

Friction Analysis of Oil Control Rings during Running-In

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ABSTRACT

This SAE Technical Paper will give a summary of the detailed results of crank-angle-resolved friction measurements which were carried out in motored and pressureless operation on a floating liner single cylinder test engine with a displacement of 0.5 liters. Friction forces that occur during running-in are introduced and discussed with regard to mixed friction and hydrodynamic friction for different oil control rings. Three different ring pre-loads are examined using constant geometries. In a second test series, the outer contour and coating is varied while the pre-load stays constant.

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INTRODUCTION

Due to current CO₂ emission laws the automobile industry in particular has come under pressure to reduce the consumption of their vehicles. This can be achieved most effectively by reducing the losses at the combustion engine. In addition to thermodynamic losses, mechanical losses play a decisive role in this matter. The largest part of mechanical losses is caused by the piston assembly, thus offering the biggest potential for reducing fuel consumption through tailored further development. As a complex tension field is active in the range of the piston assembly, the research requires a holistic approach. In this stress field, target parameters such as friction, oil consumption, blow-by and wear are to be considered. This paper gives a detailed overview of the crank-angle-resolved friction forces that are active in the piston assembly over the course of the running-in. This way an interconnection between friction and wear is achieved. The oil control ring is exemplary for the elements of the piston assembly, due to the fact that it bears the largest part of the friction losses at the piston rings. During the tests the ring pre-load varies, while constant geometries are used. In a second test series, the outer contour and coating are altered while the pre-load is maintained constant.

TEST CARRIER

A single cylinder test engine equipped with a direct friction force measuring device is used. The engine is based on a passenger car SI engine and was set up specifically for piston assembly friction measurements. The engine specifications are 82.5 mm bore and a 92.8 mm stroke,

adding up to a displacement of 0.5 liters. Maximum speed is 7000 rpm.

In order to facilitate near-series results the crank drive was derived from the basic engine, specifically the con-rod and the piston assembly. The connecting rod ratio is 0.32 [1]. For higher quality of measurement, the engine is exclusively run in motored and pressureless operation in the presented tests. Examinations were carried out at engine speeds of 680 and 1500 rpm on the engine dynamometer. The coolant and oil is supplied via an external preconditioning unit. Temperatures and pressure can be regulated. During the tests the engine is constantly operated at 90 °C oil and water temperature and 1.8 bar oil pressure. For near-series results the original piston oil cooling nozzle is used, which is permanently active. As vibrations of the basic engine must be avoided during friction force tests, a first and second order Lancaster balance shaft is employed.

The test engine operates in accordance with the floating liner method which was developed by Furuhashi [2]. Figure 1 shows a drawing of the applied measuring principle. The cylinder liner unit features an integrated water jacket. It is mounted on two piezoelectric Kistler 9061 load washers [5]. Both measuring boxes are in-stalled at the crank axis. As this layout is only equipped with two measuring boxes the piston side forces have to be held via radial mountings. One of each is installed at the pressure side and the counter-pressure side.

A Kistler 5064 unit is used as a charge amplifier. For three seconds, the applied measuring board Spectrum M2i. 4652 records friction signals and trigger signals at a sample rate of up to 3MS/s/Ch. According to the Nyquist-Shannon sampling theorem, see [9], this approach yields a sufficient number of recorded measuring points for the detailed

