SI Engine with Direct Injection of Methanol Reforming Products – First Experimental Results

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ABSTRACT

In this paper we describe conversion of the gen-set gasoline-fed carburetor single-cylinder SI engine to the direct-injection version operating with the gaseous hydrogen-rich methanol reforming products, and present the first experimental results.

It was found that engine feeding by methanol steam reforming products has a great potential of pollutant emissions mitigation as compared with gasoline. NO_x concentrations in the exhaust gas were reduced by a factor of 7 as a result of the lean combustion and lowering in-cylinder temperatures. Particle mass emissions were mitigated to zero-impact levels. Harmful emissions of the target pollutants THC, CO and the GHG gas CO_2 were reduced by a factor of 6, 25 and 1.5, respectively. The achieved efficiency improvement of the engine fed by the directly injected methanol reformate calculated for primary liquid methanol consumption was found to be in the range of 20-70% (depending on the operating mode) as compared with the carburetor-fed gasoline. Higher values of the improvement were observed at lower engine loads.

The in-cylinder maximal pressure and the value of pressure-rise rate were higher under the reformate feeding by 75% and 37%, respectively at the same ignition timing.

INTRODUCTION

It is known that about one-third of fuel energy introduced to an ICE is wasted with engine exhaust gases. Even its partial utilization can lead to a significant improvement of the ICE energy efficiency. One of the ways to recover an engine's waste heat is by using exhaust gases energy to promote fuel endothermic reactions that produce hydrogen-rich reformate. In principle, any fuel may be used, but alcohol reforming is scrutinized closely since it can be reformed at much lower temperature than conventional fuels. In this case ICE can be fed by the gaseous fuel with high content of hydrogen without the known problems of onboard hydrogen storage. This approach, called thermo-chemical recuperation (TCR), is considered nowadays as one of the promising methods of increasing powertrain efficiency and reducing emissions to zero-impact levels.

In this paper, we consider TCR by methanol steam reforming:

$$CH_3OH + H_2O \rightarrow CO_2 + 3H_2 \quad \Delta H = 50\,kJ/mol \qquad (1)$$

In this process endothermic chemical reactions of water and methanol produce hydrogen-rich reformate that may be more efficiently burned than a liquid fuel, allows lean combustion and thus reduces throttling losses. Moreover, the reformate LHV is by 14% higher than that of methanol. High hydrogen content, typical for alcohol-reforming gaseous products, leads also to increase of the flame velocity as compared to gasoline and enables higher compression ratio. However, a high hydrogen-content reformate supplied to the engine through the intake manifold also leads to the known problems of backfire, pre-ignition and reduced maximal power [1]. Authors of [2, 3] avoided these problems by using liquid unreformed fuel at high loads and partial fuel reforming at low and medium loads. It is known that pre-ignition is usually initiated by the hot surfaces of exhaust valves or a piston, residual flame in the piston crevices, oil contaminants and residual energy in the spark plug [4]. White et al. [5] reviewed various ways that could allow mitigation of these issues. However, even if the mentioned above problems caused by supply of the hydrogen-rich gaseous fuel through the engine's intake manifold would be resolved, its maximal power output still will be substantially reduced comparing with a same-size gasoline engine. This is a result of the volumetric efficiency decrease caused by high partial volume of a gaseous fuel in the cylinders intake charge [6].

Direct injection (DI) of a gaseous fuel into ICE's cylinders can solve the described above problems. Wimmer et al. [7] had shown that for a hydrogen-fueled engine direct injection of the gaseous fuel is the most promising method of achieving high BTE, power density comparable to gasoline engines and resolving the pre-ignition and backfire problems. The authors of [8, 9] studied mixture formation under injection pressures of 25-100 bar and reported on a high efficiency and low NO_x emissions of a turbocharged hydrogen-fuelled engine with injection pressure of 100 bar. In various compressed natural gas (CNG) and hydrogen DI-ICE studies [10-13], the injection pressures varied from 20 to 120 bar. There are several additional benefits of using high-pressure direct fuel injection. The first one is the injection time shortening that allows late injection start during the compression stroke and the mixture stratification. The second reason is a possibility of limiting the compression work increase caused by rise of the partial volume of gaseous fuel in the air-fuel mixture compared to the liquid-fuel counterpart. Retarded fuel injection may reduce this negative influence, but requires high injection pressure to overcome the pressure build-up in the cylinder. Other benefit of a high-pressure direct injection is increased fuel penetration into the cylinder. In the mentioned above studies, the hydrogen or CNG were stored onboard in high-pressure vessels that were pressurized outside the vehicle. Hence, the energy required for compressing the gas was evidently not considered in the overall ICE efficiency analysis. When an ICE with TCR is analyzed, the latter factor must be taken into account. As we demonstrated in [14], the DI-ICE with TCR is unviable, if reforming is carried out at atmospheric pressure. Based on this finding, we suggested a novel approach of a direct injection ICE with TCR and a high-pressure fuel reforming that can resolve the mentioned above problems [14, 15]. This would allow preventing the backfire, pre-ignition and reduced volumetric efficiency problems without a need of limiting hydrogen content in the reformate gas or injecting a liquid non-reformed fuel at high engine loads.

The main goals of the reported study were development of a direct injection SI engine (based on a conventional gen-set gasoline SI carburetor engine) fed by the gaseous methanol-reformate fuel and analysis of the achieved engine performance improvement.

METHODOLOGY

THE ENGINE

ROBIN EY20-3 air-cooled naturally aspirated, single-cylinder spark-ignition 4-stroke engine was selected for the laboratory experiments. It was coupled, as a gen-set, with the SINCRO GP100 2.2 kW AC 230V generator (Italy). The engine has side valves location (Fig.1). This configuration allowed more space in the cylinder head for installing an in-cylinder pressure transducer and a direct fuel injector. The engine main parameters are listed in Table 1.

