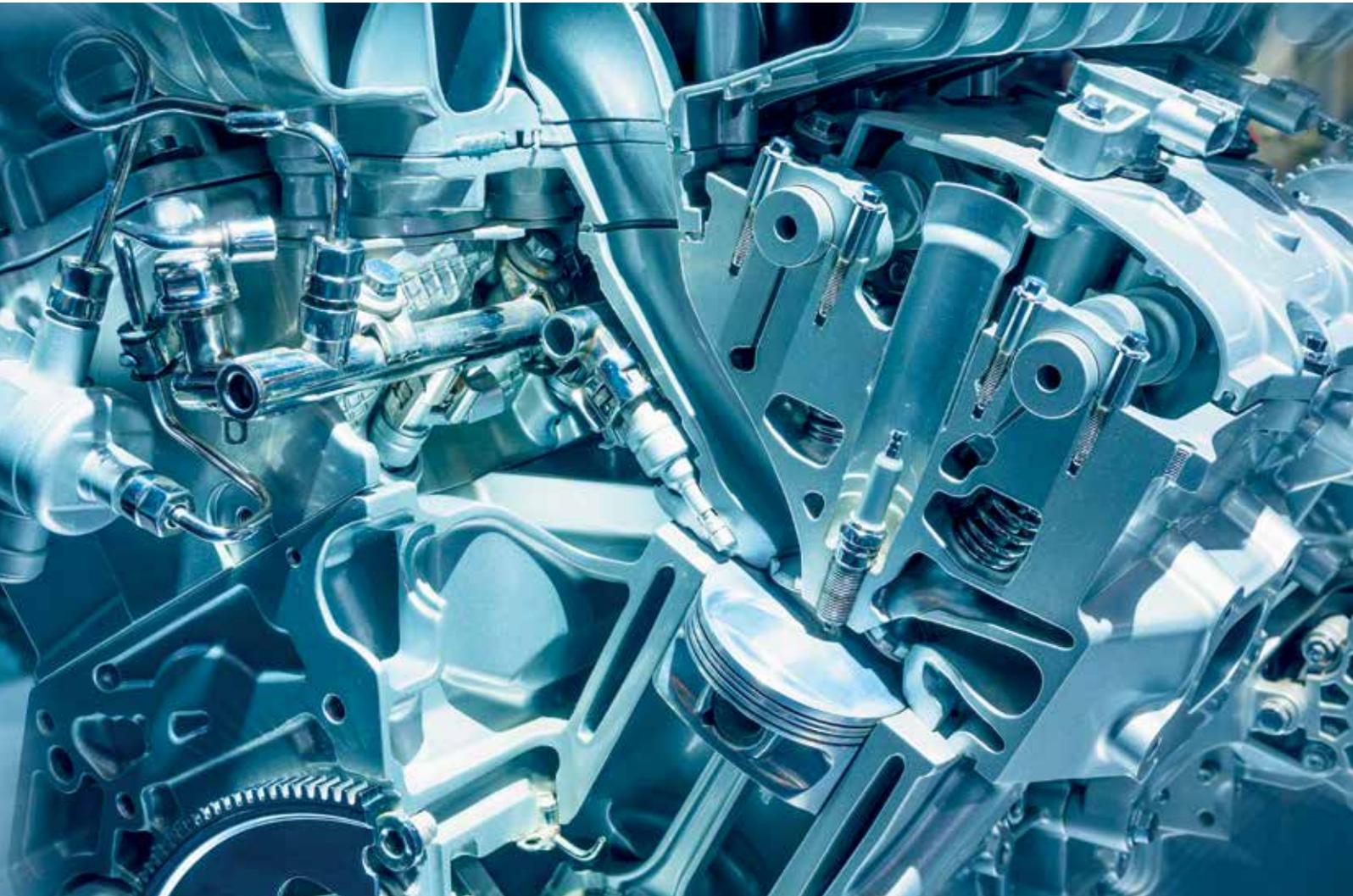


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Methods of Measuring Friction under Motored Conditions with External Charging

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Introduction

Combustion engines are still crucially important today in many areas: personal mobility, goods transportation, aviation, shipping and rail transport as well as the power supply sector, to name but a few examples. However, the outlay on developing these engines is increasing due to modern regulations on emissions – which impose ever decreasing limits – and efforts to devise more efficient drive concepts. Other factors are also becoming more critical: these include more efficient combustion processes and optimized downspeeding/downsizing concepts as well as the reduction of specific internal friction. Although the friction of the individual frictional pairs in the combustion engine has already attained a very low level, there is definitely more potential for optimization in this regard [1], [2].

In order to optimize the friction of individual components, it is ultimately necessary to furnish experimental proof – under the most real conditions possible – of savings potentials that are generally determined on the basis of simulations. For the first stage of this work, component test benches are available for qualitative comparisons of individual models. With this approach, however, there are limitations on the most realistic possible modeling of boundary conditions such as those prevailing in a com-

bustion engine; unfortunately, this means that statements about absolute friction coefficients are bound to involve errors. For this reason, an assessment of optimization measures on the components also requires an investigation covering the entire engine. Various methods (some of them relating to specific components) have been developed for this purpose over time. All of them have one thing in common: because of the steady reduction in friction over the years, they require measurement technology of increasing accuracy and higher resolution in order to supply reproducible results; they must also be able to reliably capture even small changes in the respective measurand.

We shall begin with an overview of some methods for measuring friction in combustion engines. Then, we shall use the example of friction measurement with the indication method to examine the required and attainable degree of accuracy for measurement technology nowadays (indication and torque measurement), as well as the associated potential errors that must be taken into account. In addition, we shall present a new approach to measuring frictional energy losses with external charging under motored conditions.

Methods of Measuring Frictional Power

Over the years, a wide variety of experimental measuring methods have been developed to determine frictional power in combustion engines. These can roughly be divided into two categories: "complete engine" and "single component" or "subsystem". Depending on the particular problem under consideration, each of the different methods has advantages and drawbacks, which means that a balanced decision has to be reached about the most appropriate method for the problem in every case. Some of the most important and historically relevant methods are detailed below.

Frictional Power Measurement of Single Components [3] [4]

Single-Component Test Bench

In principle, most components of a combustion engine can also be examined individually to determine their frictional behavior. For this purpose, single-component test benches are usually available. Their tasks include modeling the boundary conditions prevailing in an engine, ranging from temperatures, pressures and warpage to loads and lubrication conditions. For components that do not belong to the engine itself (such as ancillary units), a test bench of this sort can be designed with reasonable outlay. For the components of the engine block as such, however, the outlay usually exceeds the available resources. This means that accurate modeling of the boundary conditions has to be restricted or simplified, with the ultimate effect of limiting the informative value of the measurement results.

