

Analysis of the Mixed Friction in the Piston Assembly of a SI Engine

Andre Merkle
TU Muenchen

Sebastian Kunkel

Georg Wachtmeister
TU Muenchen

ABSTRACT

Presented within the framework of this SAE Technical Paper are the highly accurate results of friction experiments, performed upon a floating-liner, single-cylinder test engine with a capacity 0.5 liters and crank angle resolution during motored and fired operation. This allows for the measurement of mixed friction zones at the dead centers. These mixed friction zones can result in friction losses and lead to wear in the components in-volved. The strength of the friction forces in any given mixed friction zone is largely dependent on the operating point. This is why the influence of each of the most important operating parameters - speed (rpm), load, oil and coolant temperature - is individually analysed, before the interactions, which are depicted in the resultant engine map, are discussed.

CITATION: Merkle, A., Kunkel, S. and Wachtmeister, G., "Analysis of the Mixed Friction in the Piston Assembly of a SI Engine," *SAE Int. J. Engines* 5(3):2012, doi:10.4271/2012-01-1333.

INTRODUCTION

Due to current CO₂ emission laws, the automobile industry in particular has the responsibility to reduce the fuel consumption of its vehicles. This can be effectively achieved by reducing losses in the combustion engine. Besides thermodynamic losses, mechanical losses also play a significant role here. The majority of mechanical losses are caused by the piston assembly, which is why further focused development of the piston assembly has the greatest potential for reducing fuel consumption. Any consideration of the piston assembly does, however, require a holistic approach, which takes four variables into account: friction, oil consumption, blow-by and wear. For some time, engine developers have been able to rely, with sufficient accuracy, upon established methods for the determination of three of those variables; namely blow-by, oil consumption and wear. This has meant that those variables could be accurately taken into account during the further development of the piston assembly. However, the process of determining friction losses is significantly more complex.

Within the body of relevant literature there are occasional mentions of test engines, which allow for the determination

of friction losses in the piston assembly with a sufficient degree of accuracy. Due to the complex stroke movement of the piston, entirely different tribological conditions can be found in dead centers and the areas in between them. In the case of high oil temperatures and low speed (rpm), especially in the dead centers, there exists mixed friction with a large proportion of boundary friction. In the areas between the dead centers, hydrodynamic lubrication conditions occur due to high piston speeds. These conditions are promoted by high speed and by low oil temperatures. The piston assembly in contemporary engines is already highly sophisticated, but in order for focused further development to occur the complex tribological conditions described above must be taken into account accordingly.

Under the auspices of the Institute of Internal Combustion Engines at the Technische Universität München, a test engine was successfully built and operated, which allows for the highly accurate determination of friction forces in the piston assembly of a turbo-charged direct-injection 0.5 litre SI engine. Extensive tests were carried out to examine the friction variable in the piston assembly during motored and fired operation. The focus of this paper's analysis and interpretation of the test results is upon the friction conditions

Table 1. Key values of the engines geometry

Bore	82.5 mm
Lift	92.8 mm
Capacity	0.5 l
Connecting rod length	144 mm
Connecting rod ratio	0.32
Mean piston speed at 7000 rpm	21.7 m/s

in the dead centers, as these are particularly dependent on the engine operating parameters, speed, load, and operating temperature.

TEST CARRIER

A single-cylinder test engine equipped with a device for measuring direct friction force is used. The engine is based upon a turbo-charged direct-injection passenger car SI engine and was built specifically for the purpose of measuring the friction variable in the piston assembly. The engine features an 82.5 mm bore and a 92.8 mm lift, adding up to a capacity of 0.5 liters. It can be operated at a speed of up to 7000 rpm. In order to facilitate results that would be transferrable to the production line, the crank drive was adopted from a basic engine, whilst the connecting rod and the piston assembly are standard parts. The connecting rod ratio is 0.32. In [table 1](#) the characteristics of the test carrier are provided. For more information about the reference engine please refer to [\[2\]](#) and [\[3\]](#). Through the selection of these parts, combined with the use of the actual geometry of the cylinder head, important conditions are created, which allow for the transferability of the study's results to the "real world" of the production line.

In order to create higher measurement resolution, the engine is run at speeds between 500 and 1500 rpm. During the tests, constant speeds are maintained by a DC dynamometer. Loads start with motored measurements up to an indicated mean effective pressure of approximately 8 bar. Coolant and oil are supplied by an external conditioning unit, through which temperature and pressure are freely adjustable. During the tests, the motor is operated with oil and water temperatures at (respectively) 35 °C/35 °C, 90 °C/90 °C, and 120 °C/105 °C, as well as with an oil pressure of 1.8 bar. In order to facilitate near-production-line results, the original piston-cooling nozzle is used and permanently activated during the tests. As vibrations of the basic engine must be avoided during friction force tests, a first and second order Lanchester balance shaft is employed. An especially large flywheel is used in order to enhance the smooth running of the engine during operation, so that the oscillation angle of

the single cylinder engine is equivalent to that of a four-cylinder reference engine.

