Combined experimental and simulative approach for friction loss optimization of DLC coated piston rings

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Abstract
Piston rings cause significant friction losses within internal combustion engines. Especially the first compression ring, which is pressed onto the liner by high cylinder pressure, contributes significantly to the total friction loss of the piston assembly. The tribological behavior of the oil scraper ring is mainly related to the pretensioning force and can lead to high losses even at low and idle speed. Due to this, there is always a markable risk of wear for the contact surfaces of the piston rings and the cylinder. “Diamond-like carbon” coatings on the surface of the piston rings can prevent wear and are able to reduce friction in the ring-liner-contact. The purpose of this work was to investigate the tribological benefit of this coating-system on the compression and oil scraper ring. Experimental studies were carried out on a fired single-cylinder engine using the Indicated Instantaneous Mean Effective Pressure-method (IIMEP) for the crank angle-resolved detection of the piston assembly’s friction force. To be able to determine the component-related fractions of the friction loss and to quantify the hydrodynamic and asperity related parts locally and time dependent, an EHD/MBS model of the engine was created in AVL EXCITE and a simulative investigation was performed. This simulation was validated by the experimental work and provided detailed information about the individual contact conditions and gap height of each tribological contact of the piston group. The combined approach of measurement and simulation enabled the prediction of tribological aspects and performance in parameter studies on a virtual engine test bed.

Keywords  CO₂ reduction · DLC · Friction · Piston ring

Abbreviations

| Abbreviation | Description |
|--------------|-------------|
| a-C:H        | Hydrogen containing amorphous carbon (DLC-layer) |
| BTC/TDC/CTDC| Piston bottom/top dead center/combustion top dead center |
| DLC         | Diamond-like carbon |
| EHD/MBS     | Elasto-hydrodynamics-multi body simulation |
| FMEP        | Friction mean effective pressure |
| ICE         | Internal combustion engine |
| IMEP        | Indicated mean effective pressure |
| IIMEP       | Instantaneous indicated mean effective pressure |
| n           | Engine speed |
| NVH         | Noise vibration harshness |
| ta-C        | Hydrogen-free tetrahedral amorphous carbon (DLC-layer) |
| WLTC        | Worldwide harmonized light duty test cycle |

1 Introduction

Due to strict legislation limits, the reduction of CO₂-emissions is one of the main goal for the development of new engine concepts. Increasing the efficiency by reduction of the friction losses is one of the promising approaches for improving internal combustion engines (ICE), keeping in mind that approximately one third of the fuel energy is linked to friction in a conventional passenger car [1]. The piston assembly (piston, rings, pin) causes approximately half of the frictional power loss of an ICE [2], where the piston rings are responsible for the major part. This is a result of a high share of mixed friction in the contact area between the piston ring and the cylinder, which especially occurs in
the combination of high engine load, low rotational speeds and high temperatures of the coolant. The trend toward lower lubricant viscosity leads to further increase of the share of mixed friction between ring and cylinder. Depending on the application, specific coating systems are used to prevent the wear of ring surfaces. The latter is in particular necessary for the first ring and the oil scraper. For the first compression ring high cylinder peak pressure and for the oil scraper the high pretension force result in high solid–solid contact pressure.

Diamond-like carbon (DLC) coatings are known for their tribologically beneficial behavior with high wear resistance and low adhesion friction even under poor lubrication conditions. First automotive applications started using hydrogen containing amorphous (a-C:H) DLC-layer in the early 2000, but some were not suitable for use with molybdenum friction modifier additives [3]. Nowadays, DLC coated engine-components of the valve-train, injection system, bearing and piston assembly are Hydrogen-free. These tetrahedral-amorphous carbon (ta-C)-DLC layers, characterized by a high fraction of diamond-bonded carbon, show outstanding tribological properties: high hardness, low friction coefficient as well as high chemical and thermal stability. These layers are suitable for operation under harsh conditions, e.g. on piston rings, and are able to combine wear protection with friction reduction [4]. Furthermore it is assumed, that the higher fraction of diamond-bonds leads to greater hardness and frictional advantages [5]. Piston rings with DLC coatings are available for series engine production [6]. However, there is still some potential to increase hardness.

The aim of this work was to investigate the tribological behavior of taC coatings on piston rings. For that purpose, two different DLC-coatings were deposited on compression and oil-scraper rings from a state-of-the-art gasoline-engine. Experimental and simulative investigations were carried out, focusing the frictional performance.