Measuring the Frictional Power of Single Components of the Complete System

The frictional power of individual engine components can also be determined in the complete system. In particular, several methods are available for the "pistons/piston rings/liner" frictional pair. These include, for example, the Instantaneous IMEP and Floating Liner methods.

Instantaneous IMEP

In this measurement method, both the cylinder pressure curve and the connecting rod strains have to be measured over time with high resolution. This method can be used in real, fired engine mode. The forces present on the connecting rod can be derived from the connecting rod strains; then, with the help of the cylinder pressure, a balance of forces on the piston can be compiled. To transmit the strain signals from the connecting rod, use is generally made of a grasshopper linkage which has the effect of limiting speed and runtime. Additionally, a simulative or experimental determination of the stress-strain correlation is

required for the selected measuring points on the connecting rod. This method can achieve good results, but with major effort and outlay.

Floating Liner Method

This measuring method also allows operation in fired engine mode. For this purpose, the liner is "float-mounted" in the cylinder block, and is supported by force sensors. A particular challenge is posed here by sealing the gap that is necessary between the liner, cylinder head, water jacket and crankcase. The forces measured on the liner result primarily from the friction between the pistons and/or piston rings and the liner. On this basis, with the help of the piston's stroke profile, the frictional power can be calculated. However, this method requires very extensive mechanical interventions and modifications to the engine. The Floating Liner method can also be expected to yield good results.

Frictional Power Measurement on the Entire Engine [3] [4]

Coast-Down Tests

If the mass moments of inertia of the rotating and oscillating masses in the complete system are known, conclusions about the acting friction can be drawn from the decrease in speed as the engine coasts down freely. For this purpose, the engine (in fired mode) is accelerated to constant speed and after a steady condition is attained, the fuel supply is turned off. The advantages of this method are that it is relatively easy to implement and – in certain cases – there is no need for a dynamometer. As a disadvantage, it should be noted that in the absence of indication, both the gas exchange losses and the thermo-dynamic losses during compression and expansion are included in the frictional power. The quality of the results is heavily dependent on the accuracy of the mass moments of inertia of all the engine's moving components; outlay for this method can be classified as low.

Willans Line

The torque attained for a constant speed is plotted versus the fuel consumption in each case. If the profile is extrapolated to fuel consumption of zero, the intersection with the ordinate gives a theoretical frictional torque for the respective speed. However, this can only be described as an approximation because the linear extrapolation procedure assumes a constant ratio across all load points between the fuel energy converted into mechanical work and the total fuel energy. This method also has the advantage of being relatively simple to implement. Two disadvantages should be mentioned: low accuracy, and the fact that – without indication – the gas exchange losses are also included.

Cylinder Deactivation

After a steady, stationary operating point has been attained in fired mode (n/α regulation), one or more cylinders are deactivated. In theory, the change in measured engine torque corresponds to the "internal torque" of the deactivated cylinders. On this basis, the frictional torque of the entire engine can be determined by forming the differential. Here too, if there is no indication, there is no possible way of recording the gas exchange losses separately. Furthermore, on engines with a small number of cylinders, deactivation of cylinders results in running "out of round" – which in turn places higher demands on the torque measurement.

Measurement of Friction under Motored Conditions

In order to measure friction under motored conditions, an engine is kept at a constant speed by an external drive until the temperatures in the oil and water have stabilized, and the friction mean effective pressure is then determined from the necessary drive torque. In this case too, without indication, the gas exchange and thermodynamic losses during compression and expansion are included in the measured friction. In order to obtain a meaningful result when using this method, therefore, it is necessary to determine the indicated mean effective pressure with the help of accurate indication. The friction mean effective pressure is then obtained from the measured variables, i.e. torque and indicated mean effective pressure, by means of the known relationship $p_{mr} = p_{mi} - p_{me}$. In contradistinction to the other measurement methods for the entire engine, a dynamometer is no longer adequate with this method; instead, an active drive is required (usually, an electromotive drive). However, this method can be expected to deliver the best results – provided that the measurement technology is good.

Measuring Friction according to the Indication Method

As a general rule, the procedure for measuring friction under motored conditions can also be used in fired mode. However, this places even higher demands on the measurement technology and on the technical capabilities of the test bench itself. These include general boundary conditions as regards the fuel supply (ideally: external, high-pressure) and exhaust extraction as well as high-precision torque measurement equipment with wide measuring ranges. Furthermore, there is a substantial increase in outlay on conversion work to compare variants, which significantly extends the duration of the testing.

A New Approach: the Integral Lateral Force Method

In principle, it may be said the influences on engine friction due to the absence of combustion at the time of the measurement

are not taken into account in any of the aforementioned methods.

Particular mention should be made here of the effects due to the lack of cylinder pressure and the high combustion temperature. The combustion pressure in the cylinder has a direct effect on the piston (through its lateral force) and on the friction on the piston skirt; it also has an effect on the bearing loads in the main and connecting rod bearings, through the kinematic coupling of the piston with the connecting rod and the crankshaft. The thermal effects of combustion have a particular impact on the friction in the cylinder itself. Temperatures on the surface of the cylinder wall are usually higher than in unfired mode, with direct effects on the oil film (in particular as regards its viscosity and load-bearing capacity). Generally speaking, the piston itself also exhibits higher temperatures and temperature gradients, leading to changes in warpage and expansion behavior.

For measurements of friction under motored conditions, an attempt can be made to model the effect of the lack of cylinder pressure by means of external charging of the engine. The first investigations of this sort took place back in 1939 [5]. However, only a limited improvement in the results was obtained from past experiments in which the charging pressure was selected so that the peak pressures for a comparable fired operating point were attained [6]. The publication shows that engine friction is underestimated if charging is used to attain fired peak pressure. One reason for this is the difference in the position of the pressure curves in relation to the crank angle.

Under motored conditions, the pressure maximum is very close to the top dead center. In these ranges, however, the piston hardly moves and the connecting rod inclination is very small; despite the high pressures, therefore, very little useful work or frictional work is done. In crank angle ranges at high piston speeds and with large connecting rod inclinations, the pressure is already significantly lower; consequently, the product of force and displacement is also small (Figure 1).

