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Free-Piston Linear Generator and the Development of a Solid Lubrication System

The free-piston linear generator (FPLG) is a new electromechanical generator. It converts chemical energy into electrical energy by means of a combustion process, a linear generator, and a gas spring. The FPLG does not use any crankshaft, which is responsible for a lot of losses. Thereby, the technology aims to have better properties than other electromechanical generators: higher efficiency over wide range of operating points, better noise–vibration–harshness package. This publication deals with the explanation of the concept, the characteristics of a FPLG, and one of the challenges in the development. In order to use a port scavenging, the emission issue is the challenge and has to be solved. One possible solution is the use of solid lubricants to substitute motor oil. On this way, the development methodology and one aspect of the development is explained.
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Introduction

Efficiency improvements in order to reduce the production of CO_2 are required in all technical areas of life to limit the anticipated climate changes [1] to an extent sustainable for humanity. Current approaches in the field of transportation cover both the increase of efficiency of conventional drives and the development of alternative power trains, e.g., hybrid, battery [2], fuel cell and range extender concepts, and combinations of these. The aim of every alternative power train design is to transform the stored energy into a kinetic energy with the greatest possible efficiency. In many concepts, electrical energy is necessary in order to drive electric motors.

This requirement is met particularly well by a free-piston linear generator (FPLG). It is capable of transforming chemical energy into electrical energy by means of a combustion process. As explained in Ref. [3], the high degree of efficiency and its independence of the load level are inherent in the design of the free-piston linear generator. This is achieved by keeping the system frequency constant and adapting to the power demand by variation of the stroke and compression ratio. Furthermore, these system characteristics give the free-piston linear generator the possibility to be powered by both conventional fuels such as petrol, diesel, and natural gas and by alternative fuels such as biofuel, synthetic fuel, hydrogen, etc.

The development goal for the FPLG is to use a port scavenged central opposed piston combustion chamber in order to save space and reduce complexity. One of the main challenges in this area is to guarantee the piston lubrication. A solid lubrication system as a possible solution is described in this paper.

Free-Piston Linear Generator System

In the field of free-piston motors, the free-piston linear generator is a promising design. The system described in the following is developed at the German aerospace center (DLR). The proof of concept took place in Stuttgart at the end of 2012 [3]. It was demonstrated that the control of a free-piston engine can be designed to allow robust operation even with large combustion variations [4].

The free-piston linear generator module in Fig. 1 is used to explain the principle consisting of a piston rod connected with a piston on either side, cylinders around the pistons, and a linear generator. At one end, the cylinder forms the combustion chamber which is powered by a liquid or gaseous energy carrier. The gas exchange of the combustion chamber is controlled by valves in the cylinder head. The opposite cylinder creates an adjustable gas spring. The spring rate of the gas spring is adjusted by means of regulating the air mass in the cylinder.

The linear generator is positioned between the two cylinders. Its mover is mounted on the piston rod. The mover consists of permanent magnets glued into a plastic matrix and additionally secured by a fiber-glass reinforced bandage. Stators with integrated generator windings and a cooling system enclose the mover and complete the linear generator.

The load cycle begins with the combustion piston at the top dead center (TDC). The chamber is filled with a compressed, flammable mixture previously introduced into the combustion chamber which is ignited by a spark plug. The two pistons, the rod, and the mover move toward the bottom dead center (BDC, direction of gas spring). The motion of mover and its magnets induces a voltage in the coils of the stator, which drives a corresponding electric current. About half of the energy released in the combustion is converted during the movement from TDC to BDC by the linear generator. The other half is stored in the gas spring and extracted on the return stroke (BDC to TDC). While the gas spring is compressed (the combustion piston is at its BDC), the combustion chamber is actively scavenged with charged, fresh air. After injection of fuel, the mixture is compressed and as the double-piston system arrives at the TDC, the next load cycle can begin. The variation of the output power can be achieved by adjustment of the inlet pressure and injection period in combination with the stroke. Thereby, the stroke is controlled by the air mass in the gas spring. The mechanical frequency of the double piston system only varies slightly between multiple operating points.