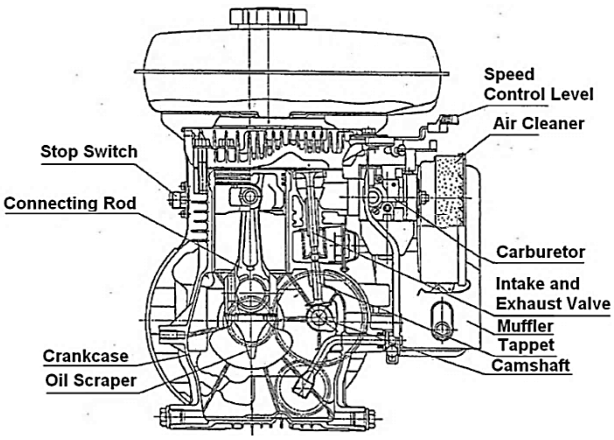

Fig.1. Cross section of the ROBIN EY20-3 engine.

As can be seen from Table 1, the engine has very low original compression ratio. It is well-known that high knock resistance of a hydrogen-rich gaseous alcohol reformates allows SI engine operating under much higher values of compression ratio. Carrying-out experiments at higher compression ratios is planned at the next stages of the research.

Table 1. Specification of the ROBIN EY20-3 engine

Bore x Stroke, mm	67x52
Displacement, cm3	183
Compression ratio	6.3
Power, kW @ speed, rpm	2.2 @ 3000
Lubrication	Splash type
Carburetor	Horizontal draft, Float type
Gasoline feed system	Gravity type
Ignition system	Flywheel magneto
Spark plug	NGK B6HS
Starting system	Recoil starter
Governor system	Centrifugal flyweight type

EXPERIMENTAL SETUP

Engine control and feeding

The study objectives required development, manufacturing and operating the laboratory experimental setup that allows control and measurement of the engine performance in the case of ICE feeding by gasoline as well as in the case of direct injection of an alcohol reformate (gaseous fuel). A conceptual scheme of the engine control system is shown in Fig.2. As can be seen, the ICE was equipped with sensors and actuators that formed the engine's closed-loop control system.

The programmable ECU 18 processes signals from the sensors and is coupled with a computer 19. The crankshaft angle position was measured instantaneously by the encoder 23. This information was used for determining engine's indicatory performance parameters and for the fuel injection and ignition timing control. The in-cylinder pressure indicating system included: the pressure transducer 5, the amplifier 21, the encoder 23, the high-speed data-logger (not shown in the scheme) and the computer 19. The laboratory experimental setup was equipped with a rheostat unit 14 that allowed variation of the engine load by changing the electric resistance on the generator 13 output. The desired engine speed was set by varying the spring tension of the governor 11 with the actuator 12. This leads to changing the throttle opening angle and the corresponding change of the intake air flow-rate. The latter was measured by the flow meter 17. Its signal was directed to the control unit 18 that controlled duration of the fuel injection event, thus tuning the fuel quantity according to a desired value of the air excess factor λ. The latter was controlled by the wide-band oxygen sensor 25 and the air-fuel ratio gauge 27. This λ value was double-checked by measuring air and fuel flow rates.

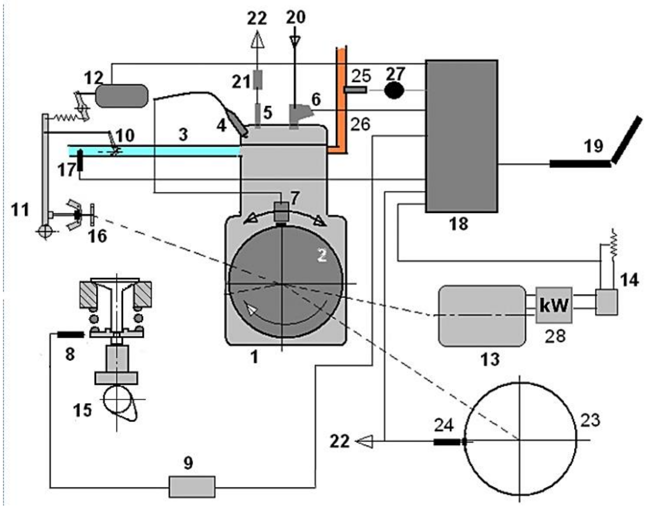

Fig.2. Conceptual scheme of the engine control system.

At this stage of the work the engine was fed by a syngas imitating methanol reformate products, which was stored in a high-pressure cylinder. The syngas was provided by a supplier of calibration gas mixtures. A conceptual scheme of the developed system of the engine feeding with a high-pressure gaseous reformate is presented in Fig 3.

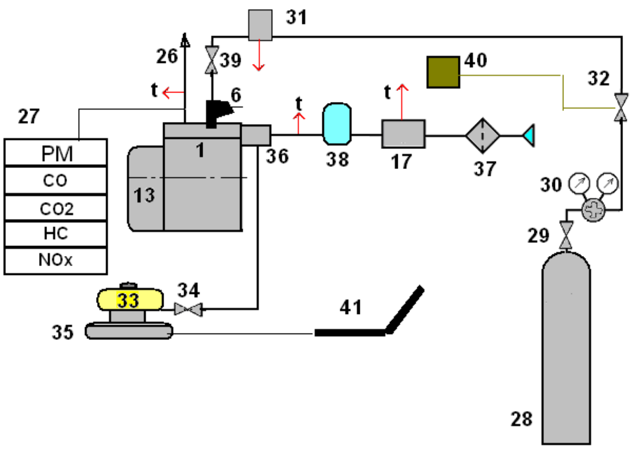

Fig.3. Conceptual scheme of the engine fuel supply system.

This Figure also shows schematically the intake air supplying line, the gasoline consumption measuring and the exhaust gas analyzing systems. Pressure of the reformate at the inlet of the gaseous fuel injector 6 was fixed at a desired constant level using the pressure reducer 30.

Information about main control and measuring devices used in the developed experimental setup is concentrated in Table 2. The provided accuracy values of the measuring devices are based on the manufacturers' data.

Table 2. Main control and measuring devices used in the experimental setup (numbers correspond to those in Figs. 2 and 3).