2 Single cylinder test engine

The tests were performed on a fired single-cylinder engine. Table 1 shows the technical data. The test bed engine is based on a state-of-the-art gasoline engine with direct injection, variable valve timing and turbocharging. To meet the particular requirements for investigations of the piston assembly—cylinder contact, an individually designed engine block, crankcase and crankshaft were used. The split design of the engine block allows the replaceable cylinder liner to be changed quickly and the piston assembly to be removed without disassembling the cylinder head. The latter, including valve train, ignition and injection system, was taken from the series engine. Figure 1 shows the engine and design details of the block.

| Table 1 | Technical data of the single cylinder engine testbench |
|---------|-------------------------------------------------------|
| Single cylinder engine testbench | |
| Engine type | Gasoline-Turbo-DI |
| Valve train variability | Inlet—Timing & Lift |
| Displacement | 0.5 l |
| Compr. ratio | 11:1 |
| Bore | 82 mm |
| Stroke | 94.6 |
| Max. cylinder pressure | 130 bar |
| Max. engine speed | 4000 min⁻¹ |
| Max. temp. coolant | 120 °C |

Determining the friction between piston assembly and cylinder is particularly instructive using crank angle-resolved methods, which is not possible with standard measurement technology and requires individual solutions. The Indicated Instantaneous Mean Effective Pressure (IIMEP)—method was first described in [7] and is an indirect technique for analyzing piston assembly friction. Due to only slight modifications on the engine, this method leaves the engine operation unaffected, which provides advantages in terms of the realistic thermal and mechanical behavior. In addition, a large range of the operating map can be used and a variation of cylinder surfaces is easier compared to direct measuring systems, like “floating liner”.

The measuring principle is based on a force equilibrium on the piston pin and considers all forces acting in the direction of piston motion (Fig. 2). The determination of exact gas and inertia forces requires a profound knowledge of this particular measurement methodology. The sensor devices itself are commercially available. The same applies to the conrod force, where the measurement is done by strain gages. Here the challenges are the data processing as well as the energy and data coupling to/from the conrod under harsh conditions in the crankcase. To minimize interferences, a system with full digital and wireless processing was developed. All required signals are evaluated in one single master indication system, where the piston assembly friction force is calculated by summing up the gas, inertia and conrod force. During test bed operation, all engine and friction-related signals and values can be displayed instantaneously.

Due to the high sensitivity of the method to some engine process parameters, in particular the magnitude and position of the maximum firing pressure, a highly automated test procedure ensures reproducible operating conditions, e.g. for temperatures and pressures of all supply media. This includes a self-developed combustion management system, which is able to control selected combustion parameters like the indicated mean effective pressure (IMEP), the maximum cylinder pressure and the center
of combustion by independently scalable adjustment of e.g. the air/fuel mass flow and the ignition system. The engine process control (valve timing, injection, ignition, lambda, etc.) is based on the series application. An automated check of very small tolerance ranges ensures, that the measured data for every single benchmark-setup of the engine was acquired under reproducible conditions on the test bench. In addition to the test procedure, a great accuracy is needed for the preparation of the test rig as well as the used components before measuring. This includes, among various aspects, a specific running-in methodology for each engine-configuration as well as an oil change procedure leaving no residual oil in the engine.

The measurement of friction losses was performed at stationary operating points, where the results for each single point are derived from 600 cycles. The tests in this study were carried out at a coolant and oil temperature of 90 °C on the engine inlet. The piston cooling nozzle was deactivated for loads below an IMEP of 10 bar. For higher loads it was operated on a fixed flow rate.

For data evaluation and tribological comparison between different engine setups, crank angle-resolved friction curves (Friction force/work/power) allow for the identification of position-dependent effects in single operation points. The sample rate can be set up to 0.1° crank angle for highly detailed studies. Cumulated friction parameters, such as
the friction mean effective pressure (FMEP), are needed to quantify effects for one work cycle and to give an overview of the overall behavior in the engine map. Assuming a proper selection of operation points and the knowledge of their time shares in a load collective, one can even give relative predictions of the impact under dynamic conditions, e.g. the fuel consumption effect in the worldwide harmonized light duty test cycle (WLTC).

