

Reduction of Friction Losses by Means of Cylinder Liner Offset in a Floating Liner Single Cylinder Engine

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ABSTRACT: Piston to liner friction is responsible for a significant part (up to 50%) of total engine friction losses. Engine manufacturers use liner offset designs to address this issue and potentially reduce friction losses as is to be expected from theoretical considerations. A "floating liner" single cylinder engine was used to directly measure the effect of such liner offset design on the friction losses. Results show benefits to be gained at moderate speeds where cylinder pressure effects are the main drivers of piston to liner contact forces. At high engine speed this trend reverses due to piston inertia effects.

KEY WORDS: heat engine, internal combustion engines, piston liner friction, floating liner engine, liner offset, fuel efficiency [A1]

1. INTRODUCTION

Friction losses as a result of piston motion along the liner surface are under influence of various design, material and operation parameters defining the central part of an engine's tribological system. Direct measurement of friction forces acting on this system is accomplished with so called "floating liner" engine configurations. Main features of such engines are the separation of the cylinder head from the liner and, consequently, the balance of forces acting on the liner by means of a liner carrier system. Various solutions have been presented, see e.g. ^(1, 2, 3). Application of such floating liner engine for the measurement and analysis of liner offset engine designs is the topic of this paper.

Offset liner engine designs are frequently used in mass production engines to take advantage from the theoretical concept of reducing piston – liner contact forces arising during the high pressure combustion phase. Parameters of such designs include the selection of the actual offset dimension, optimization of the offset in view of the engine's most likely applied speed and load range, the selection of liner and piston surface parameters as well as the components' clearance profiles.

However, the benefit of such offset designs can be compromised by piston inertia forces. Consideration of engine speed range, piston mass and design details, consequently, is part of the optimization process.

The paper presents an example of friction loss measurements with the comparison of 0 and 10 mm liner offset, but otherwise identical hardware components. The floating liner engine in this study uses a radial seal ring to connect the cylinder head with the liner and a high stiffness sensor package to balance and measure the force components acting on the liner. This design allows for simple mechanical handling and easy exchange of piston, liner and liner offset to enable hardware variants comparison. The measurement

system and test bed environment have specifically been selected for operation of such floating liner engine under precisely controlled conditions. A description of the system has been presented in $^{(3)}$.

2. TASK AND TARGETS

Piston to liner friction is under influence of

- Piston side force Fs: is a result of piston motion and cylinder pressure
- Piston speed v: as given by axial piston motion and engine speed
- Lubrication of contacting surfaces, summarized by the Stribeck friction coefficient μ .

This all results in friction power loss:

$$= Fs * v^* \mu \tag{1}$$

Each of above factors, their variation over the engine cycle, and especially their effective combination offers chances for friction force reduction. The focus of work presented in this paper was the evaluation and testing of measures to find optimum combinations of piston side forces and piston speed by means of offsetting the liner axis against the crankshaft.

3. EXPERIMENTAL

A single cylinder engine with bore and stroke dimensions typical for passenger car gasoline or diesel engines has been designed to allow the direct measurement of the friction force

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acting between piston and cylinder liner. Main design elements of this floating liner "FRISC" engine are shown in Fig. 1, a comprehensive description of design, operation and test bed requirements for such floating liner engine operation has been given in ⁽³⁾.



Fig. 1: Components of the "FRISC" floating liner engine and parameters relevant for piston to liner friction

An offset between liner and crankshaft is achieved with moving the head – liner assembly into a side position. This requires an oval opening to the crankcase, see Fig. 2, and an interface plate between liner carrier and crankcase for any given offset. The thickness of such "baseplates" is adjusted to compensate for the offset influence on compression ratio. The plates, furthermore, have coolant channels to ensure well controlled temperature conditions.



Fig. 2: Specific baseplate and seal to crankcase enable liner offset variation in FRISC engine. All other components are unchanged.

4. MEASUREMENT

The force "sensor package" as was described in detail in ⁽³⁾ is balancing and recording all force components acting on the cylinder liner. From in total 12 force signals (3 per each of the 4 sensors) axial, lateral and longitudinal force components are derived. Evaluation of the lateral force components yields the side force Fs (or Fy) acting between piston and liner. The friction force Fz is evaluated from the axial force components.

In addition to force signals it is essential to record cylinder pressure signals and use them as a filter to classify combustion cycles with respect to the actual pressure trace per cycle before performing any signal averaging or other statistical evaluation. This ensures that evaluation statistics for variants comparison do not suffer from cycle to cycle combustion variations. See also ⁽³⁾ for details of selecting cycle classes based on their cylinder pressure trace.

