# High precision piston to liner friction measurement

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**ABSTRACT:** Piston to liner friction is responsible for up to 50% of total engine friction losses. A “floating liner” single cylinder engine together with measurement and test bed modules has been developed to enable high precision piston to liner friction measurement at engine operation conditions relevant for today’s and future engine test cycles. The engine is operated with conventional cylinder heads adapted for the floating liner interface, exchanging piston or liner hardware is accomplished without interference to the engine’s sensor package. The paper describes design elements, gives insight into friction force measurement and presents typical friction phenomena.

**KEY WORDS:** internal combustion engines, piston liner friction, floating liner, fuel efficiency, multichannel dynamic force measurement system.

## 1. INTRODUCTION

The friction forces between liner and piston of an IC engine are under influence of parameters related to the choice of materials, the surface structure of contact elements, lubrication, the force components resulting from piston motion and gas pressure, piston to liner clearance and temperature influence on each of these parameters. Engine design and selection of components thus are under growing need for precise data to select materials and design specifications best suited to contribute to low friction and fuel efficient engine design.

Measurement of piston to liner friction has been introduced with the “floating liner” engine concept [1,2,3,4]. Such design requires the liner to be mounted via force sensors onto the crankcase. Contact forces between liner and cylinder head or any other parts of the engine must be avoided. With this design the liner is “floating” on its force sensor supports, any dynamic force introduced into the liner is recorded as an add-on to the static force required to hold the liner in place.

The efforts put into friction reduction by means of any of above mentioned parameters and the complexity of interrelated force dynamics define the need for high precision friction measurement. Furthermore, the practicality of engine testing requires a floating liner engine design capable of operation at relevant engine conditions and providing mechanical interfaces for an easy exchange of piston or liner without compromising the setup of the sensor system.

On this background, a floating liner engine has been designed and tested with design targets related to

- Engine operation at relevant speed and load
- A high precision friction force measurement system
- Capabilities for easy exchange of engine components
- An engine test bed environment to maintain precise definition of boundary condition

Design concept, engine test data and procedures to operate the engine for precise friction force measurement are presented.

## 2. Engine design elements

The floating liner engine concept is shown in the insert of Fig. 1A: Its central parts are the liner carrier, the force sensor package and the sealing ring between liner and cylinder head. A design example and photographs of respective engine parts are given in Fig. 1B and 1C.

Liner carrier: This module connects the liner and its cooling water jacket with the engine block via the force sensors’ central screws. The mounting screws are tightened for a precisely defined static pre-load. The liner together with cooling jacket is inserted into the carrier and fixed with an annular ring, see Fig. 2.

Sensor package: Selection of force sensor stiffness and sensor positions within the engine structure ensures static stability of the liner. The sensors' dynamic response and sensitivity are key to providing accurate dynamic force measurements.

Sealing between liner and cylinder head is accomplished with a radial seal ring. The seal ring's contact to the liner together with gas pressure acting on the ring surface imposes residual force components acting along the liner axis. Even if this axial force can be minimized with selection of seal ring design parameters, the measurement system's high sensitivity shows its effects as a residual force superimposed on the basic friction signal components. Its effects on data accuracy is discussed in chapters 3.3.2. and 8.

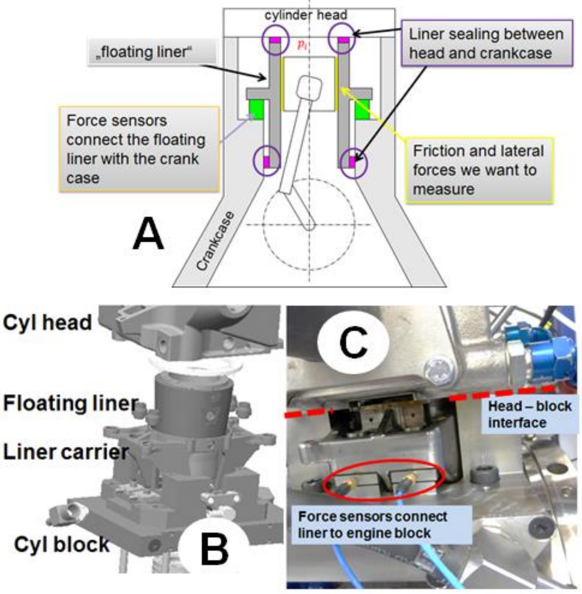


Figure 1: The floating liner concept (A), design elements with liner, liner carrier and sensor package (B), sensor group between liner carrier and engine block (C).

