# EXPERIMENTAL STUDY OF COMPRESSION AND COMBUSTION PROCESSES IN A VERY SMALL ENGINE

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***Abstract***

*The main objective of the present study is a better understanding of the fundamental compression, ignition and combustion processes in a miniature engine. To study these processes in detail a small rapid compression machine was built. The bore diameter and stroke of this machine was D=6mm and L=12mm, respectively. Stoichiometric propane-air and hydrogen-air mixtures were employed in experiments. The catalyst heated to necessary temperature was used to ignite the mixture. It was found experimentally that the compression process in such engine was isothermal. The serious problem in a rapid compression machine is also dead volume created by crevices. Isothermal compression and dead volume are responsible for very low value of the maximum compression pressure. It was also found that the mixture could not be ignited in a very small volume of the catalytic prechamber (with dimensions $\phi$2.8mm×4.7mm). Some additional volume in a form of the combustion chamber should be added to make possible ignition and following to it combustion of the mixture. Part of the experiments was carried out on a working model engine of the same piston diameter. Parameters of this model engine were measured at specially designed miniature test-bench. Cylinder pressure and engine performance were determined.*

## 1. Introduction

Progress in technology and new techniques of high-precision fabrication made possible miniaturization of mechanical and electromechanical devices. Such complex mechanical systems as robots, pumps, motors, airplanes and helicopters are now being developed in micro scale. Miniaturized mechanical devices need micropower generation units with high specific energy (they have to be small, light, durable and reliable). They have multiple applications, not only in miniaturized mechanical systems, but also in different electronic devices. In all these applications the power-supply units are limited by weight. From this point of view batteries are not acceptable as a source of energy because they have low specific energy in contrast with liquid hydrocarbon fuels. Liquid hydrocarbons have an extremely high specific energy density – typically 45 MJ/kg against 1.2 MJ/kg for the top batteries currently available. The micropower generation units produce power in the range of milliwatts to watts. The order of magnitude of such units is 1cm in size.

Up-to-date trends in development of micropower generation units are presented in a review paper by Fernandez-Pello [1]. Several microcombustors [2-5] and internal combustion microengines [6-9] are currently being developed. They appear to operate with acceptable combustion efficiency. The idea of micropower generation is very new and its practical realization still is in the feasibility stage.

Let us characterize briefly the most advanced projects. At the MIT Gas Turbine Laboratory a MEMS-based gas turbine power generator with a total volume of 300mm$^3$ is being developed to produce 10-20W of electrical power [6]. The UC Berkeley Combustion

Laboratories are developing a liquid hydrocarbon-fueled a Wankel type internal combustion rotary engine with 2.3mm and 1mm size of rotor to produce power in the miliwatt range [7]. At Honeywell a free piston microengine based on homogeneous charge compression ignition of hydrocarbon fuels is being developed to produce ~10W of electrical power [8]. The Georgia Institute of Technology is developing a free piston electrical power generator based on ferromagnetic piston oscillating in a magnetic field to produce ~12W of electrical power [9].

The interest in producing micro-power generation units creates new opportunities for studying thermodynamic processes in small size piston engines. Such processes as compression, expansion, ignition, combustion, and gas exchange are of special interest. In the engine with high aspect ratio (defined as the ratio of the surface of the combustion chamber to its volume) most of these processes strongly depend on combustor geometry. In such engine at least compression and combustion processes are much more critical than in low aspect ratio engines and this leads to significant heat losses and local flame quenching. Experimental results as well as theory predict that a critical diameter (in a millimeter length scale) is necessary for propagation of an exothermic reaction. Power generated by a micro-engine depends on the results of competition between heat losses from the flame to the cold boundary and heat release of chemical reaction from a micro-volume. The specific heat transfer rate $\dot{q}$ from a stagnant gas occupying the chamber having a volume V and a surface S can be expressed for an average heat flux and density by the expression [10]:

$$\dot{q} = \frac{\dot{Q}}{m} = \frac{\int_S q\,dS}{\int_V \rho\,dV} = \frac{\bar{q}}{\bar{\rho}}\frac{S}{V},$$

where the new symbols are: $\dot{Q}$ - heat transfer rate, q – specific heat flux, m – mass and ρ - density.

It is evident from this equation that the specific heat transfer rate increases with the surface area-to-volume ratio, S/V. This ratio is inversely proportional to the characteristic dimension of the chamber.