3 Piston assembly friction simulation

For the investigation of the detailed local friction in the tribological contacts of the piston assembly for the fired single cylinder engine, the numerical analysis of the piston and piston ring friction is performed with AVL excite power unit and AVL excite piston & rings (Fig. 3). These simulation tools are combined to investigate the friction in the fired single cylinder engine and are the industrial standard for the investigation and optimization regarding engine friction, oil consumption, and noise vibration and harshness (NVH) in the engine development process. In the simulation model, the real single cylinder engine is modeled with a detailed multibody simulation (MBS) with flexible bodies which consist of condensed finite element structures and elasto-hydrodynamic (EHD) joints for the tribological contacts. The MBS/EHD simulation model for the calculation of the piston cylinder liner friction consists of the nonlinear tribological contacts for the piston cylinder liner contact, piston pin bearings, conrod small and big end bearing as well as flexible bodies. The latter consist of condensed finite element structures for the piston, the crankcase including the cylinder liner, the piston pin and the conrod. The piston secondary motion of the piston dynamic analysis is considered in the piston ring dynamics calculation model which considers the piston, cylinder liner and three-dimensional piston rings. The simulation models consider the linear profile shape due to manufacturing, thermomechanical deformation and the piston profile due to manufacturing and thermal deformation as well as the piston ring profiles and temperature dependent lube-oil properties. In the EHD contacts, the fluid film as well as the asperity contacts and their material properties are fully considered via precalculated pressure and shear flow factors and an asperity pressure map, contact stiffness and contact ratio [8, 9]. In this context asperity means the real micro structure on the surface of any tribological partner, which can be determined with good accuracy by measurement. These are calculated in a preprocessing step with AVL Microslide using the real measured three-dimensional rough surfaces in running-in conditions. For

Fig. 3 Simulation model and tools for piston and piston ring dynamic tribological contact analysis

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the friction calculation in the EHD contacts a mixed friction model [10, 11] is used which considers boundary and viscous friction. This model was calibrated for the different piston ring cylinder liner contacts and their operating conditions by a combined measurement and simulation approach [12]. For this, a reciprocating long-stroke tribometer test bench is simulated for different tribological pairs and compared against corresponding measurements. This yields validated friction models for the fired single cylinder engine simulation, which is part of this work.

The simulation models, which consider all relevant physical effects, are validated by the measured piston assembly friction. The simulation correlated well with the measurements of the fired single cylinder engine in most areas, only during combustion some deviations are visible (Fig. 4). On the experimental side, this can be explained by the high measured force amounts on which the calculation of the friction force is based on at this time. Furthermore, the realistic representation of the thermal liner distortion in the simulation is very complex, which complicates the already challenging representation of the contact processes during the piston’s free flight. In the simulation model one major advantage is that the local friction can be analyzed for each piston ring and the piston separately for each operating point (Fig. 6). Also, in the virtual engine model the contribution of hydrodynamic and solid asperity contact friction can be analyzed in full detail. This offers the possibility to investigate and evaluate the component related and overall friction for different engine setups with different piston ring coating technologies. For this, the material properties, surface roughness and profile of the investigated piston ring were varied, while the piston, the second compression ring, cylinder liner, lube oil and engine crank train geometry were kept unchanged.

4 Use case: DLC coated piston rings

The potential of DLC coatings for the reduction of friction losses was investigated on the previously specified single cylinder engine. For that purpose, two different hydrogen-free ta-C-coatings were used in a first step on the compression ring, afterward the study focused on the effect of the oil scraper ring coated with the hard ta-C-layer. The ta-C layers were deposited by filtered laser-arc technology [13], Table 2 shows the engine configuration of the measurement campaign.

The distortion of the cylinder bore has a major influence on the piston ring behavior and is affected by the tensioning of the cylinder head. To ensure a high comparability of the experimental results within a measurement campaign, the engine setup should be changed as little as possible when converting to another test configuration. For this reason, both the liner and the piston were retained for the entire study. The change of piston rings for each test setup was done by dismounting the engine block from the crankcase, while the assembly conditions of the cylinder head remained untouched.

The running-in procedure of new components deserves special attention to achieve tribologically stable conditions
and to avoid running-in effects during the measurement run. Therefore, each test configuration had to complete an identical conditioning program prior to the measurements. Retaining the liner for the whole campaign means a compromise between repeatability and realistic run-in behavior. However, the test sequence, is adapted to the hardness of the piston ring surfaces. Softer coatings are employed first, harder ones later. This allows for all test configuration to run in on the same liner (Table 2).

The run-in program was always performed in the same order by passing through the engine speeds (1000–3000 rpm) of each load step, starting with motored points and finishing at full load at IMEP of 15 bars. Then the measurement program was performed. After this a reference point was measured. If this verification was successful, the engine was set up for the following test configuration. After dismounting, the component surfaces were measured optically and by tactile methods.