Experimental boundary conditions such as media temperatures and pressures are kept constant by means of adequate test facilities.

5. CYLINDER LINER OFFSET

Liner offset designs are frequently applied in mass production engines in order to reduce friction losses resulting from contact force peaks arising in response to peak combustion pressure at any given position of the conrod. Ideally, the conrod should be parallel to the cylinder liner axis at the moment of peak pressure in order to minimize the side force component Fs. The offset, furthermore, results in lower piston speed at the start of the expansion stroke. Both effects yield a reduction of friction losses in the early phase of the expansion stroke. The offset concept with definition of main parameters is shown in Fig. 3.



Fig. 3: The offset concept. Note that TDC is related to piston, not to crankshaft.



The overall effect of this particular friction force reduction, however, is compromised when piston inertia forces start to dominate friction losses at high engine speed. Even if an improvement of combustion pressure effects may still be achieved around the time of peak firing pressure, lateral piston acceleration arising throughout the entire engine cycle can compromise the initial benefit.

Results of an analytical analysis of piston side forces under consideration of both cylinder pressure and piston motion effects are shown in Fig. 4.



Fig. 4: Simulation of piston offset effects for high load operation at low and high speed. Gas and inertia forces have opposite influence

The influence of side force variations as shown in Fig. 4 on resultant friction mean effective pressure (FMEP) arising from the piston skirt to liner contact is given in Fig. 5. The result, as expected, shows some maximum friction improvement at low to medium speed and high load before friction again rises as inertia forces start to dominate.

In the friction power loss equation (1) the side force and piston velocity effects may be well covered by analytical relations including piston kinematics and cylinder pressure traces. Uncertainties, however, arise from the friction coefficient μ which is under influence of design and operation parameters related to the contact surfaces, the lubricant, the

clearance between piston and liner, parameters of the ring package and all of these components' response to temperature variations. The experimental test of such design and operation parameters' effects on friction behavior is subject to operation of the floating liner FRISC engine.





6. TEST EXAMPLES

Piston side force traces at motored and fired conditions at low and high engine speed are given in the examples of Fig. 6. Bore and stroke of the single cylinder test engine are typical for a 0,5 liter per cylinder gasoline engine. Tests were done with port fuel injection in naturally aspirated combustion mode.

At motored low speed operation (Fig. 6A), the effect of liner offset is simply related to the timing shift introduced by the geometric offset. The benefit of smaller side force (Fy / 10 vs Fy / 0) in the early expansion stroke is at the cost of higher side force in the late compression stroke. The net effect is close to zero or even negative.

In fired operation (Fig. 6B), with combustion pressure now peaking near piston TDC position, the expected reduction of piston side force is significantly larger than the negative effect in the compression stroke. At 1200 rpm, 8 bar IMEP, the overall reduction of piston – liner friction losses is around 20%.

At 4500 rpm, 9 bar IMEP (Fig. 6C), the signal traces show the ever growing influence of inertia on the piston side force. The 10 mm offset still yields a piston – liner related FMEP reduction of around 5%, but this benefit will disappear and even reverse at high speed low load operation.





Fig. 6: A comparison of side forces (Fy/0 and Fy/10) measured in a FRISC engine configuration with 0 and 10 mm liner offset. Effect on friction force (Fz) depends on speed and load.

7. FRICTION POWER LOSS COMPARISON

With access in the FRISC engine to axial force data Fz, friction power losses (Pf) are evaluated with

$$Pf = Fz * v \tag{2}$$

v: piston speed.

Integration of Pf traces over degree crank angle intervals yields FMEP for the selected interval:

$$FMEP_{i} = F_{Z_{i}} \cdot v_{p_{i}} \cdot \frac{\Delta t_{i}}{V \cdot 10^{-3}} \cdot 10^{-5}$$
(3)

FMEP [bar].....friction mean effective pressure

Fz [N].....friction force acting between liner and piston

Vp [m/s].....piston speed

V [liter].....cylinder displacement

 Δt_i [deg CA]...crank angle interval for FMEP evaluation

i.....interval index



Fig. 7: Friction power loss versus deg CA in motored and fired operation. Data derived from Fz traces of Fig. 6 and piston velocity. Pf /0, Pf /10: 0 and 10 mm offset.

A comparison of friction power loss traces is given in Fig. 7:

<u>Motored operation at 1200 rpm</u>: Power losses peak in the mid to late compression stroke as a result of piston speed together with side forces. In the expansion stroke power losses are found to be smaller. Both variants with and without offset show almost identical power loss traces.

