPa]	[deg]	[deg]	[MPa/deg]
16	3,2	357	11	0,10
14	2,8	366	22	0,18
12	3,0	373	25	0,10
10	2,8	381	31	0,08
8	2,6	373	21	0,08
6	2,1	384	30	0,05
4	2,4	376	20	0,09

Figure 4. Combustion history, maximal cylinder pressure rise ratio and opened indicator diagrams for methane injection with different ignition advance

It was affirmed, that the delayed moment of ignition causes the shifting of maximum pressure after TDC and its lowering. The moment of occurrence of cylinder maximum pressure is the most profitable for ignition advance 12 and 14 degrees before TDC. In these cases the tendency of increase of maximum pressure in cylinder is additionally observed.

3.2. Time of injection

Next series of experiments was made for different time of injection. Tests involved the following cases: time of injection: 320, 280, 240, 200, 160, 120, 80, 60 and 40 degrees before TDC, ignition advance: 16 degrees before TDC, equivalence ratio: 0.55, double place ignition 1D and 1F and place of fix injector A (Fig.2). Results are shown in Table 4 and Figure 5.

Table 4. Research results for different time of injection

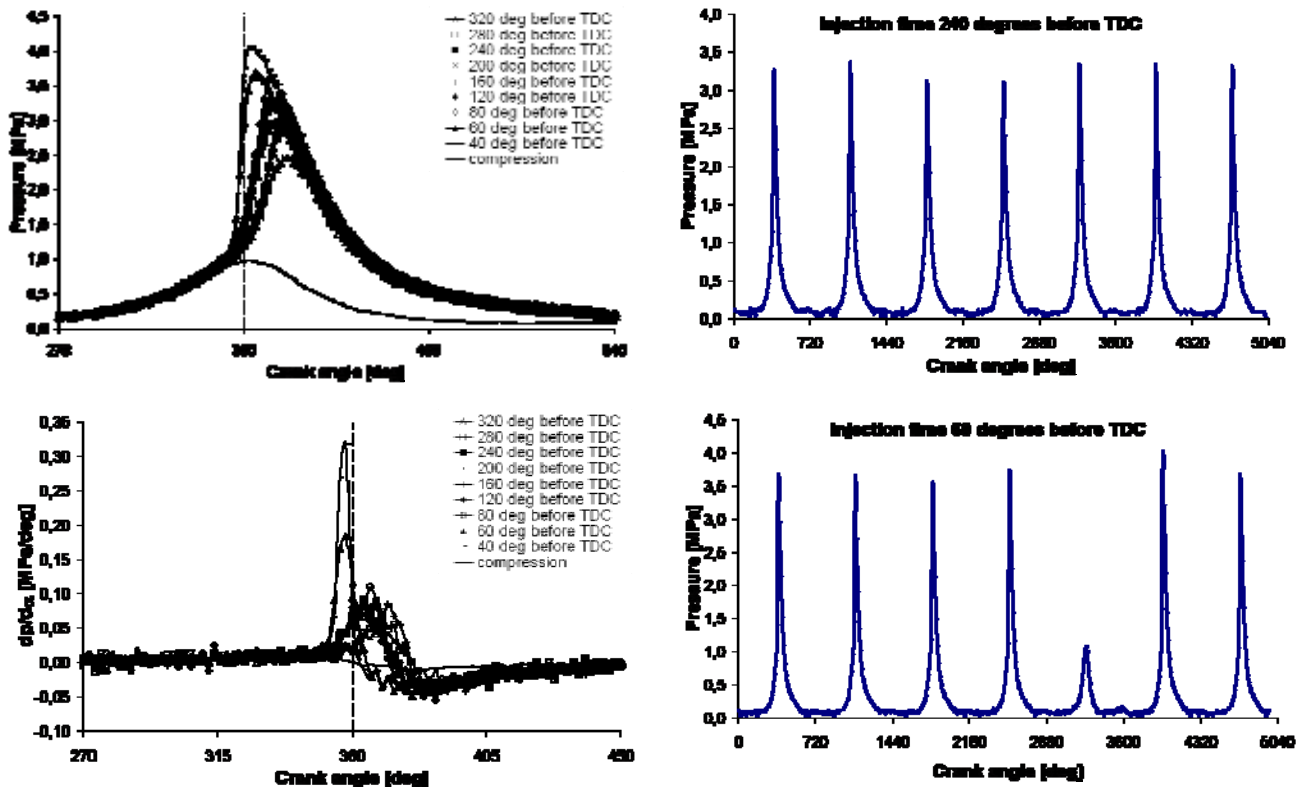| Time of injection	Maximum of pressure	Degree of maximum of pressure performance	Combustion duration	$(dp/d\alpha)_{max}$
[deg before TDC]	[MPa]	[deg]	[deg]	[MPa/deg]
320	3,0	376	32	0,08
280	2,9	381	37	0,08
240	3,3	375	31	0,10
200	2,4	381	37	0,08
160	3,3	380	36	0,10
120	3,3	377	33	0,08
80	3,6	374	30	0,11
60	3,7	366	22	0,20
40	4,1	364	20	0,32