4.1 Topring results

The first part of the campaign was focused on the effect of ta-C layers on the toprings, which are mainly stressed by the gas pressure on the piston during compression and combustion. Figure 5 shows the crank angle-resolved friction of the piston assembly for the topring-variation based on two selected operation points. A friction reduction was found for the ta-C coated topring with varying level and distribution during the work cycle. A reproducible advantage is obvious in every combustion stroke, which can be explained as follows: ta-C coatings have an advantage over conventionally treated ring in the realm of contact friction, i.e. not in hydrodynamic friction. If the ring contact pressure increases during combustion, in particular around TDC where sliding speeds are low (or zero), ta-C layers can reveal their full advantage in friction.

The friction analysis for measurement and simulation shows clearly an advantage for ta-C coated piston rings. In Fig. 6, the simulated component-related friction shows that the topring friction is reduced by the ta-C coatings. The major friction reduction potential is in the combustion cycle, where a high amount of friction with high asperity contact occurs, and this is reduced by ta-C coated piston rings because of their lower friction coefficient. Due to its higher hardness, nearly no wear occurs on the ta-C coated rings which leads to a different tribological behavior because of a sharper piston ring contour and a different surface roughness in running-in condition compared to the reference. So a larger amount of oil is scraped off from the liner wall and that reduces its hydrodynamic friction contribution compared to the reference. The sharper top ring contour reduces on one hand the amount of hydrodynamic friction but increases on the other hand the asperity contact pressure slightly. In general the asperity friction loss is influenced by

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Fig. 5 Effect of ta-C topring coatings on the piston assembly friction at two operation points
asperity pressure as well as the friction coefficient. So the increase of asperity contact pressure for ta-C coated rings due to sharper ring contours in running in condition diminishes the effect of the decrease in friction coefficient slightly. But the resulting asperity friction power loss for ta-C coated top rings is reduced because the friction coefficient is ways smaller than the increase in asperity pressure due to the sharper ring contour. All in all the resulting sharper ring contour of ta-C coated piston rings is beneficial in terms of friction because it significantly reduces the amount of hydrodynamic friction so that a slight increase of the asperity contact pressure is acceptable.

The ta-C soft coating shows lower asperity friction compared to the ta-C hard coating since its contour is more smoothened due to its lower hardness and also its friction coefficient is lower. Moreover, with the simulative analysis a high potential for further friction reduction could be identified for the relevant operation points: the oil scraper ring is a major contributor to the piston assembly friction due to its higher pretension force compared to the top ring (Table 2). So a ta-C coating would be beneficial to reduce the oil scraper ring friction which is also part of this work.

The piston assembly friction analysis for measurement and simulation shows for the investigated operating points a clear advantage for ta-C coated piston rings (Fig. 6, Fig. 7 and Fig. 8). With the detailed simulation results shown in Fig. 7, friction reduction potentials are identified for the different operating conditions. With higher specific loads the friction reduction at the topring due to ta-C coatings increases as a consequence of increasing asperity friction. For the oil scraper ring the friction contribution is high for the investigated operating points. With higher engine speeds its asperity contact friction decreases and the hydrodynamic friction increases. The second ring contributes less to the overall friction because of its low pretensioning force and the greatly reduced gas pressure behind the compression ring. So its friction reduction potential is in general negligible. Furthermore the topring friction is very load dependent because of high gas pressure induced forces and the very thin oil film. Therefore the asperity friction during the combustion cycle shows a good potential for friction reduction by ta-C coatings. The piston contributes more to the overall piston assembly friction with increasing engine speeds, due to its large surface and the resulting high amount of hydrodynamic friction, which is caused by higher shear stress in the lube oil and the differences in clearance.

Figure 8 shows the effect of the ta-C layers using FMEP-Differences relating to the reference configuration with steel-nitrided rings. It is shown, that the ta-C rings lead to benefits for the friction reduction in general. However, differences between the layer types are visible in terms of the magnitude and position of the advantageous areas in the engine map: while the soft layer performs best with engine speeds below 1500 rpm and part load, the harder coating seems to work better with higher engine speeds and loads.

Based on the WLTC time shares of the stationary operation points for a common engine-vehicle combination, one can assume fuel savings of 0.6% for the hard ta-C coating, respectively 1% for the soft ta-C layer.
4.2 Oil scraper results

After having found encouraging results for the ta-C topings, an oil scraper coated with the hard ta-C layer was investigated on the test bench. However, as shown in Fig. 9, no advantage of the ta-C layer for the oil rings was established across the engine map. Moreover, the crank angle-resolved results show a slight increase of the friction loss for nearly the entire work cycle. In particular the ranges with an upward stroke of the piston show a higher frictional force. With raised load, an additional increase of friction is found in the combustion stroke for ta-C, which is an untypical oil ring behavior and leads to the assumption that lubrication conditions have changed for the compression ring, respectively the piston assembly. This was confirmed by the analysis of the oil ring contour, which showed no wear or running-in due to the hard coating.