In fired mode, the pressure maximum occurs later and, even after top dead center, the pressure does not fall immediately to the same extent as under motored conditions. It is only after combustion is nearly complete that the negative pressure gradients rise above those of the pressure curves under motored conditions, and the fired pressure approaches the charged cylinder pressure under motored operating conditions (Figure 1). The allocation of the frictional work for the entire engine to its individual components (Figure 2, Figure 3) shows that the frictional pair "piston/piston ring/liner" with connecting rod accounts for the

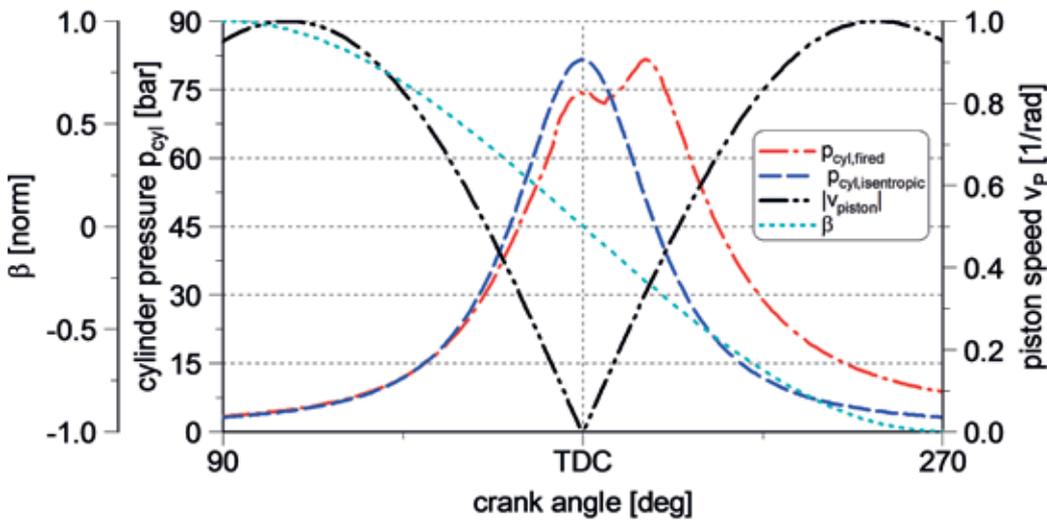


Fig. 1: Pressure curves, their gradients, the normalized value for piston speed and the normalized connecting rod tilting angle β at 2 400/min and 100 Nm

greater part (about 50 %). In particular, the piston friction results from the forces applied on the liner (lateral piston force). If we consider the other components that contribute to the total friction of the combustion engine (Figure 2), it can be assumed that in the case of the oil pump, the Lanchester balancer and the belts that the combustion and the peak cylinder pressure have only an extremely small influence on frictional behavior (for externally conditioned engines). As regards the crankshaft and the valve train, it may also be assumed that most of the friction of these components depends on other operating parameters (oil and water temperatures, and their influence on bearing play, oil pressure, etc.); and that fired mode only increases the bearing loads for the crankshaft and the connecting rod. Under motored conditions, however, external charging also produces an approximate replication of this effect.

This will be taken as the basis for the following examination of the idea of selecting the charge pressure under motored conditions so that the same integral lateral piston force is present over the high-pressure stage of a combustion cycle. We aim to examine whether this approach can produce an acceptable reduction in the difference between the frictional work at a fired engine operating point and that for a motored, externally charged operating point.

Assuming a constant coefficient of friction within the frictional pairs, this approach corresponds to the friction work done by the piston over the high-pressure stage of a combustion cycle. In both fired and motored mode, the same accelerations act on the piston at the same speeds, so the influence of inertia forces can

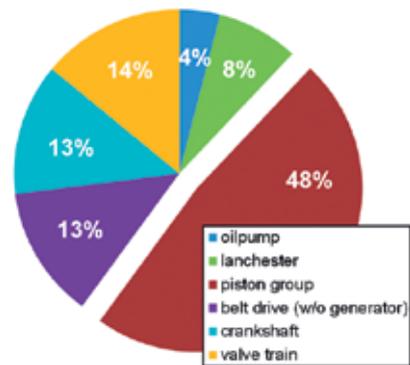


Fig. 2: Frictional power distribution of a Daimler 4-cylinder spark ignition engine according to [1]

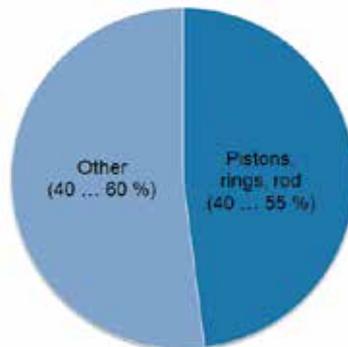


Fig. 3: Distribution of mechanical friction losses in the entire engine, according to [4]

be neglected. Likewise, the frictional force acting against the movement of the piston as such should not be taken into account, due to the unknown coefficient of friction. Consequently, only the gas forces acting on the piston need to be considered. In the case of the motored operating point, no pressure curve is known. For this reason, an isentropic pressure curve with pressure p_0 is assumed in the gas exchange BDC. Equation 1 describes the lateral piston force induced by gas forces in fired engine mode; equation 2 describes this force under motored conditions:

$$F_{N,f} = ((p_{cyl}(\alpha) - p_{cc}) \cdot A_p \div \cos(\beta)) \cdot \sin(\beta) \quad (1)$$

$$F_{N,m} = ((p_0 \cdot \left(\frac{V_{max}}{V(\alpha)}\right)^{\kappa} - p_{cc}) \cdot A_p \div \cos(\beta)) \cdot \sin(\beta) \quad (2)$$

Integration of the product of these lateral forces, of the piston speed in relation to the crank angle

$$v_p(\alpha) = \frac{ds_p}{d\alpha}$$

and of the friction coefficient across compression and expansion produces the frictional work theoretically done in the respective high-pressure stage (equations 3 and 4).

$$E_{FN,f} = \oint |F_{N,f}(\alpha) \cdot v_p(\alpha) \cdot \mu| d\alpha \quad (3)$$

$$E_{FN,m} = \oint |F_{N,m}(\alpha) \cdot v_p(\alpha) \cdot \mu| d\alpha \quad (4)$$

The approach examined here requires an equal energy balance. Due to the simplification in respect of the friction coefficient, the "hidden" gas exchange and the absence of the combustion temperature, the relationships presented here do not include all the relevant effects such as warpage, thermal expansion and varying lubrication conditions. In addition, therefore, a constant, engine-specific calibration factor must be introduced. This results in equation 5, which is the basis for this approach.

$$E_{FN,m} \stackrel{!}{=} C \cdot E_{FN,f} \quad (5)$$

All available indication software integrates a function for real-time determination of the peak pressure, so it is possible to set the charge pressure such that the calculated isentropic final compression pressure is reached. The target variable is formulated in equation 6:

$$p_{max,target} = \max \left(p_0 \cdot \left(\frac{V_{max}}{V(\alpha)}\right)^{\kappa} \right) \quad (6)$$

The section "Results on the Basis of the Integral Lateral Force Method" presents the results achieved with this method.