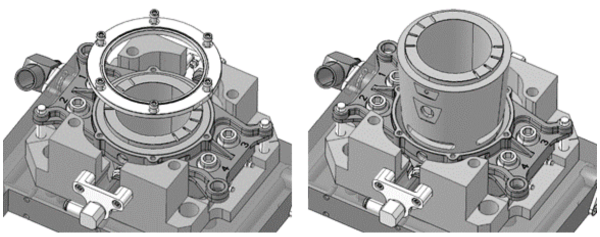

Figure 2: A design concept was selected to allow exchange of piston, piston rings and liner without interference to sensor package installation.

## 3. Signals

### 3.1. Sensor package

The sensor package comprises 4 force sensors, each providing force signals in axial, lateral and longitudinal direction. Each sensor signal component includes force components acting along the respective axis as well as residual components as a result of the sensor element's cross talk sensitivity. The relation between the engine's force components acting on the sensors to the sensors' force signals is best described with a linear equation system for each sensor. The signal to force sensitivity together with the individual cross talk sensitivity values constitute the elements of the sensitivity matrix. They are derived from factory calibration sensitivity data.

Handling the 12 force signal components is accomplished with a multichannel data recorder. The sensor package sensitivity matrix (in total comprising 36 sensitivity matrix elements) is used in a real time analyzer to provide instantaneous output of engine force components.

A schematic of the measurement system and signal sequences is shown in Fig. 3, 4 and 5. It is evident that the interdependence of sensor signal components and their contribution to the final force signal components requires highest possible precision of each individual signal. As any incorrect sensitivity value would result in inconsistent force data, signal quality evaluation by means of plausibility checks for residual data is part of the real time evaluation procedure. This ensures correct setup of the sensitivity matrix upon initial configuration of the sensor package and it allows detection of eventual sensor aging.

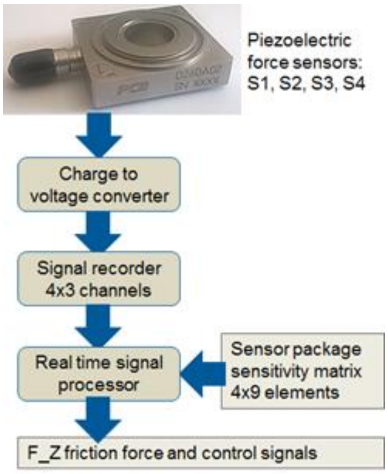


Figure 3: Force measurement system modules.

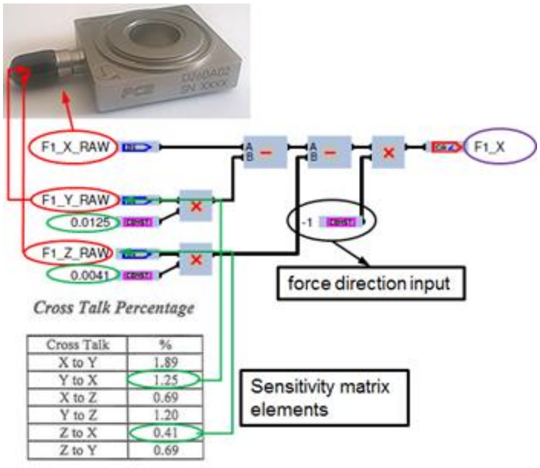


Figure 4: Force component per sensor is derived from „raw“ force signals and sensor sensitivity matrix elements.