Potentials of the Free-Piston Linear Generator

As the FPLG has no crankshaft, the compression ratio is adjustable. An appropriate fuel supply system therefore would allow

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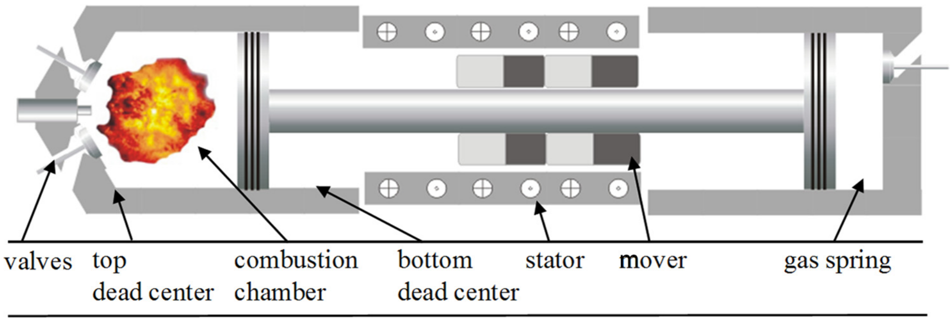


Fig. 1 Illustration of the principle of the FPLG

many types of fuel (petrol, diesel, natural gas, sun fuel, synthetic fuel, hydrogen, etc.). The fuels can be used without any constructional changes in the combustion unit, as the adaption of the compression ratio is design inherent. Thus, the combustion takes place at the maximum compression ratio of the fuel, allowing the highest possible efficiency. The system operates with petrol and peak pressures of 80 bar in spark plug ignition combustion mode and over 100 bar in homogeneous charge compression ignition combustion mode. Butanol is a very promising sun fuel and has been successfully tested in the FPLG. In Ref. [5,6], it is shown that butanol can be paired with homogeneous charge compression ignition combustion mode.

To achieve a full mass compensation, two synchronized piston units are needed. An advantage of the FPLG is the freedom of alignment. There are many ways to connect the FPLG subsystems (combustion unit, linear generator, gas spring, auxiliaries). The whole system can be built to fit under the trunk compartment or into a transmission tunnel. All possibilities only need a height of around 15 cm; therefore, integration in the floor of a vehicle is also an option. In the most recent estimates, the FPLG with a central opposed piston combustion unit (explained in the A Challenge and Solution of the Free-Piston Linear Generator section) and all auxiliaries can achieve a volumetric power density of up to 470 W/l at 50 Hz [7]. As shown in Ref. [8] the frequency is dependent on the moving mass. To achieve 50 Hz, the moving mass of the FPLG module has to be about 4–5 kg.

The noise–vibration–harshness behavior of the FPLG is aimed to be low. As described previously, each system consists of two modules. By synchronizing, these all free inertial mass forces are eliminated. Therefore, no vibrations are transferred to the vehicle body. Only the turbocharger and exhaust need standard soundproofing.

The measured break thermal efficiency is higher than 39%. The estimations based on simulations show that an opposed piston FPLG will reach a break thermal efficiency 43–47%. These high efficiencies are achieved over a wide power range, which is an important advantage of the FPLG over conventional four-stroke engines. Conventional engines only have a small operating range with high efficiency.

A Challenge and Solution of the Free-Piston Linear Generator

A FPLG module is explained in Fig. 1. A FPLG system for use as electrical power generator would consist of two of those modules, to allow for a full mass compensation. The system could look like Fig. 2(*a*). But the most advanced solution is the opposed piston combustion system shown in Fig. 2(*b*).

The opposed piston combustion unit is being preferred as FPLG system in the DLR. It has a simple structure with very few components. It has no cylinder head, and the cooling of which is difficult. This saves round about 40 cm (≈15.7 in) in length. No cam shaft or valves are needed for scavenging. The valves are replaced by ports opened by the pistons on their way to the bottom dead center. Therefore, the scavenging takes place while both ports are open. Additionally, the opposed piston combustion unit has a higher efficiency potential. The challenges of this system are the precise port timing, synchronization of the piston movement, and emissions to mention the most urgent ones.