Indication and Measurement Accuracy

The last 18 years have seen a constant increase in the attainable torque and therefore in the power of combustion engines used in passenger cars. If we consider the progression of mean effective pressures over these two decades, we find increases of about 130 % for spark ignition engines and about 85 % for diesel engines which, due to their operating principle, exhibit higher mean effective pressures (Figure 4).

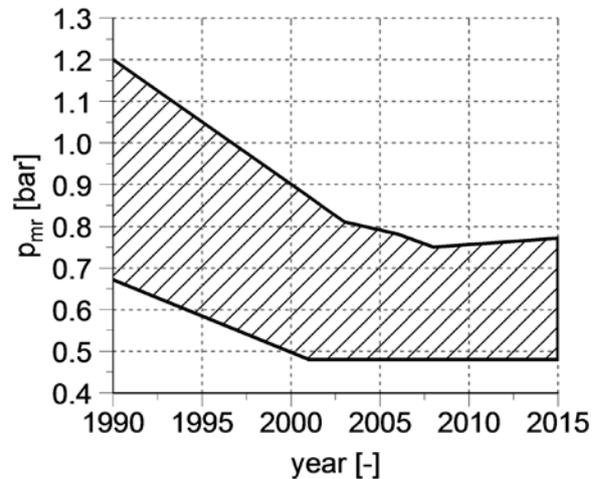
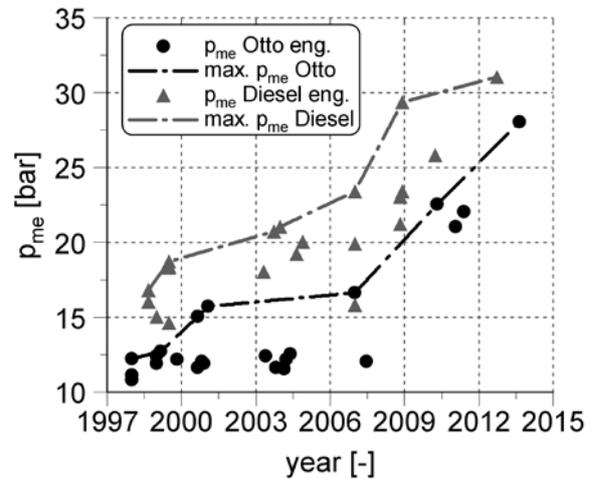


Fig. 4: Mean effective pressures of selected spark ignition and diesel engines introduced since 1998 Friction mean effective pressures for charged and uncharged spark ignition engines, derived from [7]

The trends for the friction mean effective pressures of the two engine concepts are moving in opposite directions. Since 1990, the friction mean effective pressures for spark ignition engines have decreased by about 40 % (Figure 4) [7]. This trend towards higher specific power with a simultaneous decrease in power loss ultimately results in higher efficiency levels and lower CO₂ emissions.

But on account of this trend, there has also been a constant increase in the standards expected of the technology used for power, friction and indication measurements. For the measurement of effective torque and high-pressure indication in particular, there has been a significant increase in the effects of measurement errors when measuring frictional power. The measuring ranges required for the hardware have also increased, and the same is true of the loads to which the hardware is exposed (temperatures, pressures, torques and their gradients).

For work according to the indication method ($p_{mr} = p_{mi} - p_{me}$), the ratio of friction mean effective pressure to indicated or mean effective pressure (in the full load) was still about 1 to 15 in the late 1990s. A measurement error of only one percent in the p_{mi} therefore resulted in an error of about 15 % in the p_{mr} . This ratio has now more than doubled, resulting in an error of some 30 % as regards the friction mean effective pressure.

Over the years, there have also been steady increases in the mechanical and thermal loads to which a pressure transducer is exposed in the combustion chamber of a combustion engine. Modern cylinder heads are subject to high thermal loads, leading to the possibility of warpage that will in turn introduce mechanical tensions into the sensor via the mounting bore, thereby influencing the measurement result. If, moreover, the sensors are mounted flush with the combustion chamber in highly-charged spark ignition and diesel engines, the thermal loads as such result in measurement errors that cannot be neglected unless good heat dissipation is present. This makes it necessary for the measuring technology deployed to be as precise as possible; it must feature high absolute accuracy as well as very high repeat accuracy and high load-bearing capacity, especially for comparisons of variants.

By mounting the sensors in accordance with the manufacturer's recommendations, negative thermal and mechanical influences can both be reduced.

The technical specifications of a piezoelectric pressure transducer (Table 1) that is compliant with the current industrial standard show that such a device can already contribute to a thermal shock error of up to one percent in the p_{mi} .

Manufacturer	Kistler
Model	Type 6041B
Δp [bar]	$\leq \pm 0,25$
Δp_{mi} [%]	$\leq \pm 1,0$

Table 1:
Thermal shock error for a selected pressure sensor [8]

However, the complete measuring chain for pressure indication usually consists of a sensor, a charge amplifier and an A/D converter, none of which exhibit absolute accuracy. The typical measurement error for a charge amplifier corresponding to the current state-of-the-art is quoted as 0.1 %; for the latest A/D converters, resolutions and measurement deviations are within ranges that are irrelevant for indication purposes. If the mechanical influences on the measurement technology are neglected, errors of up to 1.1 % occur across the complete measuring chain. When calculating the balance, these errors are allocated to the friction mean effective pressure, which is therefore subject to a deviation of up to 33 %.

Another critical factor in every engine indication procedure is the determination of the correct phasing of the high-pressure signal, i.e. the position of the captured measuring points in relation to the position of the crankshaft and also, therefore, to the piston position. This is the only way of correctly determining the change in the combustion chamber volume that is present in each case, and hence the work done via the line integral $p_{mi} = \oint p dV$. A relatively small phasing error already leads to noticeable changes in the indicated mean effective pressure. Figure 5 shows the resultant error in the friction mean effective pressure for a deviation of -2 to 2° of crank angle in the phasing, as a function of the compression ratio.