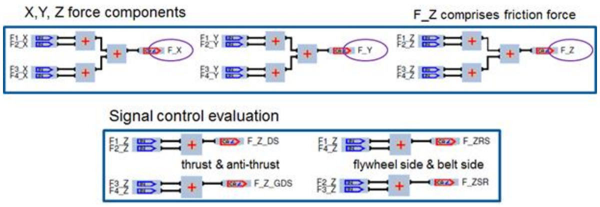


Figure 5: Resultant force components F_X, F_Y, F_Z comprise input from each sensor, each sensor component and each sensor‘s sensitivity matrix.

### 3.2. Friction signal examples

Examples for friction comparison tests in a motored Diesel and a fired gasoline engine are shown in Fig. 6 and 7. The crank angle resolved F_Z force components show the friction force evolution along the motion of the piston and under influence of cylinder pressure variation. Friction force profiles are due hydrodynamic as well as asperity effects at high and low piston velocity. The Diesel engine example highlights oil viscosity influence in response to moderate and normal engine temperature, the gasoline example shows a comparison of oil quality effects.

### 3.3. Secondary signal components

The F_Z signal traces in Fig. 6 and 7 not only show the friction force, but also include components arising from

1. oscillations and
2. the response of the sealing ring to pressure rise after TDC in the fired engine
3. Superimposed on each individual signal is sensor and amplifier noise

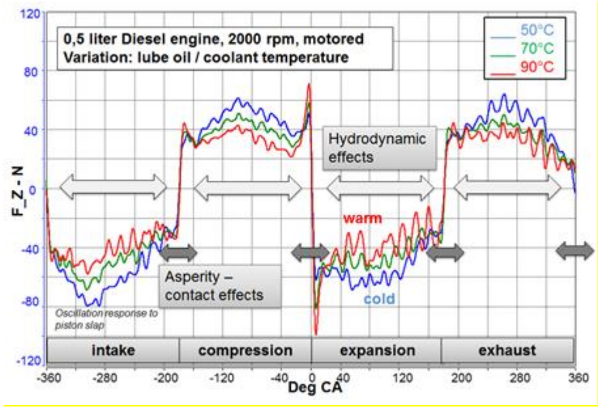


Figure 6: F_Z friction force signal examples in motored Diesel engine show temperature influence on friction.

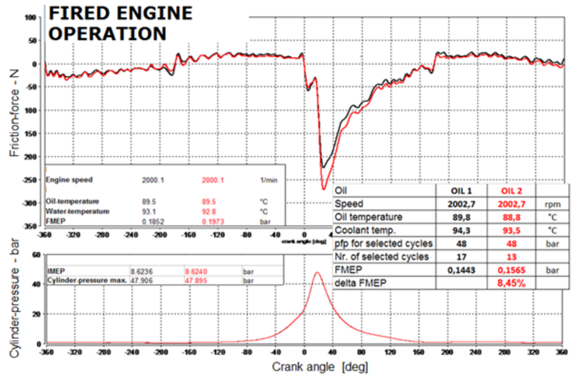


Figure 7: A friction test shows oil quality influence in a fired SI engine. Selected pfp: see chapter 6.

### 3.3.1. Oscillation signal components

Piston slap: The impact of the piston against the liner surface introduces a structural response of the liner and its carrier. The impact occurs as a consequence of the piston’s kinematics and inertia at well defined crank angle intervals and with an intensity depending on piston to liner clearance and damping effects of the lube oil. As these oscillations are transferred through the sensors, their signals show the effect of piston impact and the damped oscillation of the liner carrier structure, see Fig. 6.

Liner and liner carrier inertia: A solid body mechanical model of the forces acting on the liner is given in Fig. 8. The forces arising from piston motion and gas pressure are balanced by the reaction of the sensor “spring” package. The finite stiffness of the sensor package allows microscopic motion of the liner group with resultant solid body oscillation being balanced by reaction forces of the sensor package. Up to medium engine speed, this liner acceleration is negligible. At 4500 rpm, however, it introduces higher than real friction amplitudes. Even if the total influence on FMEP still is very small, the correction allows a high precision evaluation of asperity effects, see Fig. 9.