The main challenge every two-stroke engine faces is the avoidance of unburnt hydrocarbons (UHCs). To prevent these emissions, the FPLG uses direct injection after all ports are closed. With a stoichiometric combustion, a standard three-way catalytic converter can be used. For this purpose, the short circuit flow has to be minimized. Otherwise, a NO_x storage catalytic converter is necessary. The second source of UHCs is motor oil. As shown in Ref. [9], motor oil is responsible for 5–30% of the UHCs in conventional engines. In the FPLG, the motor oil cools and lubricates the piston which is sliding over the ports letting the oil get into the ports, the combustion chamber, and exhaust pipe. This increases the UHCs even more. In order to use an opposed piston combustion unit, this source of UHCs has to be avoided.

One possible solution which is being developed by the DLR is the replacement of the motor oil by solid lubricants. This would prevent lubricants getting in touch with the combustion and exhaust. Solid lubricants can be installed in the piston, making a complex oil lubrication system obsolete.

Development Methodology

The beginning of the development started with solid lubricated compressors which are in use for quite some time. They use up to seven layers of piston rings made of Teflon materials. There are additional strains the lubrication system has to withstand in order to be used in a combustion chamber. These strains can be divided into three categories:

The combustion develops higher temperatures and pressure gradients than the compressors and their solid lubrication are designed for. To ensure the operation, the right material for the piston rings has to be chosen. This choice will have an impact on other strains and their solutions therefore it is a significant first step. There are many important properties the material has to fulfill in order to be used in a combustion chamber. The most important properties are solid lubrication ability, high bending strength to withstand the pressure, ductility not to break while sliding over the ports, and wear resistance and heat resistance. Every material is a compromise partially satisfying the needed requirements. One of the next steps will be to understand the wear and tear of the piston rings in the combustion unit.

The compressors usually do not have any ports to be opened by pistons or rather by piston rings. The valves are usually placed in the cylinder head. The “sliding over the ports” process is a very challenging issue, even for metal rings. Additionally, carbon materials show brittle behavior, which can be dangerous in this matter. While sliding over the ports, the rings can get stuck, break and damage the machine. To prevent this from happening, the geometry of the rings and ports has to be analyzed. While sliding the piston, ring slightly dips into the port. While emerging from the port, the ring gets abraded. This increases the wear and tear

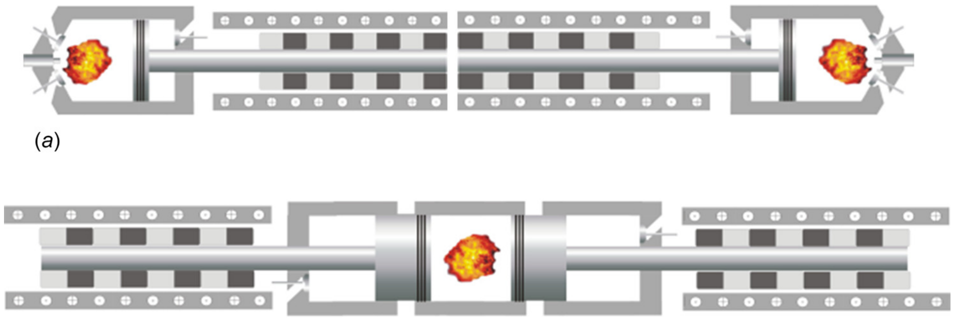


Fig. 2 FPLG systems: (*a*) dual module system and (*b*) opposed piston combustion system

Table 1 Material values of carbon material FT15HT from Schunk Carbon Technology

Density (g/cm^3)	2.00
Porosity (%)	4
Bending strength (MPa)	95
Compressive strength (MPa)	230
Young's modulus (GPa)	16
Coefficient of thermal expansion (10^{-6}/K)	7.4
Rockwell hardness HR 5/150	120

and in the worst case can lead to a ring fracture. In this matter, it is necessary to develop a geometry which prevents the abrading. To do this, it is important to know the strains acting on the ring. The Materials and Piston Ring Design section will show the development process in detail.