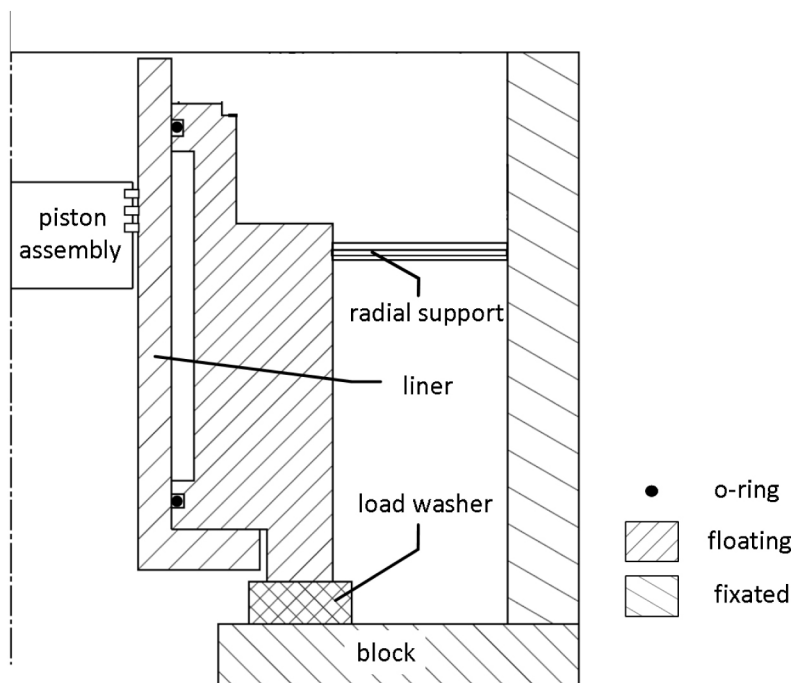


Figure 1. Schematic diagram of the measuring system

recording of mixed friction peaks at high engine speeds. The processing routine is implemented in the commercial MATLAB Software. With this method, three measured seconds are averaged for a representative combustion cycle. Inherent to its functional principle the measured friction is super-imposed by the test setup's natural frequency. For an optimized graphic presentation the data are filtered of natural frequency by band elimination.

This measuring system can also be applied under fired conditions. One of the essential features of measuring systems is the achievable reproducibility of the results. In comparison to piston and its first compression ring, the combustion chamber pressure only plays a secondary role in the friction levels of the oil control ring. The main influencing factors on the oil control ring friction, besides the pre-load factor and its geometry, are temperatures of the oil and the cylinder liner. Those are freely adjustable even without being in fired operation. With pressureless operation (the cylinder head is removed), reproducibility is improved. This is why examinations in the area of the oil control ring are carried out in pressureless and motored operation for increased reproducibility. To determine the reproducibility the examinations have to be divided into two categories. First, two tests are done without removing the piston assembly. With this, an average repeatability of 1.5 N is achieved. The second relevant category for the measures at hand is the repeatability including a change of the piston assembly. In this case, an average repeatability value between 3 and 5 N is achieved. Regarding the change of the friction level over the duration of the running-in of a fixed configuration, an improved average repeatability of 1.5 N

must be considered in the analysis and interpretation, as the piston assembly is not changed during the tests. If different configurations are to be compared, a higher average value of 3 to 5 N applies.

CHARACTERISTIC POINTS OF A FRICTION FORCE CURVE

Comparing friction force curves is a complex task. There is a smooth transition of the individual phases, from mixed friction to hydrodynamic friction, and their categorization is difficult without oil-film measurement. This is why, within the scope of the scientific research at the institute of combustion engines, a system to determine the characteristic values within the friction progression was developed. Mixed friction maxima in the area of the top (TDC) and bottom (BDC) dead center are examined. At TDC and BDC, mixed friction maxima represent part of the wear and with its part of the running-in. As hydrodynamics significantly influence the mean friction pressure, the latter is added to the observation parameters.

Figure 2 shows the friction measured in pressureless operation. Points, which were examined in systematic tests, are indicated.

In the analysis, characteristic points in TDC and BDC are simply considered as differences.