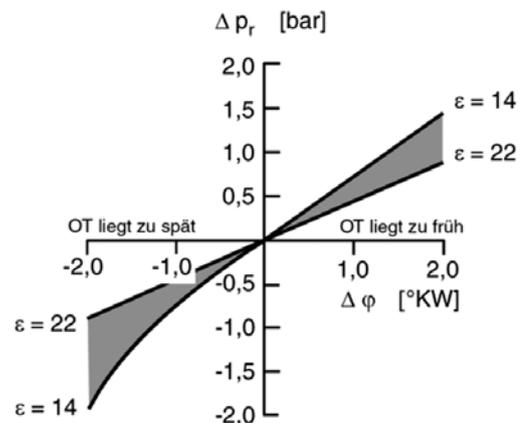


Fig. 5: Frictional power error in case of pressure signal phasing deviation [9] (graph texts: TDC is too late – TDC is too early)

So that this phasing can be determined as accurately as possible, the top dead center of the piston is usually identified and its offset in relation to the triggering mark of the crank angle encoder is determined. Various methods are available for this purpose; they entail greater or lesser outlay, and the quality of their results also differs. Outlay and accuracy for the most commonly used methods are compared in Figure 6. It is evident that greater accuracy always entails more outlay. Nevertheless, as Figure 5 shows, the most accurate possible determination of the top dead center of the engine under investigation is essential for the frictional power measurement.

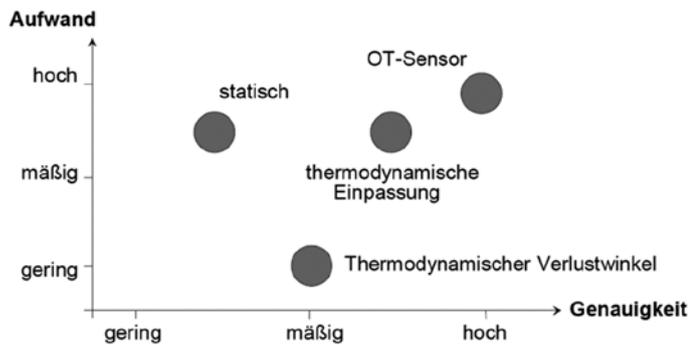


Fig. 6: Outlay and accuracy of different methods to find the TDC [9]

In addition to the absolute position of the top dead center, the stability and repeat accuracy of the crank angle measurements are also factors that determine the quality of comparative measurements. If there are slight shifts of the crank angle sensor in relation to the crankshaft during operation or during the runtime of the measurements due to inadequate stability, vibrations or excessive play in the sensor holder for the angle encoder, this will already produce deviations in the measurands above the level of repeat accuracies in the rest of the measurement technology. For example, Figure 7 shows the error in friction mean effective pressure with an unstable TDC position for the test vehicles used below, at an operating point of average load and speed

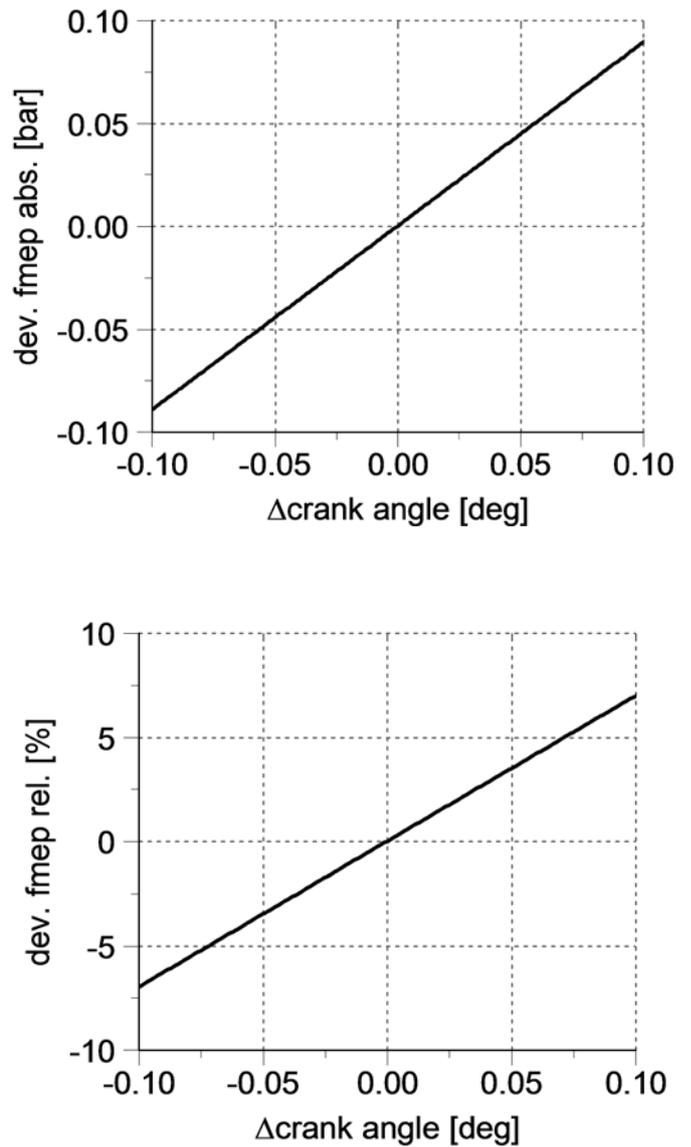


Fig. 7: Error in frictional power due to unintentional TDC shift on the test vehicle (n = 2 400\min; pme=11.9 bar)

Experimental Setup

The investigations were carried out on a current inline four-cylinder diesel engine (Table 2)

Displacement (cylinder capacity)	1.5 l
Power	88 kW
Valves	8
Valve train	SOHC

Table 2:
Key data for the test vehicle

The oil and water pumps were replaced by external conditioning devices. This made it possible to regulate the desired media temperatures to ± 0.5 K in each case. Accordingly, the charge air temperature was also set to ± 0.5 K by means of a water-air heat exchanger. The drive torque of the high-pressure fuel pump was determined from the available mapping data, and was taken into account in the frictional torque.

All the tests were carried out on a test bench that was designed specifically for investigations of friction under motored conditions, but which is also suitable for fired engine mode. The torque measurement was carried out with a precise torque measuring flange. A static examination of the flange, repeated over several days, showed very good measurement and repeat accuracies (Figure 8).

