White Paper

Influence of crankshaft torsion on cylinder pressure diagnostics

Abstract
Cylinder pressure diagnostics on engines with long or flexible crankshafts require particular attention as regards the assessment of results dependent on the crank angle. This article examines the identification and verification of measurement and analysis methods in greater detail.

The basis for any indication is highly precise and synchronous measurement of cylinder volume and cylinder pressure. Given that no simple process for direct measurement of the cylinder volume is available at present, a crank angle encoder is normally used to measure the crankshaft position. The cylinder volume is calculated with the help of the basic kinematic equation of the piston engine. Since the crankshaft torsion cannot be taken into account with this method, the calculated key indication values are bound to be subject to errors.

Measurements of crankshaft torsion using angle sensors positioned on both sides of the crankshaft have shown that the torsion angle can assume significant proportions in the case of large engines. With this method, experimental verification of the crankshaft simulation is only possible at these two reference positions. The deployment of a new type of optical measurement technology in the combustion chamber has made it possible to measure the gas exchange TDC in fired operation with adequate accuracy on all cylinders. This allows verification of the simulation and correction of the key indication values for each individual cylinder on the basis of the adapted crank angle values. Comparison of the basic indication and the corrected indication values determines the expected error for engines in this category.

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Content

measure. analyze. innovate.

1. Introduction and motivation

In addition to high-precision measurement data from cylinder pressure indication systems, ever greater requirements regarding thermodynamic analysis in engine development processes primarily demand accurate knowledge of respective cylinder volumes during fired operation.

The prerequisite for precise calculation of the cylinder volume is a suitably exact measurement of the crank angle. The significance of a crank angle deviation in thermodynamic analysis is depicted in Figure 1. The left diagram shows the basic influence of a crank angle error on the energy balance of a large gas engine, while the graph on the right shows the influence of an angle error on the indicated mean effective pressure (IMEP).

A piston top dead center (TDC) position that is too early, giving a shift to the right on the cylinder pressure curve, leads to insufficient pressure when the piston ascends and excess pressure as it descends. This results in apparent prolongation of the post-combustion phase and increased energy conversion. The circumstances are reversed when TDC is too late. The effects on angle offsets can also be seen in terms of IMEP, which becomes greater when TDC is too early and vice-versa. These effects are accompanied by higher or lower friction mean effective pressures (FMEP). If a maximum permissible deviation is defined as ±0.1 bar for IMEP and ±1% for the energy balance, then a precision requirement for TDC allocation of ±0.1° CA can be derived.

Traditionally, TDC determination is based on calculations and measurements, either when the engine is at rest (static TDC determination) or during non-fired engine operation (TDC determination via thermodynamic loss angle or by means of a capacitive sensor), as described for example in [1, 2]. The method of thermodynamic adaptation is the only one that can also be applied in fired engine operation. It is based on a calculation of the pressure curve during the phase without combustion, and it assumes precise knowledge not only of the charge mass but also of heat transfer and leakage. The entire high pressure component can then be used in non-fired operation; in fired operation, however, the maximum is limited to the range from intake valve closure to onset of combustion.

In addition to the frequent non-availability of gas exchange simulation, which is a prerequisite for calculating the cylinder-specific charge mass, this method leads to uncertainties, particularly in connection with the assumptions regarding heat transfer. Moreover, extreme valve timings (such as the Miller Cycle, which involves very late intake valve closure) cause problems with determining TDC in fired engine operation because the available adaptation range becomes very small. In principle, TDC determination in fired operation with all of the methods listed above is possible only under certain conditions, or not at all.

The mechanically ideal engine takes into account neither manufacturing tolerances nor deformations caused by thermal or mechanical load. In the case of real engines, and particularly with large engines, considerable deviations from the ideal kinematics of the crank mechanism can arise in fired engine operation, due to the large dimensions and moving masses. The TDC values of individual cylinders may thus deviate from the reference point by a range of several degrees of crank angle. In addition, the piston stroke no longer corresponds to the ideal because crankshaft torsion changes over the crank angle. This results in a changed volume function. If such influences are to be taken into account and integrated into cylinder pressure diagnostics, new methods must be applied that are capable of acquiring the real behavior of the crank mechanism under load, and of generating statements based on verified crankshaft simulation results.