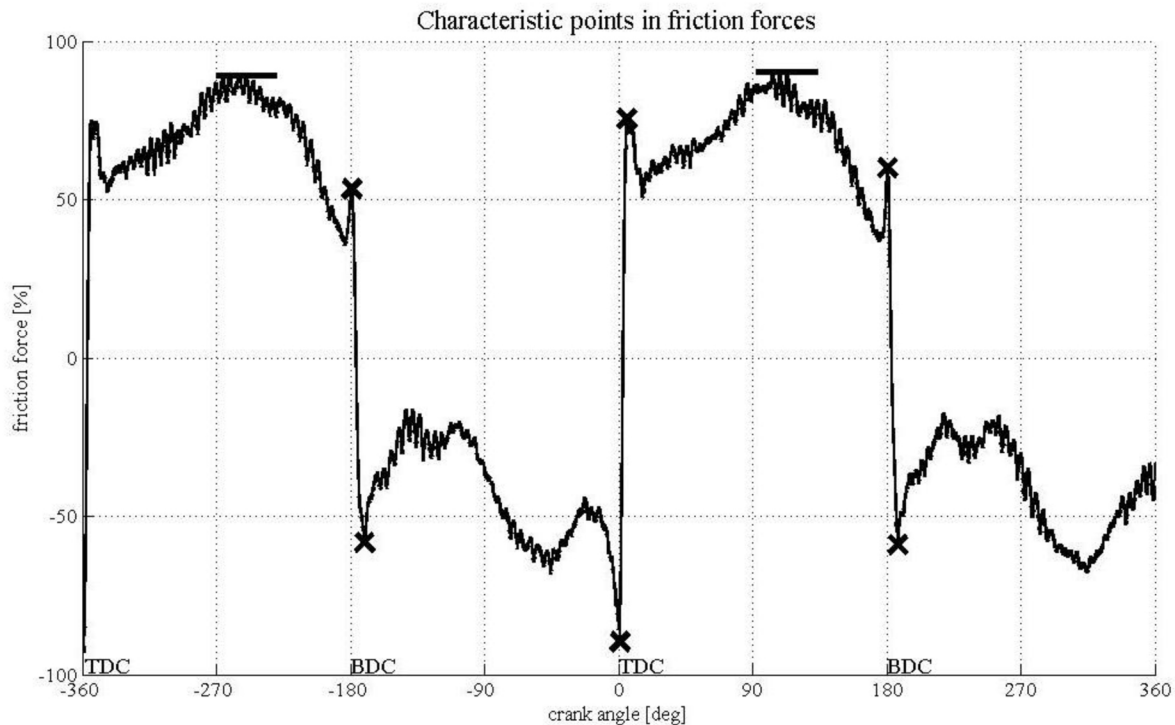


Figure 2. Characteristic points in the friction signal

CHANGES IN THE PROGRESSION OF THE FRICTION FORCE DURING RUNNING-IN

During the running-in the friction over the crank angle undergoes distinct changes. The main mechanism which appears during the running-in is the wear of the surfaces. In the tribological system of the piston assembly this applies to the piston lining, the rings and the piston skirt [8]. The wear particularly occurs in the dead centers, in which the piston speed relative to the liner is being reduced, and comes momentarily to a standstill. In this timeframe the rings may partially or entirely penetrate the oil film, mixed respectively dry friction occurs, with a large percentage of solid friction, see [4] and [8]. Due to this, the areas of the dead centers can be used as characteristics points for the representation of the running-in. The difference between maximum and minimum around the BDC, as well as around the TDC, is used as a characteristic feature, herein after referred to as step height. Tests have shown that, particularly in the BDC, more stable conditions arise, as the oil supply shows smaller cycle fluctuations here than at TDC.

During the engine running-in, roughness peaks are worn off and the surface is smoothed, see [4]. Mixed friction values decrease until stable friction characteristics are reached, see [3]. In the process the crank-angle-resolved friction signal changes are depicted in Figure 3. The chart shows the running-in characteristics of the entire piston/ring/liner-

package at 680 rpm and 90 °C oil and water temperature in pressureless operation.

There is a distinct decrease of mixed friction at the start of the running-in. At TDC and BDC the main decrease occurs within the first ten hours. The analysis of hydrodynamic friction maxima shows a similar pattern, as does the mean friction pressure. Towards the end of the component's operating life the wear strongly increases up to the point of the component's failure, see [7]. This behavior was not examined within the scope of the research project.

TRIGONOMETRIC FUNCTIONS FOR THE CHARACTERIZATION OF RUNNING-IN

There are several available approaches to characterize running-in, see Figure 4. The chart shows the running-in characteristics of the above mentioned configuration over the course of the operation time. The measured data equate to the difference of the mixed friction peaks at 680 rpm at BDC.

Several approaches were chosen to illustrate characteristic friction values: a four-degree polynomial, a logarithmic approach, and a hyperbolic cotangent. Latter approach proved to be best suited for the mathematic depiction of the measured points. With this approach, the strong decrease of the friction values at the beginning is depicted most clearly. The polynomial approach can also be used, but it tends to over-fitting to measured outliers caused by minimal oil temperature derivation during the running-in. A lower degree