## 2. Experimental details

To study fundamental processes of compression, ignition and combustion in a high aspect ratio engine a small rapid compression machine has been developed. Physical processes taking place in this machine were very similar to those in a real engine. The developed rapid compression machine is shown in Fig. 1.

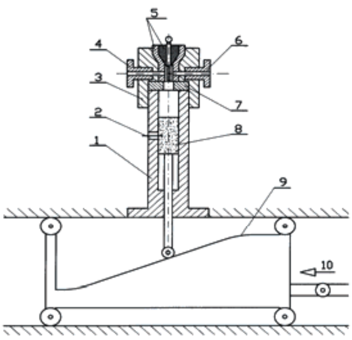


*Fig. 1. Rapid compression machine: 1 – cylinder, 2 – inlet channel, 3 – cylinder head, 4 – outlet channel with valve, 5 – catalytic igniter, 6 – fastener of pressure transducer, 7 – combustion chamber, 8 – compressing piston, 9 – cam, 10 – rod from the driving piston*

The mean piston speed was w=4m/s which corresponded to approximate engine speed n=10000 rpm. The cylinder pressure was measured by a Optrand Incorporated miniaturized pressure transducer PSI-S M3×0.5, model C832D1 with sensitivity 38.7 mV/bar. Propane and hydrogen were used as fuels in stoichiometric mixtures with air. Homogeneous mixture was composed before the experiments. It was stored in a special reservoir under small overpressure. A mixture preserved in this reservoir was used for a set of tests guaranteeing reproducibility of experimental conditions.

The rapid compression machine was equipped with a driving (10) and compressing (8) piston. To compress the mixture in a combustion chamber (7) the driving piston shifts the cam (9) from one limit position to another. The driving piston is stopped at the end of the stroke with a shock absorber. The final position of the compression piston (8) is determined by the curvature of a final segment of the cam (9). The system is set in motion by admitting air pressure into a cylinder with a driving piston. The ratio of driving to the compressing piston diameters is 3.3: 1. The rapid compression machine operated at the compression ratio, which depended on the size of the combustion chamber (7). For the combustion chambers tested in experiments it was in the range $\varepsilon$=4.5-7.6. The cylinder head was equipped with an outlet channel (4) and a channel for pressure transducer (6). Before each experiment the mixture was forced to flow through the cylinder. To this end the inlet (2) and outlet (4) channels were used. Afterwards the valve of the outlet channel was closed and the machine was ready for experiment.

The combustion chambers of different diameters and heights were examined in this cylinder head to obtain the maximum combustion pressure.

Before the experiments cooperating parts of the machine were carefully sealed up and the air-tightness of the system was positively examined at a pressure of 1MPa.

Miniature test-bench was designed and manufactured to determine model engine performance. A spindle mounted on the wall-bearings was located in a jacket of the test-bench. The microengine was placed on the one side of it and a weighed single-arm lever on the other one. The lever pressed on a scale of the precise balance.

## 3. Results and discussion

### 3.1 Functional Tests

Three types of actions were carried out during functional tests:

- Careful examination of the geometry of the combustion volume and determination of the dead space in the cylinder head.
- Examination of the air-tightness of the system, and
- Inspection of the pressure measurements correctness.

The initial working combustion volume consisted of cylindrical micro combustion chamber with dimensions $\phi$2.8mm×4.7mm. The volume of this micro chamber was 27mm$^3$. The chamber was equipped with spiral Pt-wire located at its axis. The characteristic parameters of the prechamber are indicated in Table 1, where the meaning of the appropriate symbols is:

d – diameter, h – height, $p_{max}$ – maximum compression pressure, S – surface, V – prechamber volume, $V_{1c}$ – minimum cylinder volume (complete volume taking into consideration also the dead space), $V_2$ – maximum cylinder volume, $\varepsilon$ - compression ratio and AR – aspect ratio, which is the ratio of S/V. The volume of all crevices in the cylinder head in the region of the prechamber was evaluated to be close to 25mm$^3$. Because the dead space volume was the same order of magnitude as the prechamber volume it considerably decreased the initially planned compression ratio.