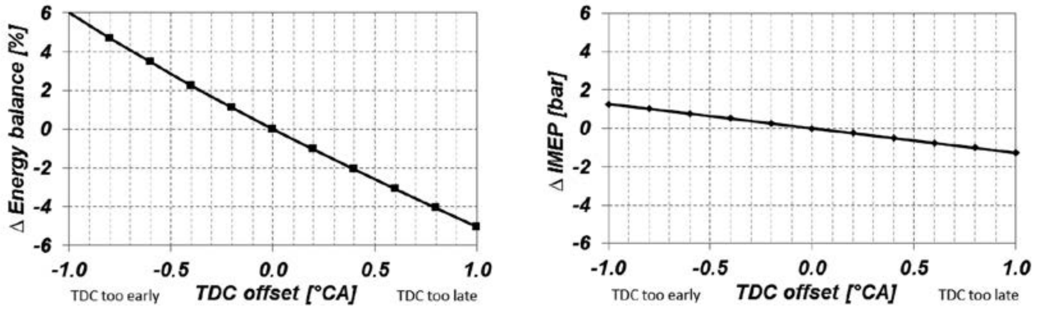

Figure 1: Influence of the crank angle error on energy balance and IMEP

2. Developed methodology for the integration of crankshaft torsion into engine process calculation

The method described below was developed and applied to a large-bore gas engine for the purposes of thermodynamic analysis, so as to take account of the effects of load-related influences on the crank mechanism. The selected approach uses detailed simulation results for crankshaft torsion, validated with the aid of various measuring methods as additional input data in the engine process calculation.

The starting point was a crank mechanism simulation with which the load-dependent torsion curves on the individual crankpins were calculated as a function of crank angle. The calculation results were validated with two different measurement methods in order to ensure their reliability. On the one hand, measurements were carried out to determine the total torsion of the crankshaft at its ends. On the other hand, an optical measurement system was used that was capable of determining the gas exchange TDC on all cylinders during fired operation. This allowed validation of the calculated, cylinder-selective torsion curves at a minimum of one point in each case. Finally, the results of real crankshaft behavior were implemented in the engine process calculation.

2.1 Simulating the crank mechanism

The computing power available today enables an extensive multibody simulation of the crank mechanism [3]. The primary objectives of this simulation are, among others: hardness analyses, determination of resonance frequencies, design of the torsional damper and improving noise, vibration and harshness (NVH). Modern calculation tools offer the possibility of calculating the torsion of individual crankpins, as described in [4]. This information can be used, for example, for cylinder-dependent definition of the ignition timing of an engine.

The crank mechanism is described using individual components such as crankshaft, connecting rod, piston, flywheel and torsional vibration damper. As the central structural element, the crankshaft is broken down into a sufficient number of individual masses, coupled by springs and dampers. The connecting rod is split into a rotating and an oscillating part. In the rotary oscillation model, the rotating part of the connecting rod is added to the corresponding crankshaft mass and the oscillating part is combined with the other masses moved by transmission (piston, piston pin, etc.), to form one oscillating mass (per cylinder). The oscillating masses can be added to the rotating masses using Frahm's approximation. The system is loaded or excited by cylinder pressure and the mass forces of the oscillating masses.

Under the influence of gas and mass forces, vibrations are imposed on the crank mechanism. The movement of these vibrations is described via a system of differential equations. As a result, the angle of torsion at a given computation point can be represented in relation to a reference point. This function is used to determine the torsion of the individual crankpins in relation to a reference point, such as the crank angle encoder. The results are shown in Figure 2.

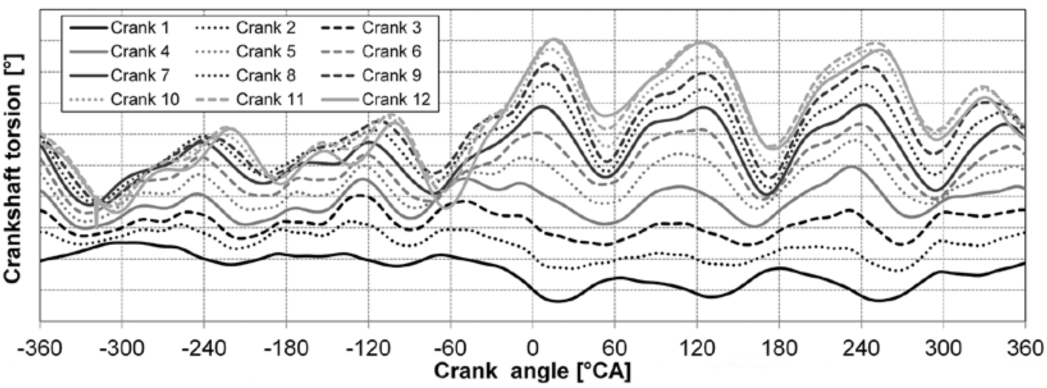

Figure 2: Angle of torsion of individual crankpins relative to a reference point

2.2 Experimental setup and measurement technology deployed

The experiments described in this article were carried out on a vee configuration, 24-cylinder large-bore gas engine (unitary displacement above three liters) from GE Jenbacher. The measurement plan included operating points across the entire load range of the test engine at a nominal speed of 1,500 min^{-1}.

In accordance with the method outlined above, the objective was to measure the relative torsion on the flywheel in relation to the crankshaft front end, in order to validate the calculation results from the multibody simulation. An optical crank angle encoder was used for this purpose, in the range of the free shaft end (on the rigidly connected part of the torsional damper). At the other end of the crankshaft (flywheel side) the existing tooth ring was used with a suitable pick-up sensor in order to determine the relative torsion of the crankshaft along its entire length, according to the method described in [5]. The signal of the encoder gave a resolution of 0.1° CA, by means of which the torsion angles of the individual flanks of the tooth ring were determined, forming the reference for the measurement as shown in Figure 3. As a result, crank angle-based data was available for relative torsion along the entire length of the crankshaft.