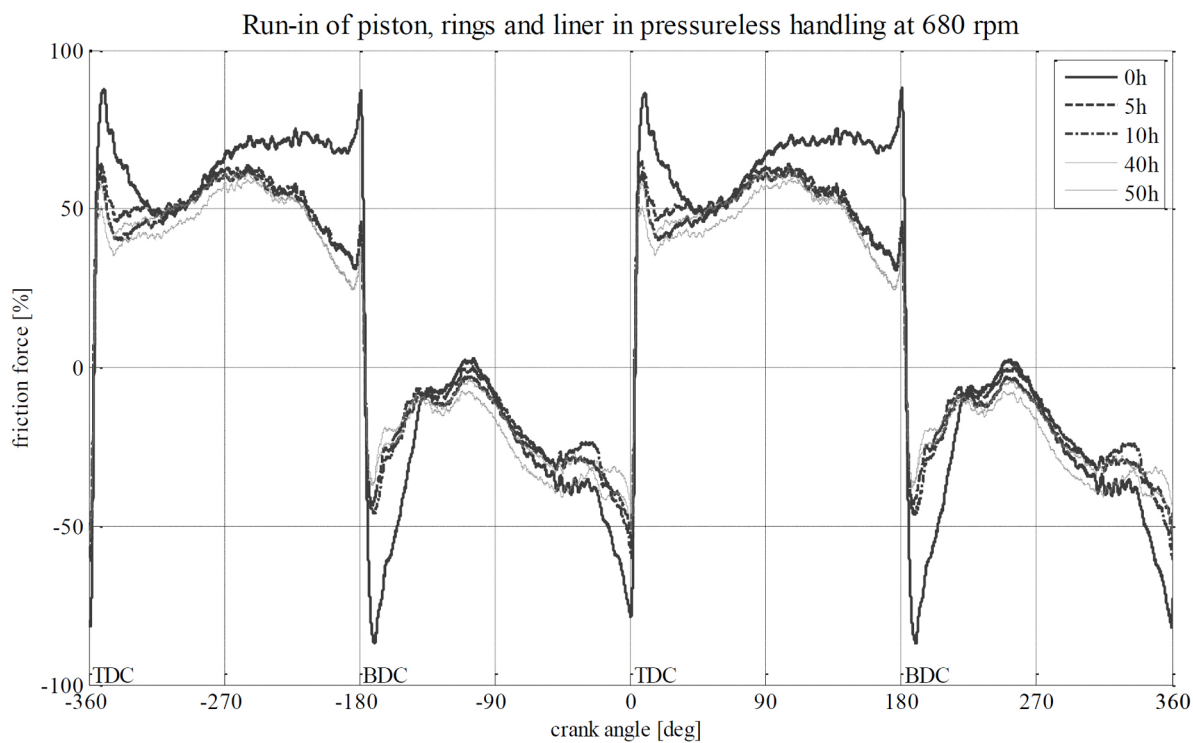


Figure 3. Change of the friction curves during the running-in

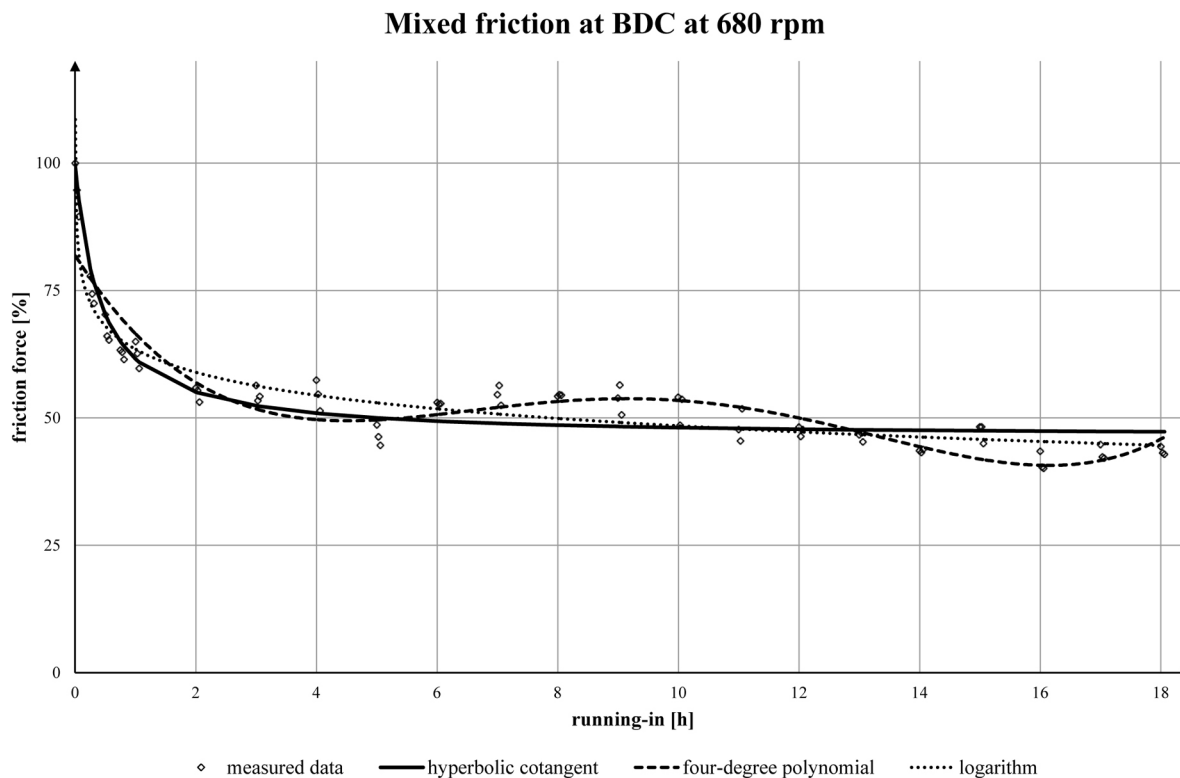


Figure 4. Approaches for the depiction of running-in characteristics