Number	Device	Manufacturer, Country of origin, (Accuracy)
23	Crankshaft encoder 2613B	Kistler Instrumente A.G., Switzerland, (Dynamic accuracy +0.02^o at 10000 rpm)
-	Charge Amplifier Type 5064B21	Kistler Instrumente A.G., Switzerland, (<±0.5% at 0-60^{o}C)
5	Water cooled pressure transducer 6061B	Kistler Instrumente A.G., Switzerland, (Max. linearity ≤±0.29% FS*)
18	Controller unit DS 1104	dSPACE GmbH, Germany
-	Indicatory diagram data-logger USB 2610	National Instruments Corporation, U.S.A.
8	Hall-effect proximity switch LCZ series	Honeywell, U.S.A.
12	Linear actuator FA 35	Firgelli Automations, U.S.A
31	Mass flow measurement RS323	Brokhorst High-Tech B.V., Netherlands, (±1% of FS*)
17	Air flow sensor VA420with integrated measuring unit	CS Intruments GmbH, Germany, (±1.5% of MV*)
22	Wide-band lambda sensor LC-2	BOSCH, Germany, (MV* of λ ±0.05)
27	Thermal Converter 501x	Teledyne Instruments, U.S.A.
27	NOx analyzer 200EH	Teledyne Instruments, U.S.A., (0.5% of MV*)
27	HC analyzer 600 series	California Analytical Instr., U.S.A., (±0.5% of FS)
27	CO, CO_2 analyzer 600 series	California Analytical Instr., U.S.A., (1% of FS)
27	Particulate mass monitor TEOM 1105 series	Ruppreht & Patashnik Co., Inc., U.S.A., (±0.75%)
28	Power gauge (watt-meter) DW-6060	Lutron Electronics Company, U.S.A., (±1%)
35	Digital scales GF-12K	A&D Ltd, Japan, (±0.1g)

* FS – full scale; MV – measured value

Engine instrumentation

The Kistler encoder was mounted on the free end of the generator shaft using a specially designed flange. The gaseous fuel injector and the Kistler in-cylinder pressure transducer were mounted on the cylinder head top surface. The original ignition system was modified to allow controllable change of the ignition advance angle. A proximity sensor 8 was installed on the exhaust valve train 15. It generated a trigger-signal for starting the actuator 9 of the fuel injector 6 during the compression stroke (with the exhaust valve closed). This allowed prevention of fuel injection during the exhaust stroke (when the exhaust valve is opened). The in-cylinder pressure measuring line (pressure transducer – amplifier – data acquisitor - computer) was calibrated with aid of the specially designed and manufactured p_{max}-meter. For the calibration purpose the latter was mounted instead of the fuel injector. The calibration was carried-out when the engine was fed by gasoline. Following the manufacturer's requirements, the in-cylinder pressure transducer (model Kistler 6061B) was cooled with the distilled water at the flowrate of no less than 1 l/min. For this purpose an appropriate cooling system was developed, designed and fabricated. It ensured flowrate of distilled water up to 1.5 l/min, continuous control of the coolant pressure and water flow presence.

Gaseous fuel direct injector

At the time of planning the study, there were no market-available injectors for direct injection of gaseous fuels. Due to this fact, a direct gas injector suited for operation with alcohol reformate had to be developed. This task was accomplished by reworking a market-available injector originally intended for direct injection of liquid fuel. Examples of the similar efforts are described in [16, 17 and 18]. One of the widespread options is the so-called Spark Plug Fuel Injector – SPFI [16]. This approach combines the fuel injector with a spark plug and does not require additional space for the injector installation in a cylinder head. The main advantage of the SPFI injector is a low cost of engine's conversion, because of preventing a need in engine head modifications. The main drawbacks of this method are:
- The long distance that the fuel should travel prior to being injected into the cylinder, thus worsening fuel supply control at high-speed and transient operating regimes;
- Limitations of the fuel metering orifice diameter because of the spark plug standard dimensions;
- Decrease of the engine compression ratio due to adding an internal volume of the SPFI injector to the compression chamber volume. This drawback is much aggravated for engines of small displacement.

Taking into account the mentioned above considerations, attempts were made to modify market-available high-pressure direct gasoline injectors for the purposes of our study. The main requirements that had to be achieved were:
- Compatibility of the modified injector to operating with a gaseous fuel from the point of view of the injector-elements' wear and overheating;
- Negligible internal volume added to the combustion chamber;
- Ability of operating at a pressure and a frequency of no less than 100 bar and 50 Hz;
- Opening/closing time – lower than 0.1 msec;
- Maximal gaseous fuel supply – at least 30 cm^3/cycle;
- Compactness that allowed the injector's mounting in the engine head;
- Low cost and possibility of fabrication on universal machine tools.

The Magneti Marelli gasoline direct injector IHP072 (Fig.4, a) was chosen as a basis for modification as it satisfied all the listed above requirements excluding the flowrate value. The reworked injector shown in Fig.4, b was used as a basis for developing a gaseous fuel direct injector used in this study. As can be seen, the nozzle part restricting the fuel flow was removed from the original injector.

The main guideline in the injector's development was a maximal use of the electronic and sealing elements of the mass-production, proven-reliability IHP072 injector and modification of the nozzle part to allow the required gas flowrate values. Various versions of the injector design, including the nozzle settle form and housing materials were investigated. The final design used in this study that ensured the flowrate stability and negligible leakage is shown in Fig. 4,c.

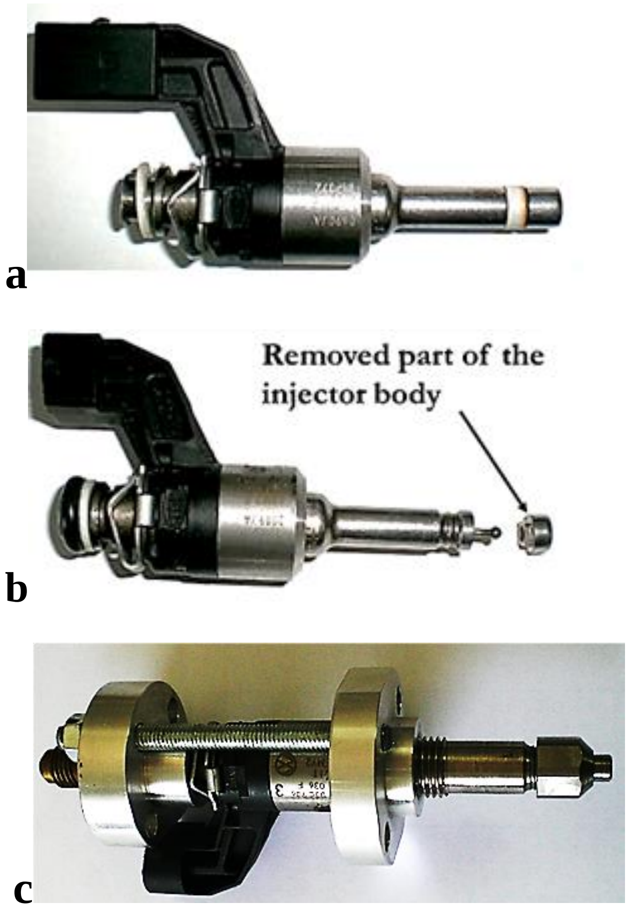

Fig.4. Gaseous fuel direct injector: a - original GDI injector; b – the injector preparation; c – final design, external view.