The third category is piston cooling. Beside lubrication, motor oil is responsible for cooling the piston. In an oil cooled piston, 30–70% of the piston heat is transported by the piston rings and the rest is absorbed by the oil. If the piston runs too hot, the engine starts to knock and the NO_x emissions increase. By removing the motor oil, a major part of the piston cooling is lost and has to be replaced in some way. Thereby, the most important issue is to separate the piston cooling system from the combustion chamber. A piston cooling system is under development and will be published in the future. The plan is to develop a passive cooling system. The goal is to keep the piston temperatures as high as or slightly higher than those of conventional pistons.

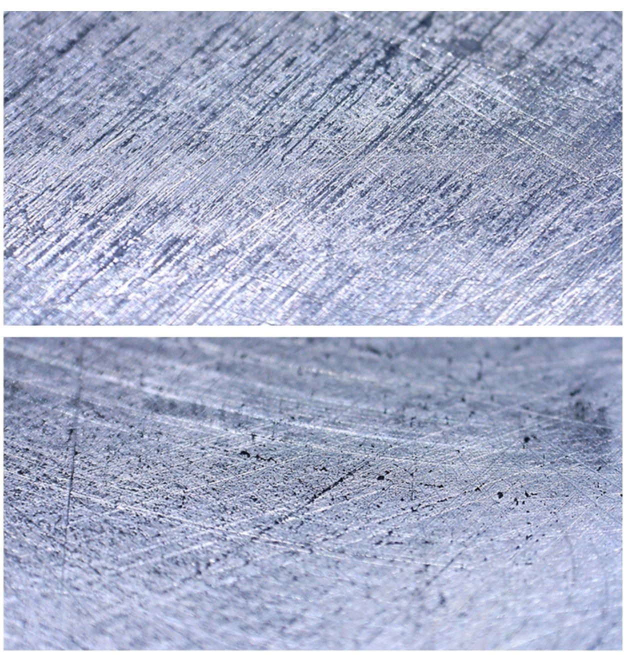


Fig. 3 Cylinder liner new (top) and after 6 h (bottom)

Materials and Piston Ring Design

In Ref. [10], a model for the contact between piston ring and cylinder is described. It shows the important role of the force distribution over the circumference in the developing process of ring design. In consideration of the operational temperature, bending strength and solid lubrication ability carbon materials seem to be the right decision. A few projects exist in which carbon materials were tested but without success in long-term operation [11].

For this project, the carbon material FT15HT from Schunk Carbon Technology was chosen. It consists of 95% carbon and 5% mineral ashes. The rings have a surface roughness of Rz = 2–3 μm. During the operation, the roughness gets even lower. The material has properties shown in Table 1.

To make the application of carbon piston rings easy, only minor changes should be applied to the combustion unit and especially the geometry of the cylinder liner should not change. So the cylinder liner remains the same except the honing. The cylinder liner was made of gray iron EN-GJL-300 (GG30) and has a surface roughness of Ra = 0.4–0.6 with cross-grinding as shown in Fig. 3 in the top picture.

The next step is to choose or develop a first ring design. As mentioned in the Development Methodology section, the starting point is solid lubricated compressors, so this is the source to adopt the first ring design from. A solution which is being developed by the Germany Aerospace center consists of carbon piston rings and carbon guide rings, displayed in the Fig. 4. Carbon has brittle behavior; therefore, the rings are divided into three segments so they can adapt better to the surface.