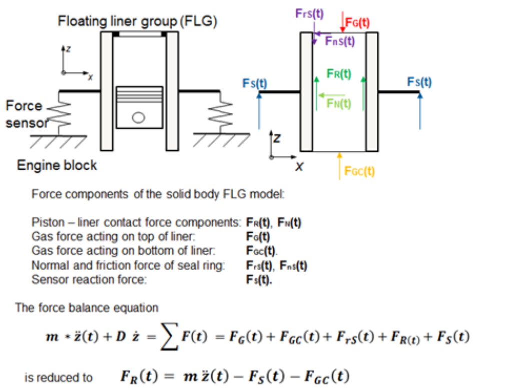


Figure 8: The solid body – spring model of the floating liner group mounted on the sensor „spring“ package. The force balance equation allows for inertia correction. This improves signal evaluation accuracy at high engine speed.

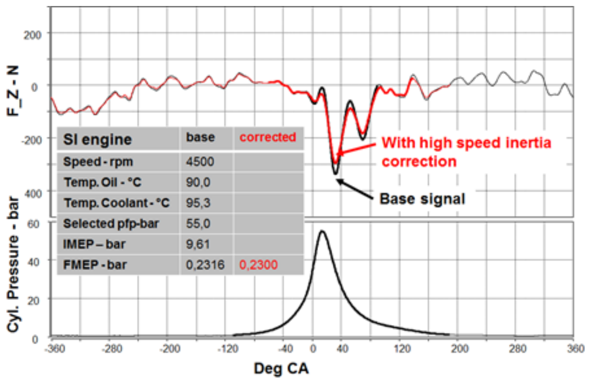


Figure 9: High speed inertia correction of friction signal improves precision of asperity evaluation. Selected pfp: see chapter 6.

### 3.3.2. The radial sealing ring motion

In fired operation, the friction signal shows a negative spike at around the crank angle position of maximum cylinder pressure. This spike arises from a microscopic motion between liner and seal ring as the liner is under the pull down by the accelerating piston and the seal ring is still under rising pressure influence from combustion. Its contact to the liner thus introduces a motion against the accelerating piston. However small this motion might be, its response is evident for a small crank angle window. Its influence on evaluation of asperity friction components is accounted for by signal integration procedures.

### 3.3.3. Sensor and amplifier noise

Sensors and signal amplifiers have been selected under the requirement of high S/N quality over the temperature and dynamic range of engine operation applications. It was found that primary signal quality was never limited by noise, any high frequency signal components are due to mechanical oscillations introduced by engine operation.

## 4. Data reduction

The force signals shown in the data examples are derived from the sensor signals with data processing procedures as described in Fig. 3-5. In order to reduce high frequency acoustic components, low pass filtering with a typical cut off frequency of 800 Hz is applied.

One aim of friction measurement is to evaluate the force data for FMEP - friction mean effective pressure. FMEP is a useful parameter for overall comparison of friction

behavior, however it is unsuited to differentiate effects of kinematic or pressure influence. As such information is contained within the friction force signals, it is straightforward to integrate the crank angle resolved signals for specific user defined windows.

In the FMEP comparison of hardware variants in Fig. 10 the friction data are integrated for each of the 4 engine strokes. The comparison, as expected, shows increasing friction with engine speed and load, differentiation of variants essentially occurs in the expansion stroke under medium to high load conditions. The example shows less friction with variant V2. Including this FMEP reduction in a fuel consumption simulation yields a BSFC improvement of 1 – 2 %, see Fig. 11.