A laboratory electronic drive (ensuring the same command signal form as in a gasoline engine injector) was developed, designed and manufactured for the purpose of this study. The drive allowed changing injection frequency and duty cycle values. Prior to being mounted on the engine, the developed injector was comprehensively tested at the flow test bench.

The tests were fulfilled using compressed air as a working fluid, at the frequency 25 Hz and various values of air intake pressure, duty-cycle and the injector's needle-lift. The pressure value at the injector outlet was 1 bar abs. An example of the test results for the virgin injector, needle lift 0.15 mm and different values of the duty-cycle is shown in Fig.5. The maximal expected value of the gaseous fuel consumption by the engine is designated by a horizontal line. The leakage values of the injector showed in this Figure were found without the injector driving (the injector was constantly closed). As can be seen, under 40 bar the leakage does not exceed 7% relative to maximal expected value of the fuel consumption. After injector’s running-in of approximately 3 hours the leakage dropped-down to negligible values lower 0.5%. The temperature of the injector external surface did not exceed 40^{o}C in any of these experiments.

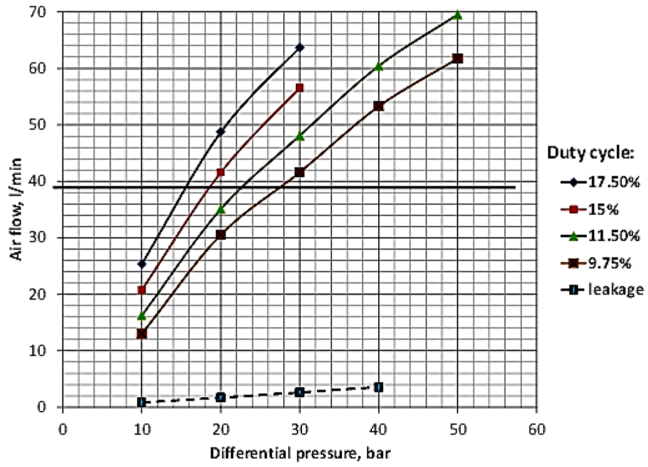

Fig.5. Flow performance of the virgin gaseous fuel injector. Working fluid – air; needle lift - 0.15 mm

Fuels studied and testing procedure

A commercial 95 RON gasoline was used for feeding the engine in original configuration. Engine performance at various ignition timing values was studied at the rated power. An optimal ignition timing corresponding to the maximal engine efficiency (minimal specific fuel consumption) was found. After that, a load characteristic at the rated engine speed and optimal ignition timing was measured. The following parameters were measured or calculated:
- The engine speed and power;
- In-cylinder maximal pressure and maximal pressure-rise rate (indicatory diagram);
- Exhaust gas temperature;
- Air flow-rate;
- Fuel consumption;
- Air excess factor (lambda);
- Concentrations of gaseous pollutants NO_x, CO, THC, CO_2 in the exhaust gas;
- PM mass emissions.

Composition of the reformate gas used in the reported study (Table 3), was found in the previous methanol reforming simulations [19] as the most probable product of the reforming designed to provide full methanol conversion in the case of water-to-methanol ratio value of 1:1. This ratio was chosen since lower steam-to-methanol ratios might cause coke formation on the catalyst and higher steam-to-methanol ratios reduce specific and volumetric energy density of the primary mixture.

Table 3. Considered methanol reformate

H_2, molar/mass fraction, %	CO_2, molar/mass fraction, %	Air/Fuel mass stoichiometry ratio	Lower heating value, MJ/kg
75/12	25/88	4.13	14.61

Engine performance at various ignition timing values was studied at the maximal achievable power value. The gaseous fuel pressure at the injector inlet was varied in the range 30 – 50 bar and the air excess factor was varied in the range 1.2 – 1.5 under constant ignition timing and constant fuel injection start phasing. The load characteristic was measured for these tuning values and a comparison with the engine performance measured under gasoline feeding was carried out.

Values of the in-cylinder maximal pressure and the maximal pressure rise rate were obtained by processing the indicatory diagrams measured at each operating regime. Mean weighted values of 25 consecutive working cycles were used as the measurement results.

Engine efficiency calculations

In the case of the methanol reformate feeding the engine efficiency η_r was calculated using as a basis the primary liquid methanol consumption and following the methodology described in [21]:

$$\eta_r = 3600/(bsfc_{liq} \cdot LHV_{liq}) \qquad (2)$$

Where: $bsfc_{liq}$ is a brake specific consumption of the primary liquid fuel, g/(kW·h); LHV_{liq} is a lower heating value of liquid methanol, MJ/kg.

$$bsfc_{liq} = bsfc_{mix}/m_{mix} \qquad (3)$$

Here: $bsfc_{mix}$ is a brake specific consumption of the methanol reformate; m_{mix} is a mass of the methanol reformate per 1 kg of the primary liquid methanol. The value of m_{mix} was calculated as follows:

$$m_{mix} = (M_{wat}/M_{met}) \cdot (W/A) + 1 \qquad (4)$$

Here M_{wat} and M_{met} are molecular weights of water and methanol, respectively; W/A – water-to-alcohol ratio.
Values of η_r were compared with a brake efficiency of the engine fed by gasoline.

RESULTS AND DISCUSSION

Since an active resistive load (rheostat) was used for the gen-set loading, the entire engine performance parameters were referred to the generator electric power. The rated value of the latter was measured to be 1.6 kW at 3000 rpm (the engine brake power 2.2 kW) that closely corresponds to the typical efficiency of this generator 0.73 [20].