Figure 5. Combustion history, maximal cylinder pressure rise ratio and opened indicator diagrams for methane injection with different time of injection

For the late time of injection the improvement of parameters of combustion in the cylinder was observed. The latest times of methane injection i.e. 40 and 60 degrees before TDC were the best. We can observe that sometimes the air-methane mixture does not ignite. This is unfavourable phenomenon.

3.3. Equivalence ratio

All results presented in this chapter were obtained for the second injector (position E on the view of head – Fig. 2). Research involved following cases: equivalence ratios: 0.6, 0.7, 0.8, 1.05, 1.4, ignition advance: 10 degrees before TDC, injection beginning: 60 degrees before TDC, double place ignition 1D and 1F (Fig.2). Results are shown in Table 5 and Figures 6.

Table 5. Research results for different equivalence ratios

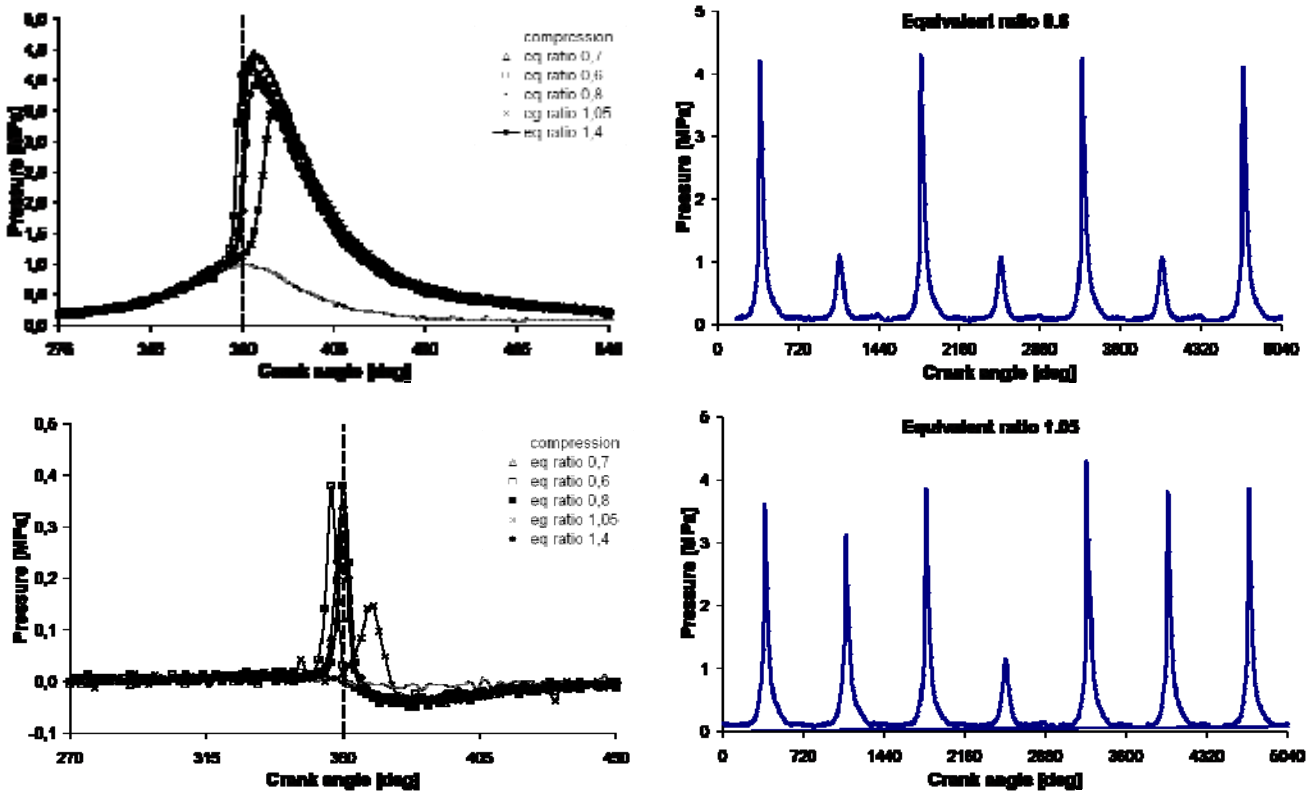| Equivalence ratio	Maximum of pressure	Degree of maximum of pressure performance	Combustion duration	$(dp/d\alpha)_{max}$
[-]	[MPa]	[deg]	[deg]	[MPa/deg]
0,6	4,2	362	16	0,38
0,7	4,4	366	12	0,34
0,8	4,5	366	16	0,38
1,05	3,6	376	26	0,15
1,4	3,9	368	18	0,28