The test engine operates in accordance with the floating-liner method developed by Furuhami (also known as Musashi Engine) - see [\[3\]](#) and [\[5\]](#). [Figure 1](#) is a schematic diagram of the applied measuring system. The cylinder liner unit features an integrated cooling jacket and it is mounted on two piezoelectric Kistler 9061 measuring boxes (see [\[6\]](#)). Both measuring boxes are mounted on the crank axis. As this set-up is only equipped with two measuring boxes the piston side forces have to be supported via two radial mountings, one of which is installed on the pressure side, whilst the other is installed on the counter-pressure side. As is commonplace in the floating liner method, a gas-balance device is employed to minimize the impact of the cylinder pressure on the friction measurements. In the measuring system at hand the unit does not require space in the cylinder head or in the piston assembly, allowing for the use of the original piston and cylinder head geometries. As a result of this, the original position of the fuel injector can also be maintained. The volume of the gas-balance device lowers the compression ratio. In essence, the friction forces that act on the compression chamber side of the piston assembly are only influenced by the cylinder's pressure and temperature. Therefore, an appropriate application strategy allows for the sufficiently accurate emulation of the characteristics of those parameters, as they would be in production-line engine conditions.

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 3 mega samples with chronological synchronization via an internal clock. According to the Nyquist-Shannon sampling theorem (see [\[7\]](#)), this approach yields a sufficient number of recorded measuring points for a detailed reading even of mixed friction peaks at high speeds. The processing routine is implemented via the commercially available MATLAB soft-ware, which creates an average from the measurements taken over those three seconds, thereby providing the figures for a representative engine cycle. The

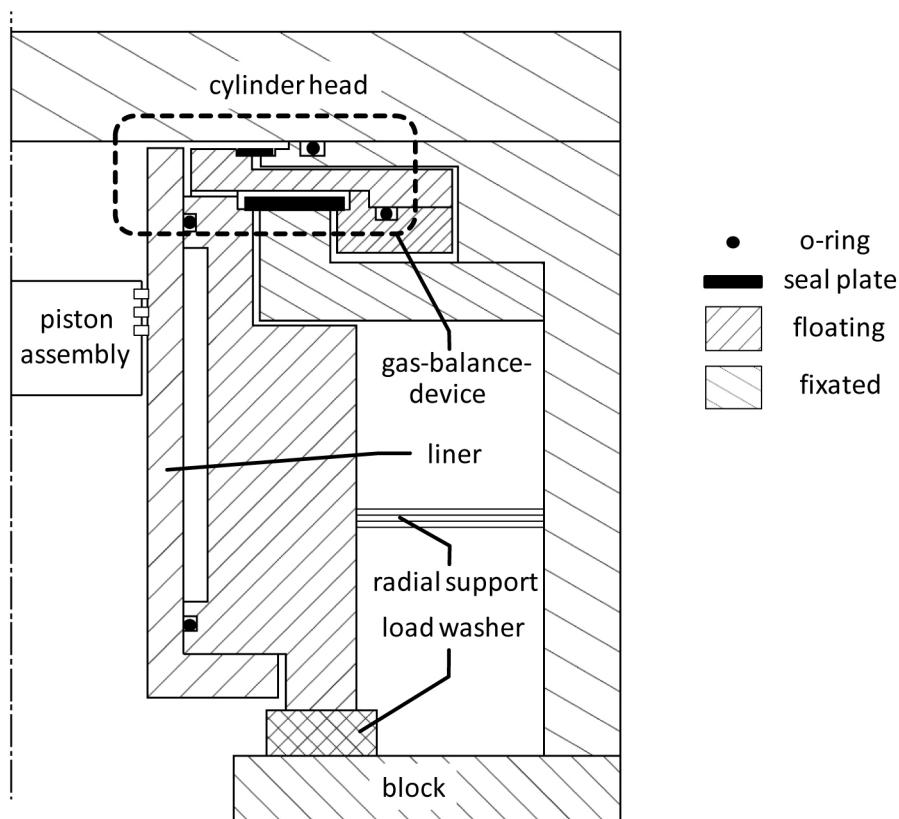


Figure 1. Schematic diagram of the measuring system

nature of the test setup being used dictates that friction forces are superimposed by the setup's self-oscillation. In order to create an optimized graphic presentation of the measurements, the data is filtered of the effect of self-oscillation by a band elimination filter and further processing is carried out.

In order to guarantee the appropriate reproducibility of the test results, the engine is first subjected to preliminary test-runs, which have identical operating points to those of the test proper. This means that upon commencement of the measurements, the tribological conditions are a constant. Measurements are taken for each operating point three times, and the results are compared. An average value, which acts as the representative measurement for that particular operating point, will only be chosen if the friction force curves correspond to a sufficient degree.