*Table 1. The most important parameters of the prechamber*

| d mm | h mm | $p_{max}$ bar | S $mm^2$ | V $mm^3$ | $V_{1c}$ $mm^3$ | $V_2$ $mm^3$ | ε | AR $mm^{-1}$ |
|---|---|---|---|---|---|---|---|---|
| 2.8 | 4,7 | 7.6 | 53.6 | 27.0 | 52.0 | 391 | 7.5 | 2.0 |

The air-tightness of the cylindrical head and the complete system was checked in special experiments and appeared to be satisfactory in a small scale of time.
A special cylinder head was fabricated to examine correctness of the pressure measurements. The volume above the piston in this cylindrical head was connected by means of a valve with a large vessel containing air under known pressure. Opening the valve made possible to register a pressure rise up to the given known level by a pressure measuring system.

### 3.2 Compression

Compression tests on the rapid compression machine equipped purely with the prechamber showed that the maximum compression the overpressure was equal to $\Delta p$=6.6bar (the mean value from many tests). Such low value of overpressure is a result of extremely high heat loss to the walls under the influence of high aspect ratio. The aspect ratio of the combustor studied is $S/V \approx 2.0$. Its value is at least the order of magnitude bigger than the combustor aspect ratio of conventional engines. Under such conditions the compression process practically proceeds isothermally. An increase of the combustion volume does not change much the aspect ratio (see Table 2) and this was responsible for a high level of heat loss and an appropriate isothermal compression.

*Table 2. Parameters of the combustion chambers, which were used in compression experiments together with prechamber*

| n | d mm | h mm | $p_{max}$ bar | S $mm^2$ | V $mm^3$ | $V_{1c}$ $mm^3$ | $V_2$ $mm^3$ | ε | AR $mm^{-1}$ |
|---|---|---|---|---|---|---|---|---|---|
| A | 3.0 | 2.5 | 5.6 | 85.6 | 44.6 | 69.6 | 409 | 5.8 | 1.9 |
| B | 3.0 | 3.0 | 5.8 | 90.3 | 48.2 | 73.2 | 413 | 5.6 | 1.9 |
| C | 4.0 | 2.5 | 5.0 | 104.4 | 58.4 | 83.4 | 423 | 5.0 | 1.8 |
| D | 4.0 | 4.0 | 4.8 | 123.2 | 77.2 | 102 | 442 | 4.3 | 1.6 |
| E | 5.5 | 2.5 | 4.5 | 138.6 | 86.3 | 111 | 451 | 4.0 | 1.6 |
| F | 3.0 | 1.0 | 5.3 | 71.4 | 34.1 | 59.1 | 398 | 6.7 | 2.1 |
| G | 3.0/6.0 | 1.5 | 4.9 | 113.1 | 51.7 | 76.7 | 416 | 5.4 | 2.2 |

*$p_{max}$ is the maximum compression pressure

To conform suspicion of near isothermal compression it was decided to measure the instant temperature inside the combustion chamber. Appropriate temperature measuring system was developed by the Institute of Orogen Mechanics in Krakow. A sensitive resistant probe was located in one of the combustion chamber. The outline of the combustion chamber with thin wire temperature probe is shown in Fig. 2 and an exemplary result of temperature measurement during compression stroke in Fig. 3.
The maximum temperature rise at the center of the combustion chamber was 56°C against 290.6°C of adiabatic compression temperature calculated on the basis of compression ratio ε=5.6. Temperature measurements confirmed significant role played by a cooling effect of the walls on the compression and combustion processes.

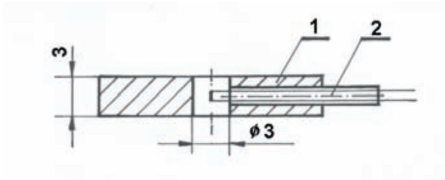

*Fig. 2. Outline of the combustion chamber (type B in Table 2) together with the temperature probe: 1 – combustion chamber, 2 – temperature probe*

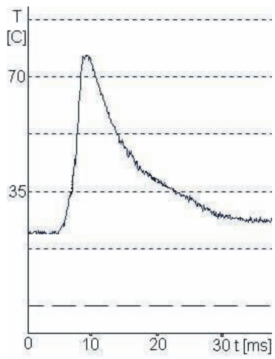

*Fig. 3. Temperature inside the combustion chamber during compression process*

### 3.3 Ignition

The most important part of catalytic igniter (5 in Fig. 1) is small prechamber of 2.8mm diameter with a centrally located catalytic spiral wire. The wire can be heated externally to the temperature up to 880°C. In the course of the working cycle a contact between the fresh fuel-air-mixture with the catalytic wire is in progress during the compression stroke. It is essential that heterogeneous ignition of the mixture occurs on the surface of the catalyst during the flow of the fresh mixture into the small chamber with catalyst. Heterogeneous catalysis on the developed surface of the catalyst induces chemical activity of homogeneous mixture. In such case a later autothermal ignition can meet chemically active mixture. It follows from the earlier experiments [11], that the chemical reactions in the fuel-air mixture at the surface of the catalyst are going on, if mixture and catalyst temperature exceeds by a certain value the temperature of the beginning of the heterogeneous or autothermal reactions. Thus it is essential that for propane/air mixture the temperature of the catalyst should exceed, during the compression stroke, the temperature of the beginning of autothermal ignition, which is in the range from 250 to 500°C (its value depends on the equivalence ratio).