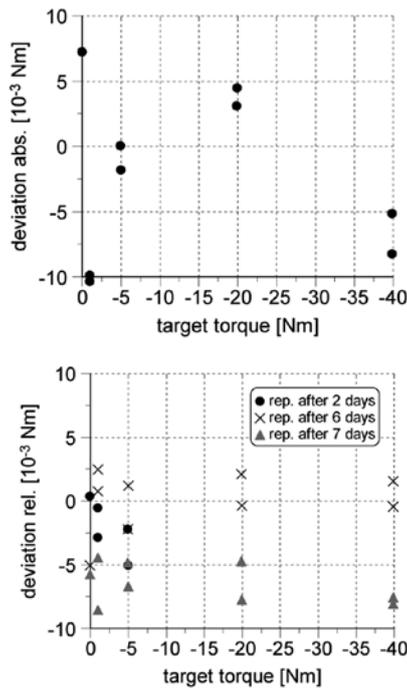


Fig. 8: Absolute accuracy and repeat accuracy of the measuring flange used, under static load

If, in this context, we assume the average New European Driving Cycle (NEDC) data for the test vehicle used, taking an average speed of 1,500 – 2,000 rpm in the cycle, the absolute deviations would correspond to a difference of about 0.05 g CO₂/km. After conversion, the absolute deviations would correspond to an error of less than 0.001 bar in the mean effective pressure. It should nevertheless be stated explicitly that these are static accuracies.

The pressure sensors used for indication are water-cooled piezoelectric M8 sensors from Kistler, Type 6041B. The glow plug bores were modified and widened for installation in the cylinder head (Figure 9).

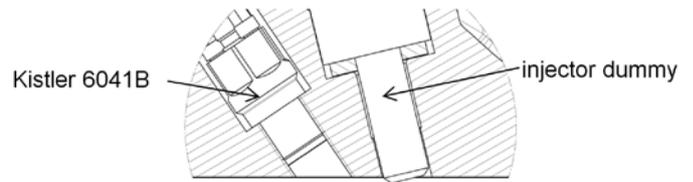


Fig. 9: Installed positions of the pressure sensors in the cylinder head

The charge amplifiers used were Kistler Type 5064, operated with drift compensation and -8 V offset. At the start of the measurements, the entire measuring chain (pressure transducer \rightarrow charge amplifier \rightarrow A/D converter) was calibrated dynamically using a pressure signal, imprinted with a pressure compensator, which rose and fell again in ramp form. Prior to the actual calibration, the sensors were also tempered for more than one hour at 50 °C, using a Kistler conditioning system Type 2621F that was also used for all the other measurements. The calibrations that were performed showed good linearity and low hysteresis for the measuring chain, and also very little deviation from the factory calibration values for the sensors. The hysteresis profile for the piezoelectric sensors for one calibration run is plotted versus the pressure of the reference sensor in Figure 10.

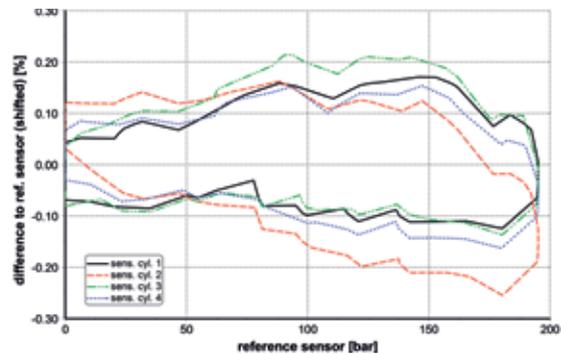


Fig. 10: Hysteresis curves for the entire measuring chain, for the pressure sensors used, in relation to 200 bar

The TDC was located with the use of a TDC capacitance probe (Kistler Type 2629B), and the procedure was also repeated over several days. The results achieved show very high reproduction accuracy (Figure 11), with a maximum deviation of 0.03 bar and an absolute mean deviation of 0.01 bar in the friction mean effective pressure. For this purpose, half of all the fired measurements were repeated at random intervals of between one and 17 days. In this case, the reproduction measurements cover all the speeds that were investigated, and they supply a cross section of the engine's entire load spectrum.

The maximum relative deviation of the friction mean effective pressure within the reproduction measurements is approximately 2.5 %; and in the quantity-based mean, it is 1.1 %. In relation to the indicated mean effective pressure, this corresponds to a maximum deviation of about 0.4 % and a mean deviation in relation to quantity of 0.2 %. It is evident here that the repeat accuracy of the sensors used is significantly less than the maximum permitted error rate on the data sheet. It should also be noted that the measurement errors for both the torque measurement and the high-pressure indication are incorporated into these results.