5 Predictive friction loss optimisation

In this simulative investigation, the three piece oil scraper ring material was changed to a “soft” ta-C coating by considering its material properties, surface roughness and friction behavior. Because of the negligible wear on the oil scraper ring rails observed in test bench results, which was obviously caused by the higher hardness of the ta-C coating, the initial ring profiles of the rails did not wear and that led apparently to a missing adaptability of the oil scraper ring to the tribological system. That caused a sharper ring profile and a higher surface pressure distribution which resulted in the measured non advantageous friction results as described in chapter 4.

Therefore, to investigate the friction potential of ta-C coated oil scraper rings, the simulation model assumed an...
oil ring contour, comparable to the nitride reference oil scraper ring contour in running-in condition. For these adapted conditions, the simulative friction analysis clearly showed the most advantageous results for the “soft” ta-C coated topring in combination with a “soft” ta-C coated oil scraper ring (Fig. 10).

This is a consequence of the reduction of asperity friction as a result of the “soft” ta-C coating on both rings as well as a positive impact on the oil ring’s hydrodynamic friction due to the adaptation of the contour. However, it can also be seen that because of this change there is a slight negative influence over the ring pack on the topring lubrication and

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**Fig. 9** Effect of ta-C oil ring coatings on the piston assembly friction

**Fig. 10** Simulated friction power loss: topring- and oil scraper ring-coatings in relation to the reference configuration
subsequently hydrodynamic friction compared to the variant with “soft” ta-C coating only on the topring. The overall friction across the piston assembly nevertheless is reduced since the friction reduction on the oil scraper ring is more dominant. This is, because the oil scraper ring has a high pretension force and therefore contributes highly to the total piston assembly friction.

In the simulation analysis of different operating points shown in Fig. 11, friction reduction potentials are identified for the shown ta-C coated variants. The potential in friction reduction of ta-C coatings on the oil scraper ring and the topring is increasing with decreasing speeds and increasing loads. The investigated reduction of asperity as well as hydrodynamic friction shows a good potential for improving the mechanical efficiency by ta-C coatings for piston ring applications. One precondition for the transfer of the simulation results to the engine test is, that it will be possible to reach an optimal ta-C layer adhesion on the oil scraper ring. Another prerequisite is, that the ring profiles need to be manufactured corresponding to the run-in shape of an comparable nitride ring, to fit perfectly into the tribological system.

Thanks to their properties, DLC coatings are versatilely applicable. Especially the ta-C type with its high hardness and low friction coefficient can help to reduce CO₂ emissions in engine design. The presented work shows the frictional effect of two different ta-C coatings (“ta-C soft” / “ta-C hard”), which were applied on series compression- and oil scraper rings of a four-stroke gasoline piston engine. A combined approach of experiment and simulation was chosen for this investigation. Tests were carried out on a fired single-cylinder engine using the IIMEP-method for analyzing the crank angle-resolved friction loss of the piston assembly. Furthermore, an EHD/MBS model of the engine was built in AVL Excite, which gives detailed information about component-specific friction components, individual contact conditions and also enables the prediction of tribological aspects.

### 6 Summary

The study shows significant advantages with ta-C coated compressions rings of up to 14% less friction of the piston assembly compared to the series engine configuration (steel-nitrided compressions ring). Beneficial map ranges differ depending on the used coating, resulting in estimated fuel savings of 0.6% for the hard and 1% for the soft ta-C coating, based on the WLTC. The analysis of the crank angle friction signals shows an advantageous behavior of the ta-C compression rings especially for the compression and combustion cycle. However, the hard taC coating of the series oil scraper rings showed some disadvantages. This seems to be a result of the missing running-in of the oil ring contour due to the hard coating. Furthermore, this leads to a modified lubricant supply to the whole piston assembly with

![Fig. 11 Simulated component-related friction loss: topring and oil-scraper ring-coatings in relation to the reference configuration for different operation](image)
increased hydrodynamic friction in almost all crank angle ranges. The simulation showed however, that an adjustment of the oil scraper contour could possibly optimize the lubricant supply on the cylinder wall for a friction reduction on the oil as well as the compression rings.

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**Declarations**

**Conflict of interest** The corresponding author states that there is no conflict of interest.

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