Figure 6. Combustion history, maximal cylinder pressure rise ratio and opened indicator diagrams for methane injection with different equivalence ratios

The tests have confirmed that burning of lean mixtures is possible, but often they can not be ignited. The very good parameters of burning were got for stoichiometric mixtures. It was confirmed simultaneously, that rich mixtures have worse parameters of burning than lean mixtures.

3.4. Visualization of methane injection and combustion

Figure 7 presents the injection and combustion process in the combustion chamber of engine. First frame (2,0 ms after time of injection) shows the injected stream of the fuel that is

passing across the chamber and enriching the area of ignition. The next frames present methane injection and mixing of methane jet with air in the combustion chamber (5,2 and 6,8 ms). The ignition takes place at 20 degrees BTDC and this moment is presented on forth frame (10 ms). The next frames (from 11,2 to 13,6 ms) were made just before TDC and show flame propagation process. The combustion has been already transferred in the whole combustion chamber.
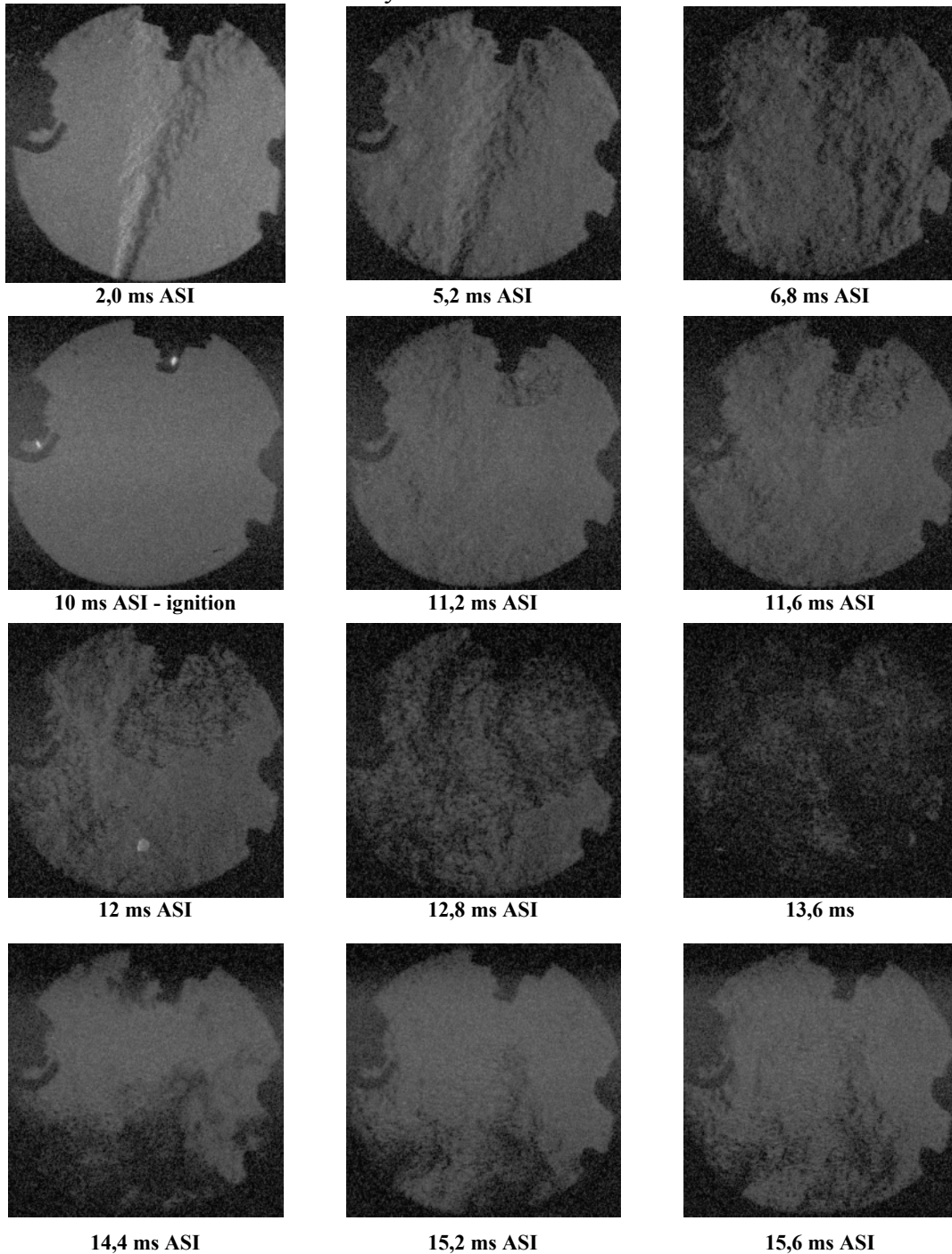

2,0 ms ASI	5,2 ms ASI	6,8 ms ASI
10 ms ASI - ignition	11,2 ms ASI	11,6 ms ASI
12 ms ASI	12,8 ms ASI	13,6 ms
14,4 ms ASI	15,2 ms ASI	15,6 ms ASI

Figure 7. Visualization of methane injection and combustion (ASI - after start of injection)

4. Summary

On base of investigations the following general conclusions can be draw:

- In direct methane injection engines there is a possibility of combustion of stoichiometric mixtures and lean mixtures in broad range of composition.
- Very good parameters can be obtained during the late injection to cylinder.
- Natural gas direct injection combustion can obtain the same high combustion parameters as that of homogeneous mixture combustion. We can make a high stratification mixture in cylinder and the combustion can be possible.

Acknowledgement

References

[1] A.Thorsten, C.Bischoff and J.Foerester, *Natural Gas As An Alternative Fuel For Motor Vehicles,* FISITA, World Automotive Congress, 23-27.05.2004
[2] Web site: http://www.ifp.fr – Institut Francais Du Petrole
[3] Web site: http://iangv.org - International Association of Natural Gas Vehicles
[4] Huang, Z., Zeng, K. and Yang, Z., *Study on combustion characteristics of direct injection natural gas engine by a rapid compression machine*. Trans. CSICE, 2001, 19(4), 314-321
[5] Web site: http://www.unece.org - United Nations Economic Commission for Europe