Fig. 4 Pressure ring, sealing ring, and a guide ring

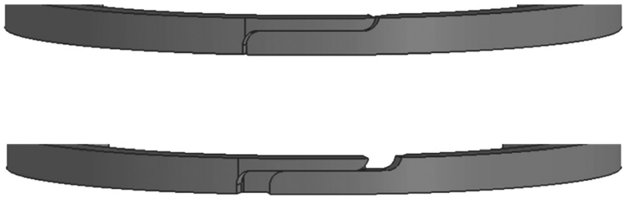


Fig. 5 Joint of a carbon sealing ring

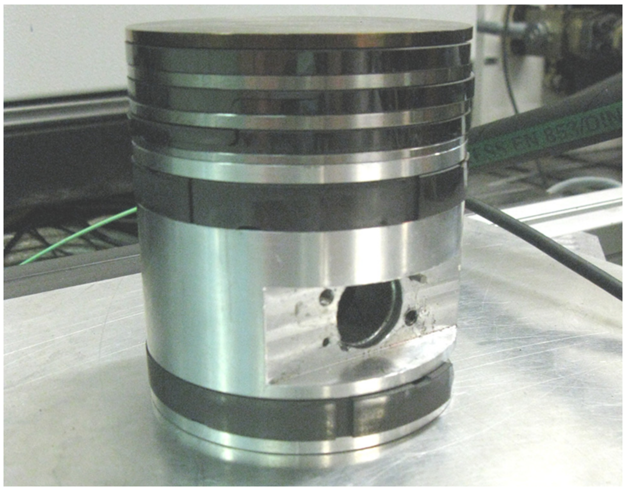


Fig. 6 Piston with carbon ring packet

The top ring in Fig. 4 is a pressure ring. The joint is simple; therefore, the rings are stable enough to withstand the peak pressure and pressure gradients in the combustion chamber. This ring absorbs the shock through the high-pressure gradients and protects the other rings.

The second ring is a so-called piston sealing ring. The complex joint is too fragile to withstand the high-pressure gradients. Its joint is complex and fragile to maximize the sealing ability as shown in Fig. 5.

In the top of Fig. 5, a new ring with a closed joint is illustrated; below, a partially worn out one with a more opened joint is shown for comparison. The joint is "more opened" but still sealed for the blowby. The ring is 5 mm thick to compensate the material wear.

The third ring is a guide ring. It keeps the piston equally spaced to the cylinder liner. It is designed for the maximum temperature to ensure that the thermal expansion of all materials do not make the piston stick. As it is not supposed to seal, it has axial notches.