VARIATION IN MOTOR OPERATION PARAMETERS

In a tribological system, generally two different types of friction can occur. During direct contact between two objects boundary friction occurs. If the friction partners are completely separated by an oil film, the force that needs to be overcome is made up entirely of the inner friction of the fluid. This is called hydrodynamic friction. Both types of friction may appear simultaneously, in this case it is referred to as

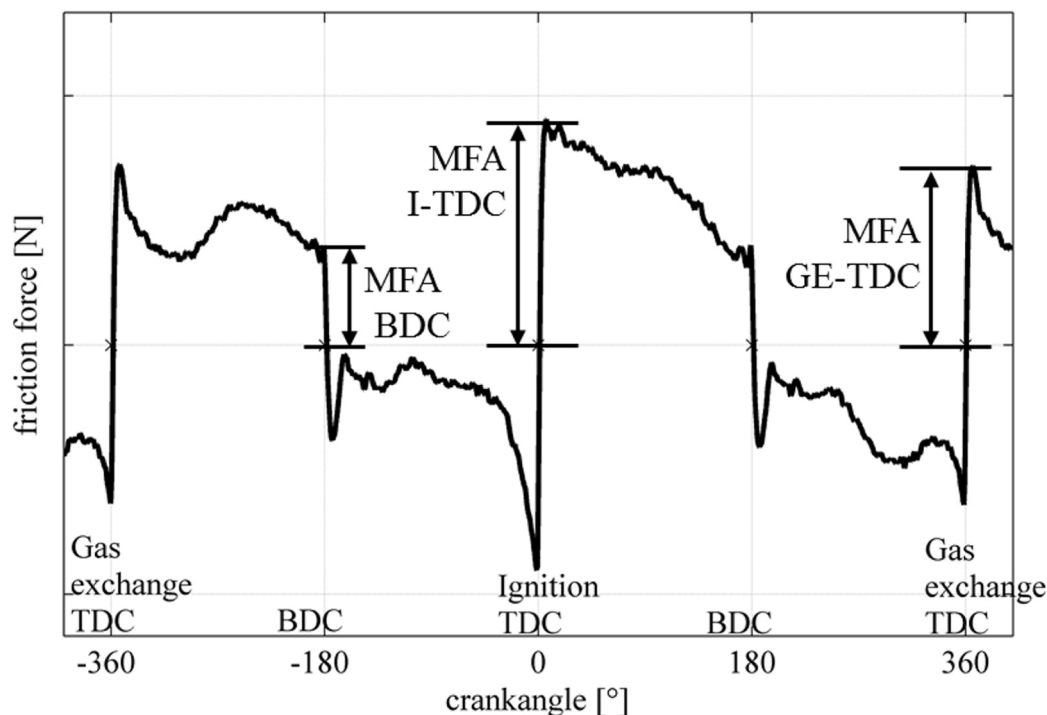
mixed friction [10]. In the piston assembly system high levels of mixed friction can occur in the top dead center (TDC), when large loads are present, particularly in the high-pressure phase. These mixed friction forces are said to cause liner wear at the TDC [8], [9]. If liner wear exceeds a certain point, oil consumption and blow-by increase to an impermissible level. This is why the analysis of mixed friction as a function of motor operating parameters is particularly important with regard to the TDC.

In the piston assembly, lubrication conditions leading to mixed friction predominantly occur in the dead centers. In this instance, the oil film can be penetrated, due to the fact that the sliding speed is approaching zero. Under these conditions, the retention force of the oil film is not sufficient to keep the friction partners apart. This results in direct contact between the friction partners, which leads, accordingly, to increased friction forces and wear.

Below, the results of tests measuring the mixed friction characteristics of the piston assembly are presented against engine operating parameters such as speed, load and temperature. In order to examine the exclusive influence of these parameters, they were varied individually during the first step. In the subsequent holistic examination, the entire experiment space was studied, so that interactions could be observed as they occurred. With regard to the piston assembly, the chosen parameters - speed, load, and oil

Table 2. Influences of operating parameters on friction within the piston assembly

Operating Parameter	Influence on the tribological system of the piston assembly	Influence on the friction within the piston assembly
Speed	▪ sliding speed ▪ local duration of the load	▪ lubrication condition ▪ friction power loss
Load	▪ axial force (over piston side force and pressure on the piston ring) ▪ complete system temperature	▪ lubrication condition ▪ direct friction (increased normal force)
Oil temperature	▪ viscosity of intermediate ▪ complete system temperature	▪ lubrication condition ▪ shearing resistance and pressure buildup in the oil film

**Figure 2. Characterization of mixed friction in the friction force signal**

temperature - are the most important engine operating parameters, as all of them directly influence the lubrication conditions. [Table 2](#) presents a summary of the each parameter's influence on the tribological system of the piston assembly and the friction within the piston assembly accordingly. All geometrical parameters of the piston assembly, like ring tangential force and piston cold clearance, are kept constant during the experiments.

In order to characterize the mixed friction that occurs, the recorded value of the friction force in the dead centers is

used. This value may be considered as the amplitude of the friction signal; the so-called mixed friction amplitude (MFA). Different MFAs for the respective dead centers were determined during the tests, making it possible to differentiate between the MFA in the top dead center during the high-pressure phase (I-TDC) and the MFA in the top dead center during the gas exchange phase (GE-TDC). [Figure 2](#) illustrates this relationship.