Attempts to ignite stoichiometric propane/air mixture in a catalytic prechamber completely failed in the entire range of the catalyst temperature up to 880°C. It was assumed that the wall quenching effect counteracts flame formation because the quenching distance for such flame is of the same order of magnitude as a prechamber diameter. In subsequent experiments stoichiometric hydrogen/air mixture was used. Quenching distance of hydrogen flames is several times less than that of propane flames, therefore it was expected that the stoichiometric hydrogen/air mixture would be ignited. However, experiments showed that it was not possible to ignite also this mixture. The explanation of this phenomenon may be twofold: 1) The catalyst could not ignite the mixture in a form of a flame in vicinity of the cold walls, 2) The reaction rate initiated by a catalyst is too low to rise the pressure in the presence of the cold walls. This observation indicates that the boundary conditions have different effect on flame propagation and its initiation by a catalyst.

Instant ignition of both propane and hydrogen mixtures with air was obtained in the case when additional combustion chamber was attached to the prechamber. Earlier experiments showed that the rise of the catalyst temperature decreased ignition delay time and that the pure catalytic ignition was possible, when the catalyst temperature was higher than 700°C [11]. It was decided to use in the present experiments the highest available temperature of 880°C.

### 3.4 Combustion

To optimize the combustion process several micro combustion chambers were prepared and tested in the compression experiments. Some of their most important parameters are given in Table 2. Letter notations are the same as in Table 1.
Configuration of different combustion chambers in relation to prechamber and engine cylinder is shown in Fig. 4.

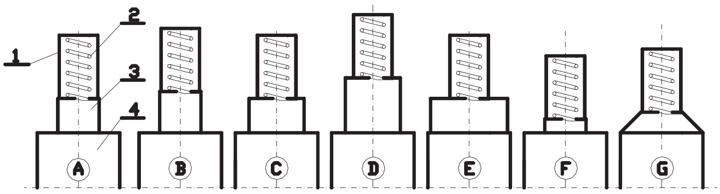


*Fig. 4. Configuration of different combustion chambers in relation to prechamber and engine cylinder (in scale): 1 – prechamber, 2 – catalytic wire, 3 – combustion chamber, 4 – engine cylinder. Letters in a circle indicate the notation shown in Table 2*

Experiments showed that the pressure rise as a result of combustion is limited by a value of about 4.0-4.5 bars. It is very low increase in comparison with conventional engines.
The qualitative indicator diagrams of the engine cycle characterized by a substantial loss to the walls of the heat released in the process of compression and combustion are shown in Figs. 5 and 6.

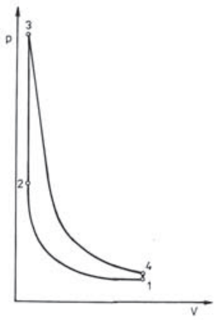


*Fig. 5. Qualitative cycle of a high aspect ratio engine in p-V coordinates*

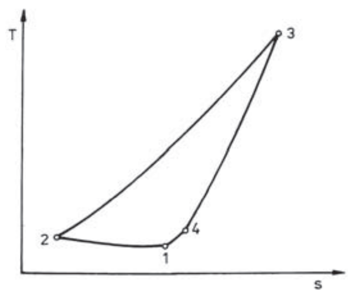


*Fig. 6. Qualitative cycle of a high aspect ratio engine in T-s coordinates*

### 3.5. Model engine tests

A prototype of a model engine was built on the basis of parts of the commercial engine. Such parts as piston, cylinder and crankshaft were used. The cylinder head was modified: pressure transducer was installed to measure cylinder pressure and the new cooling system was designed to make it more reliable. The outline of the model engine on the test bench is shown in Fig. 7 and the cylinder pressure record in Fig. 8.The combustion chamber of type F was used in the model engine (see Fig. 4 and Table 2).
Measurements of air compression pressure in the cylinder at the engine rotational speed of 6000 rev/min showed its increase up to Δp=7bar. This pressure is considerably higher than

that determined at the same combustion chamber but in the rapid compression machine. We can conclude from this that a relatively large dead space volume present in this machine plays very negative role during compression stroke.