GASOLINE FEEDING

Figs. 6 and 7 show dependence of the engine performance and harmful emissions on ignition timing.

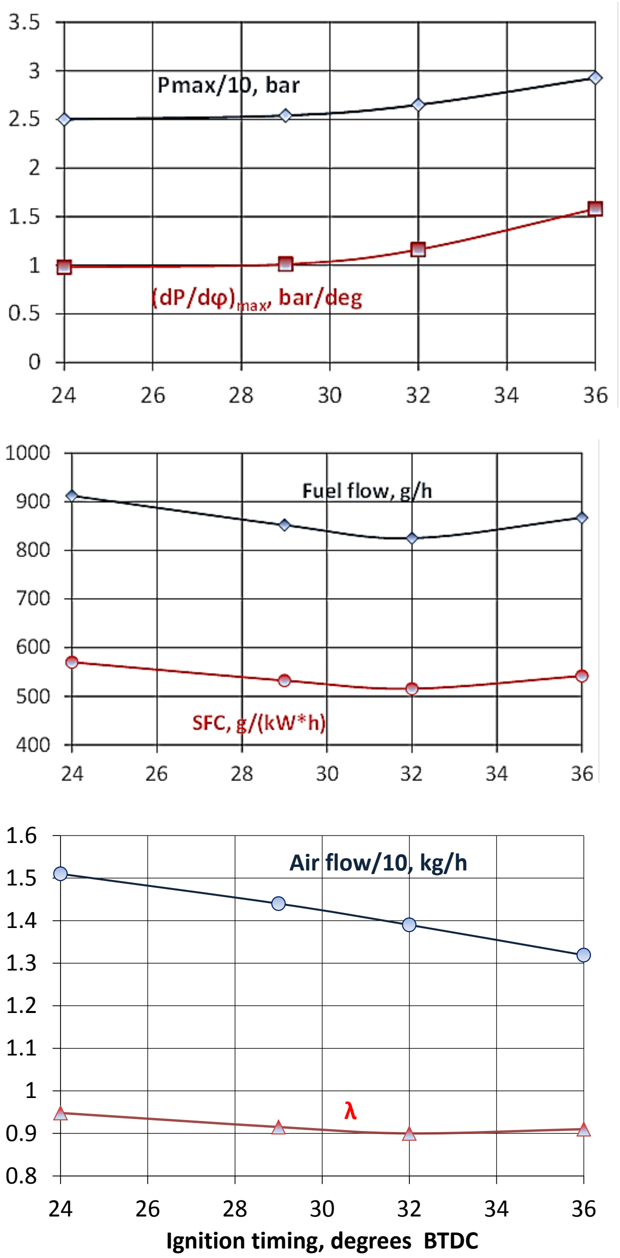

Fig.6. Dependence of the ROBIN EY20 engine performance on ignition timing: fuel – gasoline; 1.6 kW @ 3000 rpm.

As can be seen, minimal value of the specific fuel consumption (the engine maximal efficiency) was found under the ignition timing value of 32^o BTDC. At this tuning the in-cylinder maximal pressure and maximal pressure rise rate were found to be 26.5 bar and 1.2 bar/deg.

Dependence of the gasoline-fed engine performance and harmful emissions on the generator load at 3000 rpm and the ignition timing of 32^o BTDC is demonstrated in Fig.8 and 9. As can be seen in Fig.8, cycle-to-cycle variability of the in-cylinder maximal pressure at the rated power was found to be +3…-5% and was increasing up to ±50% under low loads. It seems that this results from the simple and cheap design of the ROBIN EY20 governor and carburetor.

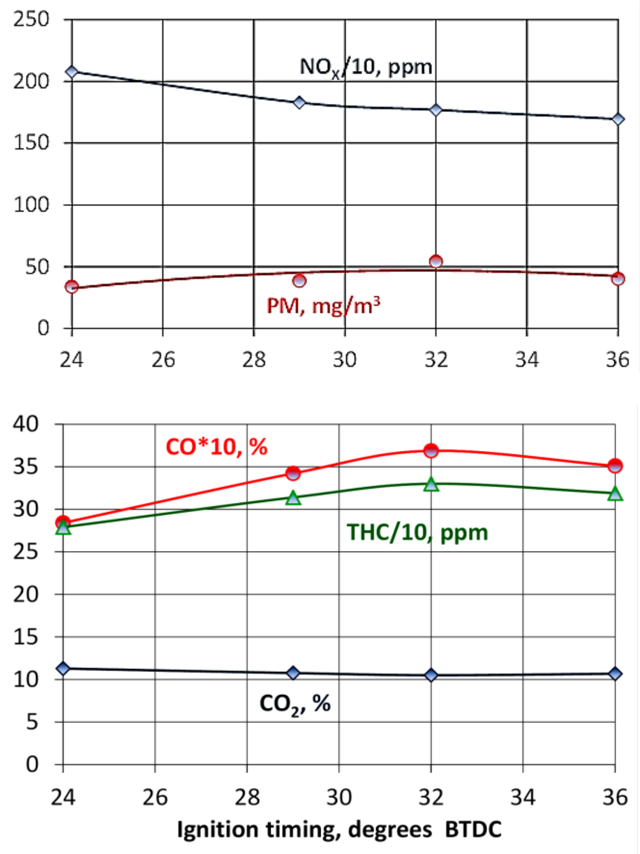

Fig.7. Dependence of the ROBIN EY20 engine harmful emissions on ignition timing: fuel – gasoline; 1.6 kW @ 3000 rpm.

REFORMATE FEEDING

The engine performance presented below was measured for the methanol reformate composition shown in Table 3. The following injection parameters were achieved at this stage of the study: the pressure at the injector's inlet was 40±2 bar, the needle lift - $0.15^{+0.05}$ mm and the maximal injection duration 100 deg. of the crankshaft angle. The maximal generator power achieved with these injection parameters was 1000 W @ 3000 rpm. Ignition timing was kept the same as under gasoline feeding (32^o BTDC) that allowed revealing more clearly the gaseous fuel combustion peculiarities.