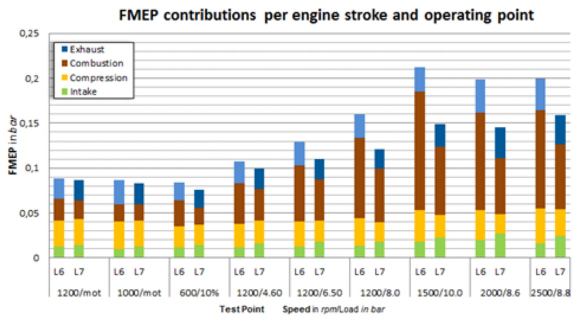


Figure 10: Gasoline engine, V1 / V2 variants comparison on basis of FMEP data evaluated in motored and fired operation. FMEP contributions over stroke intervals. Noticeable variants' benefits become apparent under fired conditions only.

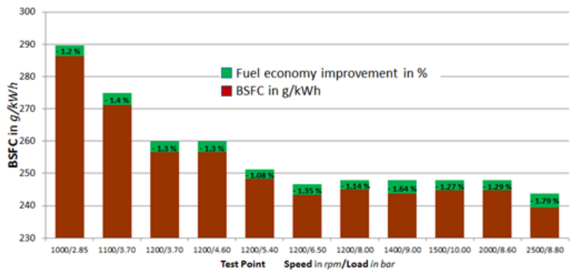


Figure 11: BSFC simulation based on FMEP comparison for V1 / V2 variants.

## 5. Test requirements

FMEP or BSFC comparison as described in chapter 5 is at the end of a friction analysis program to support decisions on selected hardware or lube oil variants. Analysis of friction details is of course performed with data comparisons as were already shown in previous examples. Here it was found to be mandatory to provide and maintain high precision boundary conditions including temperature stability, engine rpm and combustion pressure traces for given test points. Even in stationary operating points, with all external parameters kept constant, combustion pressure fluctuations in both amplitude and phasing have influence on signal repeatability. Cycle averaging, thus, should consider such influence and first select classes of cycles defined on basis of their pressure trace before performing cycle average operations on selected groups of cycles. Especially in SI engines with considerable cycle to cycle combustion fluctuations, such cycle selection before averaging was found to significantly enhance the quality and precision of variants comparison, see the data examples in Fig. 7 and 9.

## 6. Engine handling

The engine comprises of the three component groups: 1) cylinder head, 2) crankcase and 3) floating liner group. Design concept, layout and an engine details photograph have been shown in Fig. 1.The interfaces between the component groups allow exchange of piston and liner variants without dismounting the liner carrier from the crankcase. Exchanging components is thus done without any interference to the mechanical adjustment of the sensor package. Typical tests for any friction evaluation at stationary speed / load operation points include change of engine temperature, lube oil, or piston, piston ring and liner variants. The actions for such variants exchange and typical time requirements from engine stop to re-start are given in table 1.

| Variant | Action | Duration |
| --- | --- | --- |
| Temperature of lube oil and coolant | With an external conditioning unit | 2 h to stabilize and balance internal engine temperature |
| Changing lube oil | With external pump and reservoir | 1 h to drain, spill and re-fill oil circuit |
| Changing piston | Head, conrod, piston | 4 h |
| Changing liner | Head, conrod, piston, liner together with its water jacket | 5 h |

Table 1: Exchange of engine variants, typical time requirements

Cylinder head: Prototype or mass production cylinder heads need some minor modification to provide the space for mounting the seal ring between head and liner. Camshaft operation is achieved with a tooth belt, tooth wheels are mounted on to the front end of the camshafts. Original cylinder head bolts and bolt positions are maintained. Coolant and lubrication are provided via suitable hose connections to the base engine and further on to the media supply unit. With such modifications a multicylinder head is operated as a single cylinder engine, and engine operation can use original stationary engine calibration parameters.

Crankcase: The crankcase provides a module to allow the mechanical interface to the cylinder head bolts, the interface to the force sensors between crankcase and floating liner, and the housing for cranktrain and balancer shafts. Side openings provide a mechanical window to access the conrod screws. The engine uses first and second order balancer shafts to minimize crankcase vibrations.