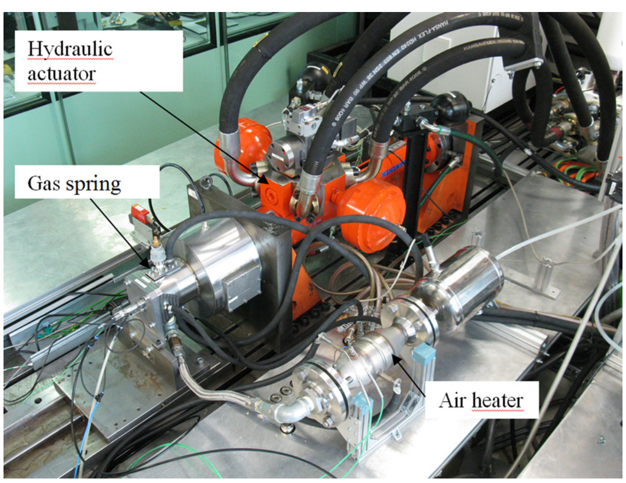


Fig. 7 Test bench for carbon piston rings

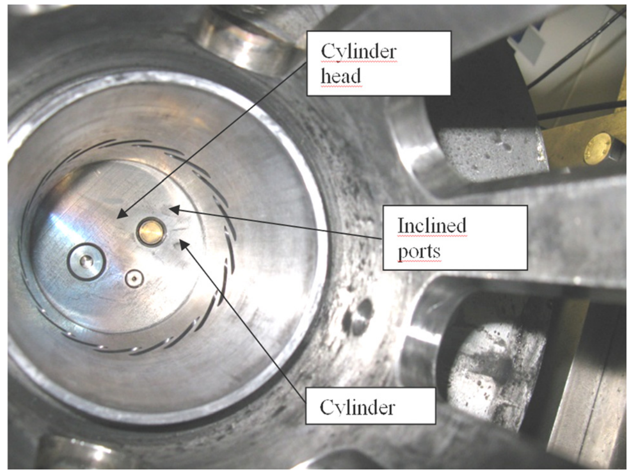


Fig. 8 Cylinder liner with ports in a test bench

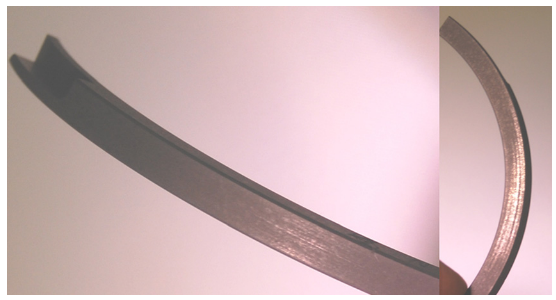


Fig. 9 New carbon ring segment

The whole ring packet consists of one pressure ring with simple joint, two sealing rings with complex joints and two guide rings which keep the piston in line. This ring packet installed in a piston is shown in the Fig. 6.

Test Bench

A test bench was specifically designed and built to test the mechanical properties of the rings. The goal is to test the ports and the lubrication system separately from the other strains like combustion. Therefore, the tests were performed in a gas spring mode. The test bench is shown in the Fig. 7.

The orange hydraulic actuator has a power of 600 kW and has a flow rate of up to 1000 l of hydraulic oil per minute. A gas spring unit (gray) is connected to the actuator. A pressure sensor and a temperature sensor are mounted in the cylinder head. Fresh air is scavenged through ports. The coolant temperature is 60 °C. In these tests, a compression ratio of 1:12 and a piston frequency of

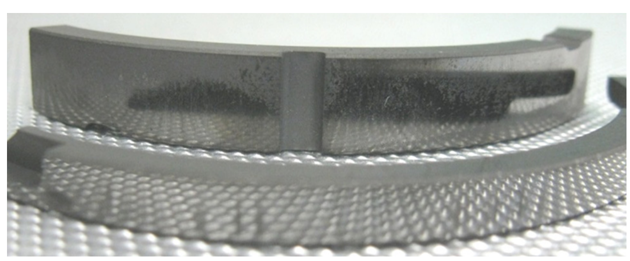
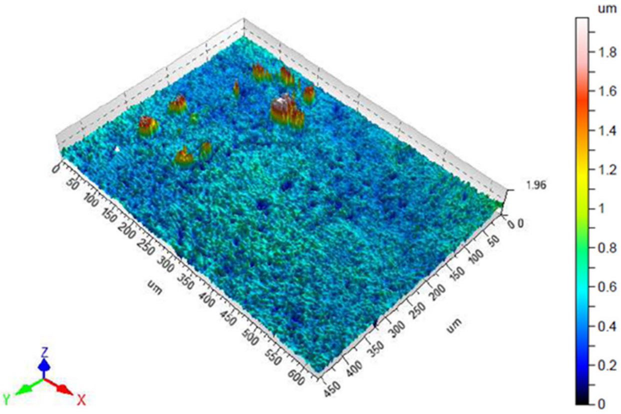


Fig. 10 Carbon ring segments after 6 h (from guide ring and pressure ring)



ISO 4287			
Amplituden-Parameter - Rauheitsprofil			
Rp	46.9	nm	Doppelter Gaussfilter, 0.08 mm
Rv	43.4	nm	Doppelter Gaussfilter, 0.08 mm
Rz	90.4	nm	Doppelter Gaussfilter, 0.08 mm
Rc	49.1	nm	Doppelter Gaussfilter, 0.08 mm
Rt	113	nm	Doppelter Gaussfilter, 0.08 mm
Ra	16.1	nm	Doppelter Gaussfilter, 0.08 mm
Rq	20.5	nm	Doppelter Gaussfilter, 0.08 mm
Rsk	0.484		Doppelter Gaussfilter, 0.08 mm
Rku	2.95		Doppelter Gaussfilter, 0.08 mm

Fig. 11 Piston ring measured by confocal microscope

20 Hz are used. In Fig. 8, a cylinder liner with ports is shown, which is used on the test bench. Also, the cylinder head with the sensors is visible. Normally the piston rings need an anti-twist protection so their joints do not dip in while sliding over the ports. To keep the piston rings simple, the ports are inclined, making the anti-twist protection unnecessary.