Fig.10 shows dependence of the engine performance parameters on the generator load under engine feeding by methanol reformate. Dashed lines show maximal and minimal values of the in-cylinder maximal pressure. As can be seen, the in-cylinder maximal pressure value was substantially higher compared with gasoline at the same load - 26 bar and 15 bar for the methanol reformate and gasoline, respectively. The same effect was found for the maximal pressure rise rate value - 1.3 bar/deg. and 0.95 bar/deg. for the methanol reformate and gasoline, respectively. These findings are explained by the higher combustion velocity of the hydrogen-rich gaseous fuel.

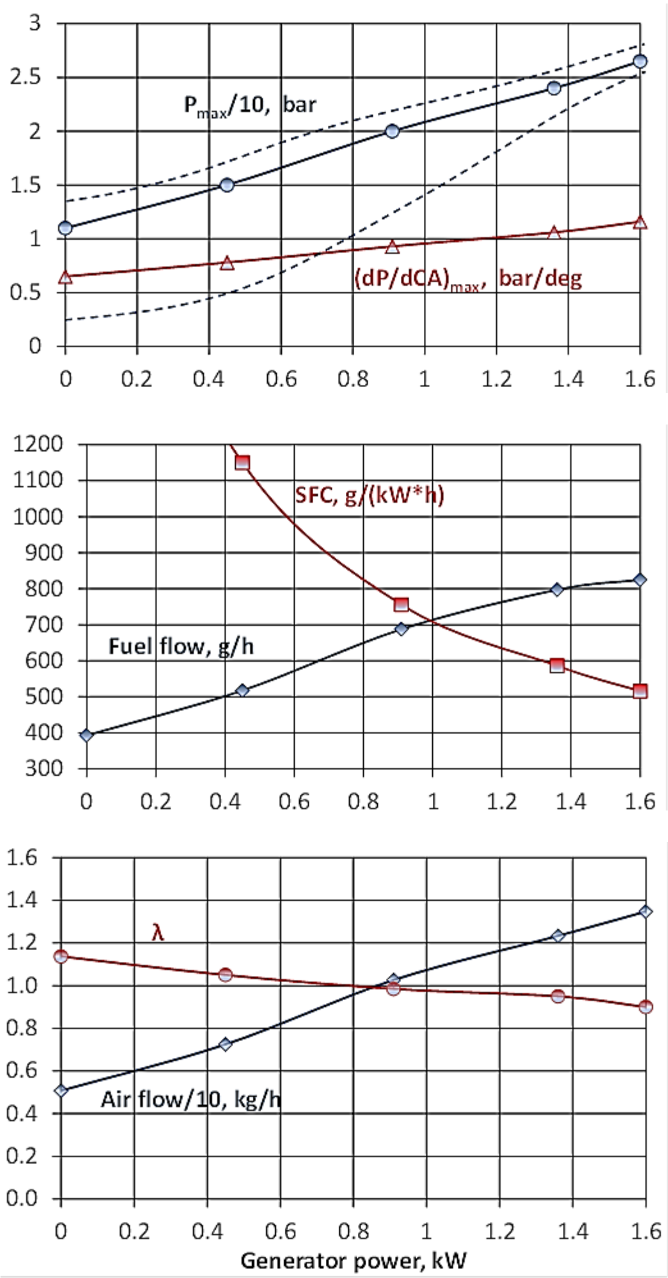

Fig. 8. Load performance of the ROBIN EY20 engine: fuel – gasoline; engine speed - 3000 rpm; ignition timing - 32^o BTDC.

Much lower values of cycle-to-cycle variability of the in-cylinder maximal pressure under the gaseous fuel operating in comparison with gasoline were noted: +7.5%...-2% for the gaseous fuel and ±27% for gasoline at 1000 W operating mode. This finding is proving that under reformate direct injection the used ECU ensured more stable engine operation.

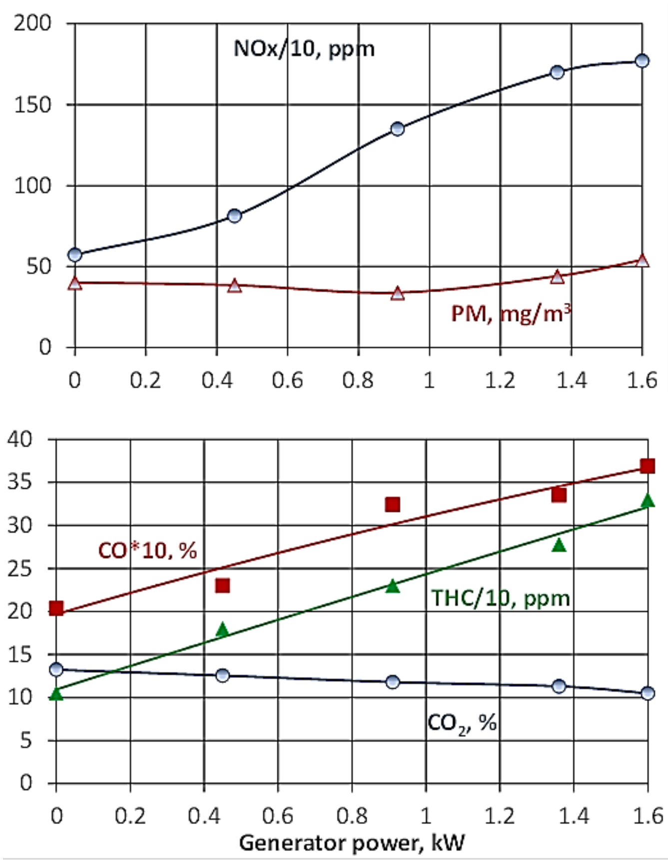

Fig.9. Dependence of the engine harmful emissions on the generator load: fuel – gasoline; engine speed - 3000 rpm; ignition timing - 32^o BTDC.

Substantially higher specific fuel consumption (SFC) was found under the engine feeding with methanol reformate as compared to gasoline. Even under λ=1.5, the SFC value was 2000 g/(kW*h), i.e. higher by a factor of 2.8 than under gasoline feeding at the same operating regime. Decrease of the λ value down to 1.2 led to the SFC rising up to 2600 g/(kW*h). This fact is explained by the substantially lower (by a factor of 3) heating value of this gaseous fuel compared to gasoline. This is a result of the methanol reformate composition used (Table 3). Mass fraction of the combustible part (H_2) is only 12%, while the rest is a diluent - the non-combustible component CO_2. It should be mentioned here that there is a potential of the SFC decrease by optimizing the methanol reformate composition and increasing the engine compression ratio. The latter is possible because of the high knock resistance of hydrogen-based gaseous fuels.