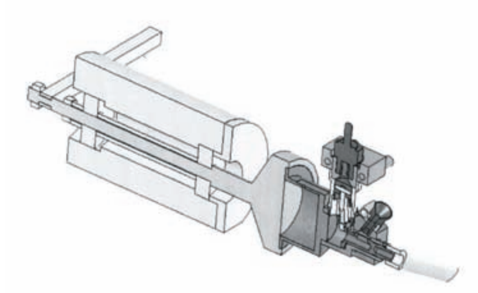

*Fig. 7. Outline of the model engine on the test bench*

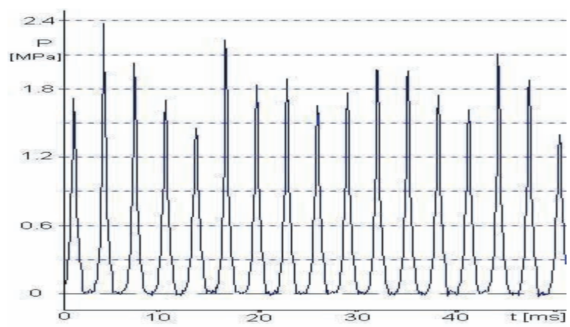


*Fig. 8. Cylinder pressure as a function of time for model engine*

It was also found from the indicated diagram of the engine that the pressure rise caused by combustion is usually much higher than that observed in a rapid compression machine (relation like ≈ 16/10). The observed phenomenon can be explained by smaller dead space volume and higher engine rotational speed.

Usually pressure rise is markedly higher for higher engine rotational speed. It is evident that for higher engine rotational speed relation between chemical reaction time and cooling effect time improves. Some beneficial role may also play thermal state of the cylinder head: its temperature is usually much higher than temperature of rapid compression machine.

Performance of the engine in a form of torque as a function of engine rotational speed was determined for three different kinds of fuel: MN60 (60% of methanol, 25% of castor oil and 15% of nitromethane), EN60 (60% of ethanol, 25% of castor oil and 15% of nitromethane) and M80 (80% of methanol and 20% of castor oil). Exemplary drawing of torque as a function of engine rotational speed is shown in Fig. 9.

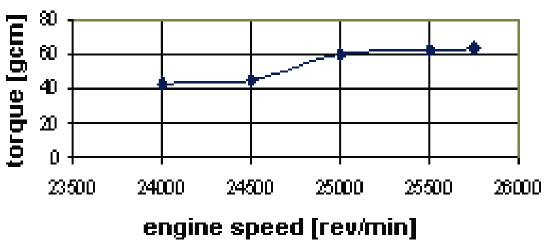


*Fig. 9. Exemplary drawing of torque as a function of engine rotational speed*

## 4. Conclusions

It was found experimentally that in a micro-engine with a very high aspect ratio the compression process is isothermal, which could be explained by intensive heat loss to the walls. The serious problem in this engine is also dead volume created by crevices, which is relatively big in comparison with the volume of the combustion chamber. The dead volume considerably decreases compression ratio. Both factors (isothermal compression and dead volume) are responsible for a relatively low value of the maximum compression pressure.

It was also found that it is not possible to ignite the stoichiometric mixture of propane or hydrogen with air by means of a catalyst in a very small prechamber with diameter of 2.8mm even if the catalytic wire is heated to the temperature of 880°C. This was unexpected observation because the prechamber's diameter exceeds many times the quenching diameter for stoichiometric hydrogen flames. Most probably the catalyst could not form a flame in vicinity of the cold walls or the reaction rate initiated by a catalyst was too low to raise the pressure in the presence of the cold walls.

Small additional combustion chamber attached to the prechamber made possible ignition and following to it combustion.

Experiments showed very low pressure increase from combustion. This can be explained by heat losses to the walls. The pressure increase in investigated engine is at least two times lower than that in conventional engines.

Future work will include experimental study of mass and heat loss during compression stroke and combustion, turbulence and flame quenching by a cold wall. Additionally, an instant temperature will be measured during compression stroke and combustion. The compression and combustion processes will be still optimized.

**Acknowledgments**

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