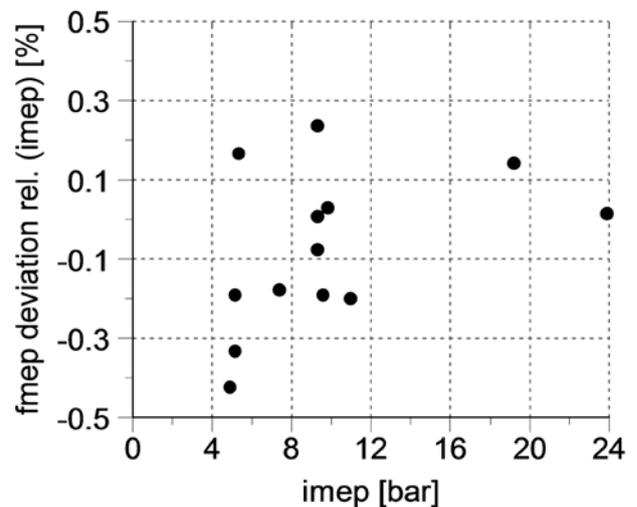
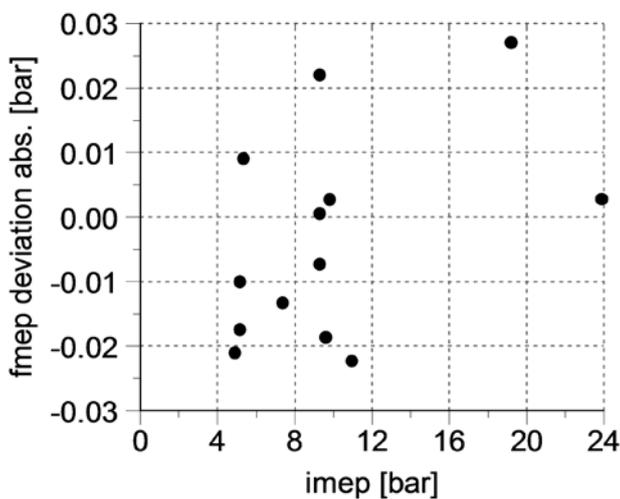
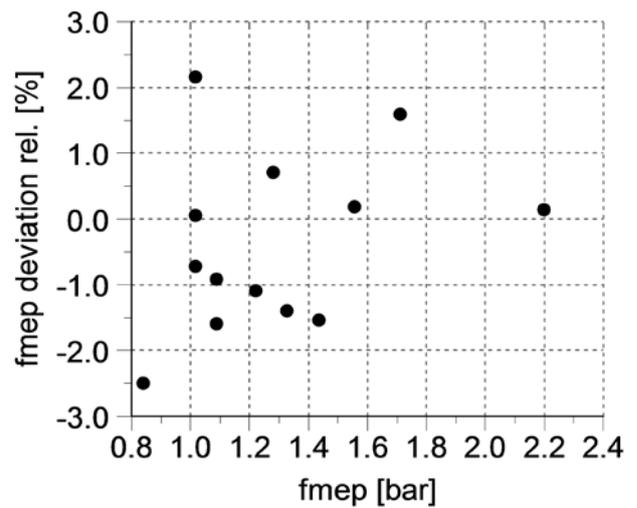
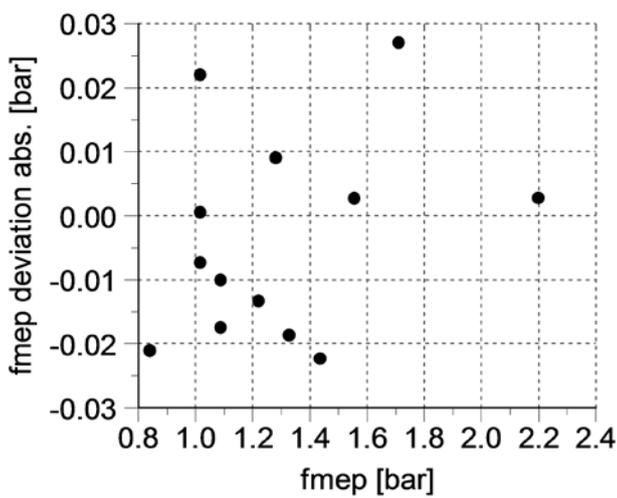


Fig. 11: Spread of friction mean effective pressure across all reproduction measurements (fired) in relation to the friction mean effective pressure and indicated mean effective pressure

Results on the Basis of the Integral Lateral Force Method

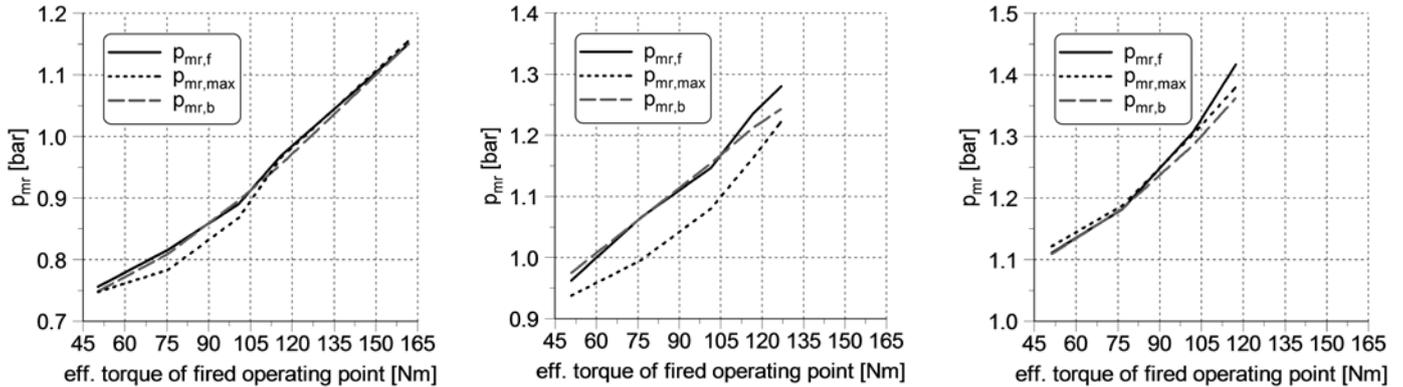


Fig. 12: Measured frictional power at 1 500 rpm (top left), 2 400 rpm (top right) and 3 000 rpm (below) in fired mode ($p_{mr,t}$), with external charging according to the peak pressure method ($p_{mr,max}$) and the newly presented method ($p_{mr,b}$)

For the purposes of the present work, an initial limitation to cycle-relevant operating points was applied (1 500 ... 3 000/min). $C = 62\%$ (cf. eq. 5) was determined at one operating point as the calibration factor for the test setup that was used, and this was adopted for all further measurements. All the media (water, oil, charge air) were conditioned according to the fired measurements, with waiting periods until steady conditions were obtained. The following figure (Figure 12) shows the results from the integral lateral force method for the three speeds ($p_{mr,b}$) examined versus the effective torque of the fired operating point to be modeled in each case ($p_{mr,t}$). The profile of the friction mean effective pressures is also plotted for a charge level set after the peak pressures of combustion ($p_{mr,max}$). The resolution of the ordinates for all three graphs was selected so as to guarantee good comparability of the differences.

It is evident that over large parts of the range examined, the friction mean effective pressures are underestimated by the peak pressure method. Charging in accordance with the method presented here supplies noticeably more precise results. Even in areas where the peak pressure method supplies good results, no deviations or only minor deviations can generally be identified within the spread of the fired measurements when the integral lateral force method is used. Around 120 Nm, a slight disadvantage for the new approach can be identified on the engine that was examined; as the load increases, however, this disadvantage disappears again (cf. Figure 12 top left).

In the integral lateral force method, the effect of the cylinder pressure is generally dependent on the inclination of the connecting rod. Towards the crank angle position at maximum lateral piston force, the effect of higher cylinder pressure on the theoretical frictional force is therefore disproportionately greater (also cf. Fig-

ure 1). With constant mean effective pressure and increasingly later and longer combustion, the necessary charging level therefore increases slightly with this method, although there is a decrease in the peak combustion pressure at the respective operating point.