It is important to note that even under the idle operating regime the exhaust gas temperatures were by a factor of 1.3 – 1.5 higher than the temperature required for full conversion of methanol (300^{o}C).

The measured values of pollutant emissions at the operating mode of 1000 W at 3000 rpm are shown in Table 4.

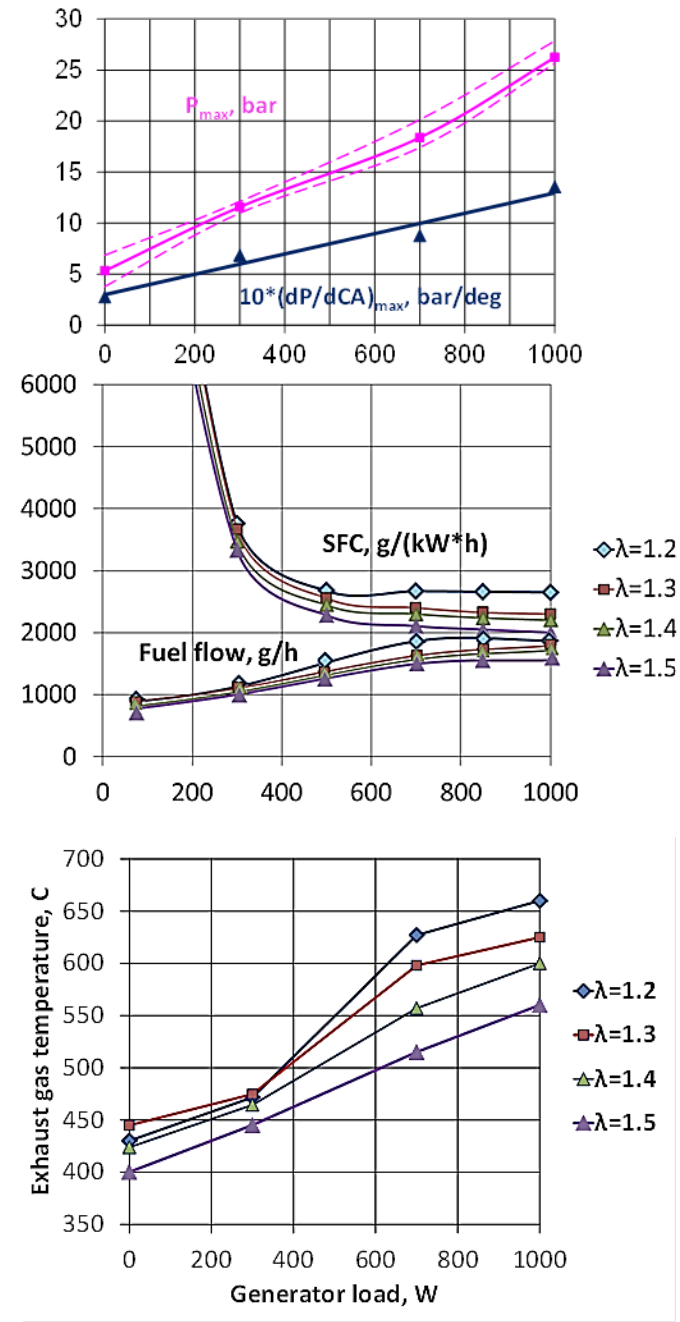

Fig.10. Load performance of the ROBIN EY20 engine: fuel - methanol reformate; engine speed - 3000 rpm; ignition advance 32^o BTDC.

Table 4. Exhaust gas harmful emissions: 1000W @ 3000 rpm.

Type of fuel	NOx, ppm	THC, ppm	CO, %	CO_2, %	PM, mg/m^3
Gasoline, λ =1.12	1410	243	3.1	11.8	37
Methanol reformate, λ =1.5	207	53	0.12	8.0	0.05

The data presented in Table 4 show that feeding the SI engine with methanol steam-reforming products has a great potential of emissions mitigation, as compared with gasoline. The following findings may be noted:

- Decrease of NO_x emissions by a factor of 6.8 due to lean combustion and the subsequent reduction of maximal in-cylinder temperatures;
- Zero-impact PM emissions at the limit of the measuring device sensitivity;
- Reduction of the total hydrocarbon (THC) emissions by a factor of 6;
- Reduction of CO emissions by a factor of 25;
- Reduction of CO_2 emissions by a factor of 1.5.

The obtained drastic reduction of pollutants emission allows considering a possibility of excluding catalytic converter in an SI engine fed by the gaseous methanol reformate.

It must be noted that the products of non-complete combustion (THC, CO and PM) measured during the engine operating with methanol reformate are most probably the result of the engine's lubricant penetration into the combustion chamber. This fact suggests ways of mitigating these pollutants emission. Among the possible solutions, improvement of the lubricant quality, valves stem sealing, piston rings and improvement of the piston external surface design can be mentioned.

Presence of CO_2 in the exhaust gas of the engine fed by the methanol reformate is not a result of the fuel combustion process as well. This component enters into the cylinder as a part of the gaseous fuel – products of methanol steam reforming (the methanol reformate considered in this work contained 25% molar fraction of CO_2). Some amount of CO_2 is generated as a result of lubricant oil combustion. Further reduction of CO_2 emission can be achieved by improving the engine's efficiency and using suitable lubricants.

ENGINE EFFICIENCY

The engine efficiency was calculated taking into account the generator efficiency value 0.73 (see above). For the case of the engine feeding with the methanol reformate the efficiency was calculated based on the primary liquid methanol consumption (for the water/methanol molar ratio 1:1) using the eq. (2)-(4). Obtained values of the engine efficiency for the cases of the engine feeding by gasoline and methanol reformate are shown in Fig. 11. The achieved improvement in the engine efficiency under the feeding with methanol reformate as compared to gasoline, is shown in Fig. 12.