Results

The test runs were performed to check how the piston rings behave around the ports. Each run lasted for about 6 h and was performed with 20 Hz, up to 70 bar peak pressure and with a stroke of 65 mm. The temperature sensor in the cylinder head measured temperatures about 210 °C. In this configuration, blowby measurements are not possible. But the blowby is expected to be higher than that of conventional oil lubricated piston rings. After each test, the test bench was disassembled and the piston rings examined. In Fig. 3, a new cylinder liner surface is shown in the top picture. In the bottom picture, grooves from the cross grinding are filled with graphite particle providing a lubrication reservoir.

Figure 9 shows a new carbon ring before it is set into the test bench. The color of the new ring is mat. This allows a simple check to find out if the ring works properly. When a carbon surface is being worn, it changes from mat to shiny/polished. After the rings are broke in during a test run, the surface of the rings look polished as the Fig. 10 shows.

In the figure, a segment of a guide ring and of a sealing ring is shown. The whole surface of the segment is shiny, which means the whole ring worked properly and broke in after less than 6 h. The confocal microscope confirms the good surface properties like low roughness of Rz = 0.09 μm (Fig. 11). This low roughness means that the piston rings slide over a graphite lubrication layer and the grooves works like a lubrication reservoir.

However, not everything worked perfect in the test runs. 30% of the rings were broken or chipped. After examining the rings, couple of problems were detected.

In Fig. 12, an example of a broken ring packet is shown. The failure after 6 h has mechanical sources, which will be examined