Floating liner group (FLG): The FLG comprises the liner carrier, mounted via force sensors to the crankcase, and the liner with its cooling water jacket, see Fig. 2. A fixation ring connects the liner to the liner carrier. The design provides high stiffness and enables easy exchange of the liner together with its cooling jacket. Some design details are given in Fig. 2.

## 7. Floating liner engine versus normal engine

There are two features which are specific for the floating liner design when compared to the normal engine:

1. Liner – head seal ring: In fired operation, the response of the ring to rising pressure together with the downward motion of the piston introduces a microscopic stick – slip motion with a consequent reaction force seen in the force signals. The signal appears as a spike superimposed and counteracting to the friction signal of the accelerating piston after combustion TDC. Seal ring tension and contact force to the liner are the parameters of influence. As the sealing functionality requires a minimum contact force, effort is put into maintaining repeatability of the stick-slip effect rather than eliminating its signal component. In practical terms, the signal has an effect on direct evaluation of asperity effects within a crank angle interval around 10 to 30 deg CA ATDC.

2. Liner distortion: The distortion of the liner and consequently, the liner to piston clearance depends on the mounting force flux between liner and its surrounding structure. For high precision evaluation of piston to liner clearance effects, a liner honing procedure is applied to duplicate original liner distortion and thus maintain correct clearance dimensions.

## 8. System modules

As this paper has described engine and measurement system and has given some insight into data evaluation, it was emphasized that results accuracy depends on precise repeatability of test conditions. Consequently, friction measurement with a floating liner engine and its sensor and signal evaluation system, gains in accuracy and results quality with the use of a test bed system that provides well defined boundary condition. The friction test results presented in this paper were achieved on a test bed system with modules as described in Fig. 12.

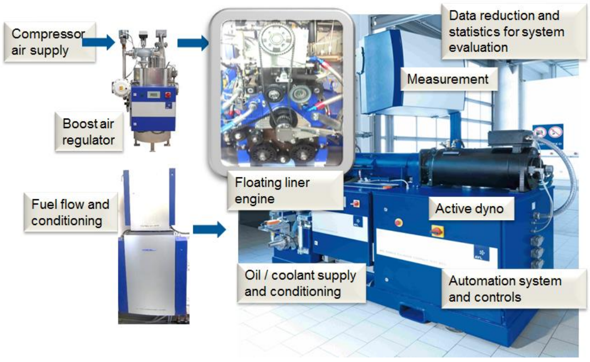


Figure 12: The floating liner engine is part of a test bed system comprising modules to enable precisely defined engine operation and measurement.

## Summary


A floating liner engine together with a force measurement system and peripheral engine conditioning and support modules has been designed and tested in order to achieve high precision piston to liner friction data. Measurement accuracy has been shown to be suited for the analysis of hydrodynamic and asperity friction forces in realistic motored and fired engine operation modes. Variants tests are supported by the simple exchange procedure of piston or liner variants.


## References


[1] Kwangsoo K., Paras, S., Takiguchi, M., Aoki, S.: “A study of friction and lubrication behavior for gasoline piston skirt profile concepts”, SAE 09PFL-1163, 2009.

[2] Koch, F., Geiger, U., and Hermsen, F., "PIFFO - Piston Friction Force Measurements During Engine Operation," SAE Technical Paper 960306, 1996, doi:10.4271/960306.

[3] Merkle, A., Kunkel, S., Wachtmeister, G.,: "Analysis of the Mixed Friction in the Piston Assembly of a SI Engine," *SAE Int. J. Engines* 5(3):1487-1497, 2012, doi:10.4271/2012-01-1333.

[4] Gore, M., Howell-Smith, S.J., King,P.D., Rahnejat,H.: “Measurement of in-cylinder friction using the floating liner principle”. IN: Proceedings of the ASME 2012 Internal Combustion Engine Division Spring Technical Conference (ICES 2012), 6 - 9 May, Torino, Italy.