Relatively low values of the efficiency measured are explained by the low-load regimes of the engine operating and low compression ratio of the studied engine. As can be seen in Fig.12, in our experiments efficiency of the engine fed with the methanol reformate was found to be higher by about 20% at 1000 W and by 65 - 70% at low loads. This resulted mainly due to the following reasons. Firstly, the actual reformate LHV is about 14% higher than that of liquid methanol while the efficiency is calculated based on the primary liquid methanol consumption – eq. (2) - (4). Secondly, the engine conversion from carburetor to direct injection technology allows improving cycle-to-cycle identity and excluding the working mixture loss at the time of valves overlapping. Thirdly, the lean-burn operation with methanol reformate significantly reduces the throttling losses, which are increasing with the load reduction. This explains the fact that the greatest efficiency improvement was achieved at the low loads. According to the results of previously performed simulations [21], efficiency improvement resulting from the lean-burn operating with methanol reformate and its higher LHV can be assessed as approximately 13% for the rated power operating mode.

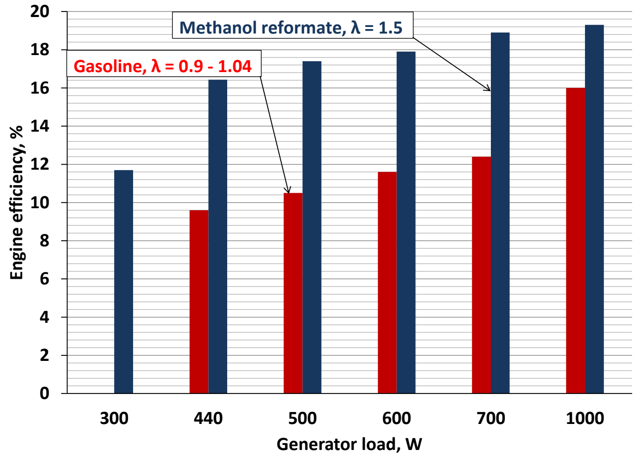

Fig.11. Efficiency of the ROBIN EY20 fed by gasoline and methanol reformate.

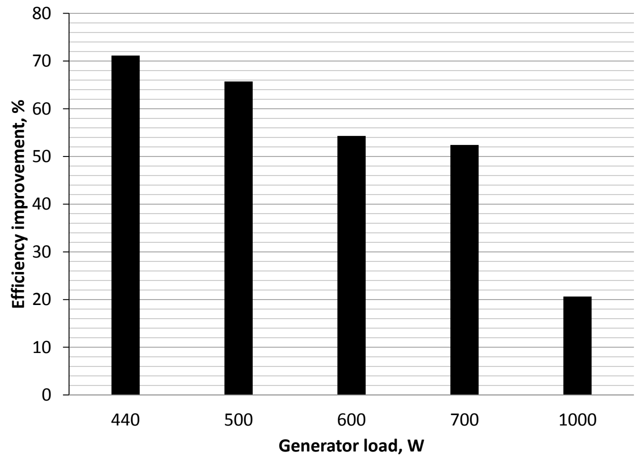

Fig.12. Engine efficiency improvement under the feeding with methanol reformate as compared to gasoline.

SUMMARY AND CONCLUSIONS

A novel approach of a direct injection ICE with TCR and a high-pressure fuel reforming was suggested that can eliminate the backfire, pre-ignition and reduced volumetric efficiency problems without a need of limiting hydrogen content in the reformate gas or injecting a liquid non-reformed fuel at high engine loads. A concept of the SI engine with direct-injection of methanol reforming products that can serve as a basis of a DI ICE with high-pressure TCR was developed and built for the first time. An injector for direct injection of gaseous reformate was developed, manufactured and tested for the purpose of this study.

Comparison of the engine performance under gasoline and methanol reformate feeding was carried out at the optimal ignition timing for gasoline that ensured maximal engine thermal efficiency. In-cylinder maximal pressure and the value of pressure-rise rate were higher under the reformate feeding by 75% and 37%, respectively. Much lower value of an in-cylinder maximal pressure cycle-to-cycle variability under the reformate operation in comparison with gasoline was noted: +7.5%...-2% for the gaseous fuel (methanol reformate) and ±27% for gasoline at 1000 W operating regime. This finding may be explained by high cycle-to-cycle variability under gasoline operating due to the simplified and cheap design of the engine original governor. Under the reformate operation the ECU ensured more stable engine operating.

It was found that engine feeding by methanol steam reforming products has a great potential of pollutant emissions mitigation as compared with gasoline. NO_x concentrations in the exhaust gas were reduced by a factor of 7 as a result of the lean combustion and lowering in-cylinder temperatures. Particle mass emissions were decreased to zero-impact levels. Harmful emissions of the target pollutants THC, CO and the GHG gas CO_2 were reduced by a factor of 6, 25 and 1.5, respectively. The obtained drastic reduction of pollutants emission allows considering a possibility of excluding catalytic converter in an SI engine fed by the gaseous methanol reformate.

Further decrease in emission of the products of hydrocarbons non-complete combustion (THC, CO and PM) may be achieved through diminishing the engine's lubricant penetration into the combustion chamber by valves stem sealing, improvement of piston rings and a piston external surface design. Optimization of the lubricant composition is another important step toward further emissions reduction down to zero-impact levels. An experience gained by the automotive industry in this field can be very useful for small engine applications.

Efficiency of the engine fed by the methanol reformate was found to be higher by about 20% at the maximal tested power. The achieved efficiency improvement grows up to some 70% under low-load operating modes. It is clear that the measured efficiency gain is the combined effect of the following two steps: a) the engine conversion from carburetor to direct injection technology and b) lean-burn operating with methanol reformate.

Our forthcoming research is planned to be focused on:
- Comparative analysis of the SI engine performance for the cases of gasoline and methanol reformate direct injection;
- Study of compression ratio increase potential for both mentioned above cases;
- Study of an engine performance improvement potential by changing the methanol reformate composition.

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ABBREVIATIONS

BTDC	before top dead center
BTE	break thermal efficiency
CNG	compressed natural gas
DI	direct injection
ECU	engine control unit
GDI	gasoline direct injection
GHG	greenhouse gas
ICE	internal combustion engine
LHV	lower heating value
RON	research octane number
SFC	specific fuel consumption
SI	spark ignition
SPFI	spark plug fuel injector
TCR	thermo-chemical recuperation