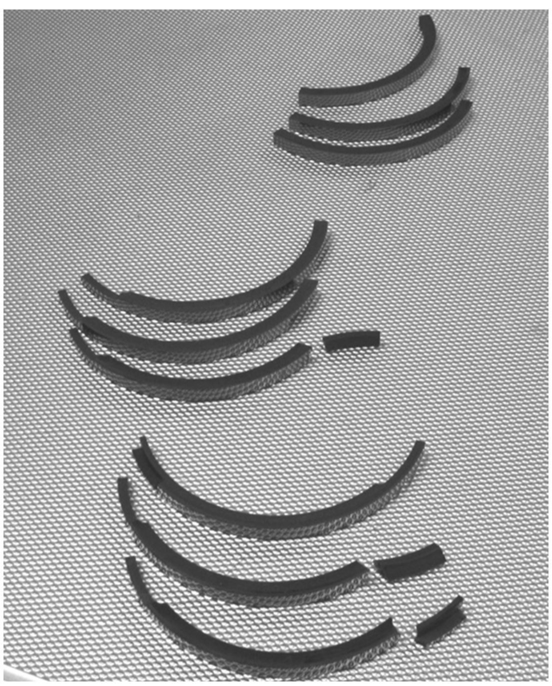


Fig. 12 Broken ring packet

Fig. 13 Broken sealing ring

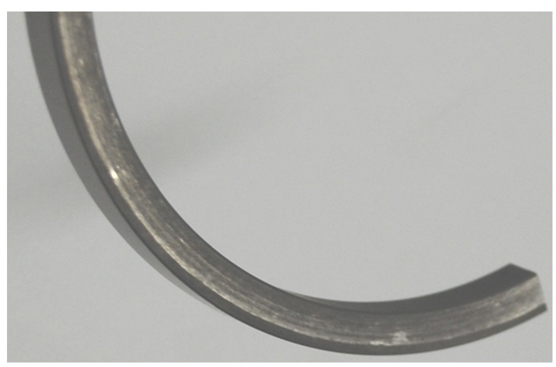


Fig. 14 Ring contact surface

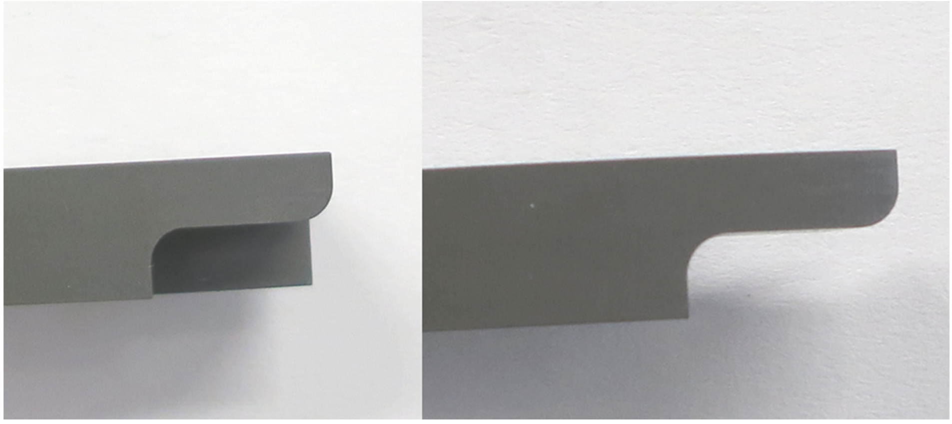


Fig. 15 Second generation of piston rings

Fig. 16 Piston rings after the test runs

in this section. Most frequently, segments of the middle ring, the first sealing ring, broke. The broken parts are at the fragile joints of the sealing rings. This ring has a male and female part of the joint. There are breaks on both joint parts, but most breaks happen on the thicker female side as shown in Fig. 13 and are caused by the air pressure.

The granular surface of the break is shiny, which confirms the brittle behavior of carbon. There are two different surface forms in the right photos (Fig. 13). The one in the upper circle originated from a direct impact, and the lower one emerged from the following break.

In Fig. 14, the top surface of the pressure ring consists of a matt black outer part and more or less shiny inner part. The matt black outer part of the surface has no contact to the piston ring notch. The inner part looks more or less shiny/bright, that means it has regular contact to the piston notch.

The guide rings did not experience any damage even though they were in contact with the ports as much as the sealing rings.

The examinations lead to following reasons for the malfunction. The guide rings have a defined position, so the amount of shocks through vibrations while operating is small. The piston rings in comparison have multiple degrees-of-freedom to adjust their position. Through change of the contact surface, they experience a series of shocks. To seal and stay in a defined position, the pressure and sealing rings need a differential pressure. Around the ports, there is no pressure difference; therefore, shocks through change of the contact surface occur. The position of the rings around the ports is undefined and they can slightly turn and dip into the ports. This can lead to an immediate breaking of a ring segment.

Fig. 17 Piston ring surface

To prevent these breaks, changes in design were made to the piston rings as shown in the Fig. 15. The geometry of the joints was adjusted. The joints are shorter and the radiuses are larger, so they cannot dip into ports as easy as before. Additionally, the horizontal part of the female joint was designed a little thicker, because this was the place with the most damages caused by the air pressure. As final adjustment, a spring provides for a defined position of the rings. They are pressed against the wall and therefore experience fewer shocks than before. The spring is positioned on the inner side of the piston rings in a notch, so the wall thickness of the new piston ring had to be increased.

The test runs were mostly successful, and none of the piston rings broke. As shown in Fig. 16, the surface looks shiny, which means the piston rings are broke in.

The results show that the piston rings perform better than the first generation. However, the edges of the ring and the corresponding surfaces shown in Fig. 17 point toward a new problem. Pieces of the ring edge are broke out and scratch the corresponding surface of the ring. The work on the solution is in progress.

Conclusion

The concept of FPLG was explained. It consists of three subsystems linear generator, gas spring, and combustion section. The characteristic properties of the FPLG were presented together with one of the development challenges. In order to use an opposed combustion chamber, the emission issue has to be solved. As an answer, one possible solution, solid lubrication, is presented. In this paper, the mechanical stresses of the piston rings were closely looked at. The first generation of the rings had some weaknesses, which could to be eliminated. The solid lubrication should be able to withstand the combustion stress after solving the challenges mentioned in the Development Methodology section.

The FPLG has the potential to be a key component in future drivetrain concepts. Several advantages including improved efficiency are possible. Today's measurement results indicate that the realization of the FPLG within a car may become possible.

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