A theoretical consideration of this effect is supplied by Figure 13 and Figure 14 which, respectively, represent: the theoretically expended friction energy which increases over the crank angle; and the peak combustion pressures and the peak pressures calculated according to the integral lateral force method for a real six-cylinder spark ignition engine. The calibration factor for the integral lateral force method was assumed as a value of $C = 100\%$ (cf. eq. 5), so the absolute position of the curve is not binding.

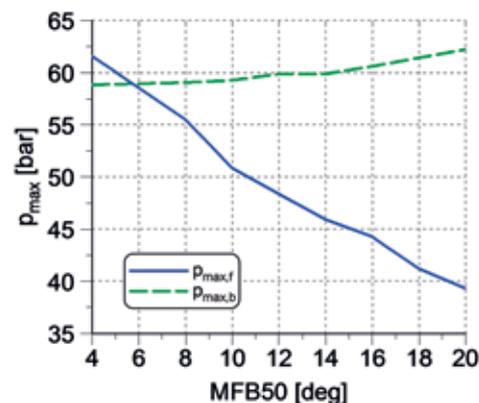


Fig. 13: Peak pressure required according to the integral lateral force method ($p_{max,b}$) and the peak pressure method ($p_{max,t}$) over the Mass Fraction Burned (MFB50) with constant p_{me} ($C = 1$) (cf. eq. 5)

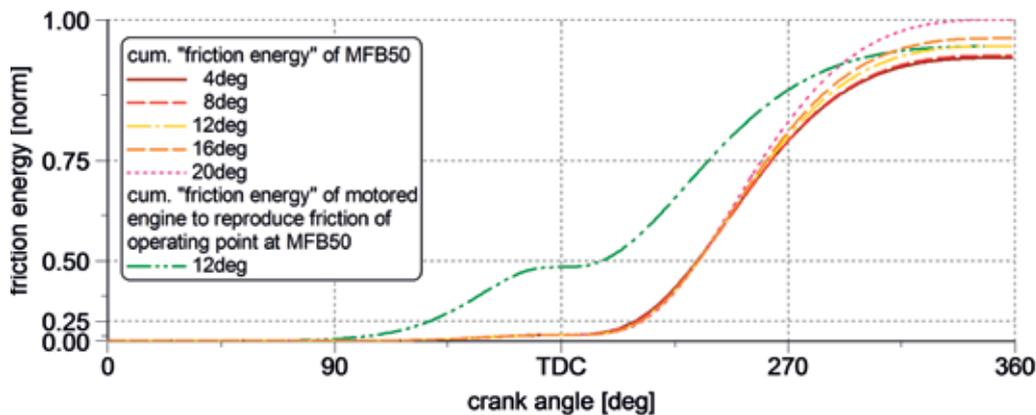


Fig. 14: Increasing frictional energy for different MFBs and precalculated profile for a selected motored operating point $n = \text{konst.}; p_{me} = \text{konst.}$

Figure 14 shows clearly that the frictional work in charged engine mode under motored conditions in the isentropic case is symmetrically distributed around the top dead center, and is not per-

formed mainly towards the ignition TDC, as is the case in fired mode.

Summary

Increasingly accurate measurement technology is required to resolve the problems of determining the pmr by forming the differential between p_{mi} and p_{me} , whose values have increased steadily over recent decades, whereas the values for friction mean effective pressure show a constant decrease. Even today, indication is still responsible for the largest error in the overall balance. The absolute accuracy and the repeat accuracy of the torque measurement are less by about one power of ten than the values for the indication. If the indication is split into determination of the crank angle and recording of pressure, a small error in the crank angle measurement ($\Delta\alpha < 0.1^\circ\text{CA}$) will already cause substantial deviations of the integral variables, such as the friction mean effective pressure. The use of unsuitable sensor technology to capture the cylinder pressure may also cause errors of the same magnitude. When planning and performing indication measurements in order to determine frictional power, it is therefore necessary to ensure optimal conditions for the application and the hardware deployed. By selecting suitable pressure transducers and a robust method of determining the crank angle, a particularly decisive influence can be exerted on the quality of the results. However, it also became apparent that the reproduction quality of the sensor technology used here was significantly better than might have been expected from the technical specifications ($< 0.5\%$ of IMEP and/or $< 3\%$ of FMEP).

Using the integral lateral force method, the relative deviation between fired and motored friction mean effective pressure in the cycle-relevant range could be lowered from 2.72 % to 1.17 %, as compared to previous measurements of friction under motored conditions with external charging according to the peak pressure method. There were minor disadvantages for the new approach in only a few operating points; in relation to the peak pressure method, however, these were already at the same level as the spread of the fired reference measurements. Until now, this method has only been examined for certification-relevant operating points up to average loads. For the selected test engine, the power of the high-pressure fuel pump could only be taken into account on the basis of mapping data. Minor uncertainties cannot be ruled out here. An externally driven high-pressure injection system in fired mode would further improve the quality of the results. Basically, however, the new approach can be seen to have the advantage. More extensive investigations with higher loads and additional test vehicles are nevertheless required in order to arrive at definitive conclusions.

List of Symbols Used in Formulas /Abbreviations

Symbols used in formulas

α	crank angle
A_p	piston area
β	connecting rod tilting angle
C	calibration factor
$E_{FN,f/m}$	theoretical frictional work (fired/motored)
$F_{N,f/m}$	lateral piston force (fired/motored)
K	isentropic exponent
n	speed
P_0	pressure at start of isentr. compression
P_{cc}	pressure in crankcase
P_{cyl}	cylinder pressure
$P_{max,target}$	final compression pressure according to the int. lateral force method
P_{me}	mean effective pressure
P_{mi}	indicated mean effective pressure
P_{mr}	friction mean effective pressure
$P_{mr,b/f/max}$	friction mean effective pressure according to the: <ul style="list-style-type: none"> • integral lateral force method • fired engine mode • peak pressure method
μ	coefficient of friction
V	cylinder volume
V_{max}	maximum cylinder volume
V_p	piston speed

Abbreviations

BMEP	Break Mean Effective Pressure (p_{me})
FMEP	Friction Mean Effective Pressure (p_0)
IMEP	Indicated Mean Effective Pressure (p_{mi})
CA	Crank Angle
MFB50	50 % Point of Mass Fraction Burned
NEDC	New European Driving Cycle
TDC	Top Dead Center
BDC	Bottom Dead Center

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