Fired operation at 1200 rpm, 8 bar IMEP: This low speed high load point shows a significant benefit with the 10 mm liner offset. As expected from simulations, shifting the peak pressure interval closer to TDC reduces the side force components and simultaneously benefits from lower piston speed when side forces start to rise with the beginning expansion stroke. The overall reduction of FMEP (calculated over the entire engine cycle) related to piston – liner friction is up to 20%.

<u>Fired operation at 4500 rpm, 9 bar IMEP</u>: This operating point first of all shows the large influence of engine speed on friction, with power losses peaking in the range of 8000 Watt against the 600 Watt at 1200 rpm. The piston side impact on the liner now forms a dominant part of the signal with oscillations related to the lateral piston motion. Liner offset still shows a benefit during the high pressure combustion phase, its overall improvement, however, is reduced to around 5% of the piston - liner related friction losses.



The power loss signals for the two fired load cases show a signal artefact at around the time before peak combustion pressure. This effect has been described in ⁽³⁾. It is related to a stick-slip reaction of the head to liner seal ring. Its effect on variants comparison has been found to be small as long as test reproducibility is maintained.

8. TEST RESULTS

The application of above described test procedure for the evaluation of liner offset effects on an engine's typical speed and load spectrum needs to apply combustion modes as given by standard engine calibration. This implies adaptation of an original multicylinder head for operation on the single cylinder FRISC engine and the use of the engine's standard calibration parameters. An alternative to using a multicylinder head in single cylinder operation is the reproduction of the original engine's combustion system in a single cylinder head, operation again is with the use of its standard calibration map.

The result of testing the piston – liner related friction losses at a liner offset of 10 mm against the same configuration without offset is shown in Fig. 8. The x-axis comprises speed / load combinations starting with 1000 rpm / 3 bar IMEP and ending with 4500 rpm / 9 bar IMEP. Peak firing pressure levels are given in the diagram for each operating point. The difference between offset / no offset pressures is due to calibration requirements.



Fig. 8: Friction force measurement evaluated for FMEP. Example shows largest friction reduction at moderate speed and high load (below 2500 rpm, up to 10 bar IMEP)

The FMEP test results show friction improvements to be gained with the 10 mm offset. As to be expected from the simulation analysis, friction reduction is most effective at moderate speed and high combustion pressure. The FMEP data in Fig. 7 of course relate to the entire piston package including skirt and piston rings, whereas the simulation analysis with the data of Fig. 5 relates to piston skirt effects only.

The benefit of testing friction behavior in a FRISC engine configuration is seen in the engine's flexibility to exchange and compare hardware variants. An example is given in Fig. 9 with a comparison of liner variants 1 and 2, both tested with and without a 10 mm offset. Under same operating conditions, variant 2 shows lower friction losses. The improvements achieved with the offset are similar, see the data and trend given in Fig. 9.



Fig. 9: Liner variants comparison. The low friction liner (variant 2) also benefits from offset, but overall improvement is smaller than with variant 1.

9. ENGINE HANDLING FOR OFFSET VARIANTS

Practical testing with a floating liner engine requires easy access to relevant engine components without interference to the sensor assembly. This requirement is met

1. With the radial gasket solution providing an easy to use and reliable interface between cylinder head and liner

2. With the sensor package design which, after initial adjustment, is kept sealed for the rest of a test series. Its design allows exchange of piston, liner or offset modules without disassembling the sensor package. This avoids inaccuracies arising from re-adjustments and thus ensures good test repeatability.

Operation of the engine and related test bed modules for high precision measurement has been described in ⁽³⁾.

SUMMARY

Liner offset designs are frequently used in production engines to optimize friction behavior. Theoretical considerations are well understood, the trends shown in this paper confirm the basic considerations. The direct measurement of friction forces acting between piston and liner furthermore shows the absolute amount of power loss improvements to be achieved by the selected offset of 10 mm. The dependence on speed and load follows the trends to be expected with respect to cylinder pressure and speed. Comparison in above examples for liner variants shows the influence of liner surface quality on friction.

The data examples given in this paper in Fig. 6 and 7, with all details of crank angle resolved friction loss events show the quality of analysis to be achieved with the design and operation of the FRISC engine and its measurement and test procedures. The engine shows good applicability up to an engine speed of 4500 rpm. Data reduction procedures are implemented in standard engine data acquisition systems with access available to both integral FMEP and IMEP data as well as detailed insight in to crank angle resolved sensor signals.

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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) Winklhofer E., Loesch S., Satschen S.: "High precision piston to liner friction measurement", 2016, JSAE Annual Congress Proceedings, No. 75-16, p. 1864-1869 (2016)