An optical measurement system based on light conductor measurement technology was used so that cylinder-specific measurement data could also be obtained. This made it possible to acquire the gas exchange TDC (GETDC) throughout the entire operating range [6]. Using this measuring system, the measurement program was carried out on each of the 24 cylinders in direct succession.

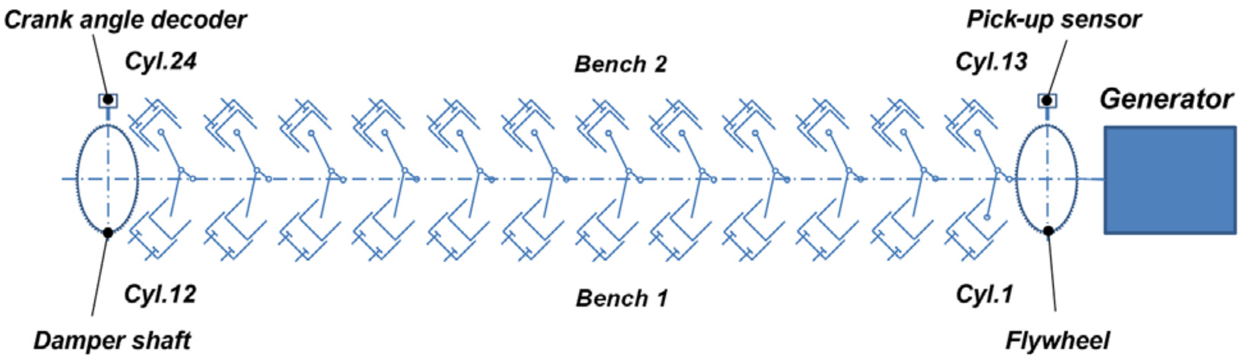

Figure 3: Schematic measuring setup to determine the overall torsion

The system projects a laser beam into the combustion chamber through an optical access point and detects the light reflected by the piston surface, as illustrated in Figure 4. The measured intensity of the reflected light is a dimension for piston position. This allows qualitative statements about the piston stroke based on crank angle, so TDC can be determined with the vertical section process [7]. A suitably robust structure enables unlimited operation of the measuring system, even during fired engine operation. For the measuring campaign, the optical TDC sensor was mounted in the cylinder heads, practially flush with the combustion chamber. Depending on the applicable prerequisites for the test engine, the GETDC can be determined with favorable accuracy (±0.2° CA) in fired engine operation, as can the ignition TDC (ITDC) to some extent. Criteria for determining ITDC in fired engine operation are, on the one hand, structural conditions (sensor to piston distance at TDC, angle alignment on the surface of the piston, etc.) and on the other, flame propagation in the "field of vision" of the sensor (flame type, intensity, etc.). On the large gas engine that was measured, verified determination of TDC under load was only possible in terms of GETDC.

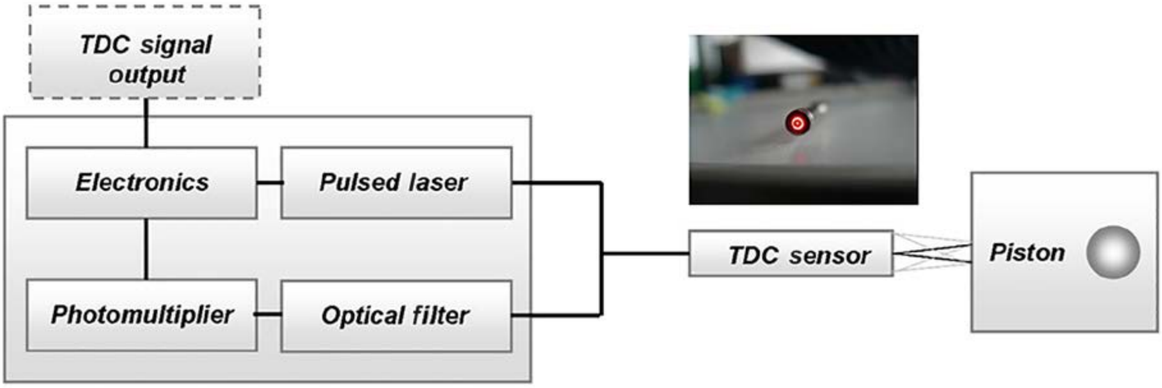

Figure 4: Basic structure of the optical measurement system, incl. frontal illustration of the sensor

3. Measured results

The test program included different load points at 1,500 min^{-1}, whereby all measuring points were approached a total of 24 times; this was because the optical TDC measurements were carried out sequentially on all cylinders with only one sensor.

3.1 Overall crankshaft torsion

During the measurement of relative torsion on the flywheel in relation to the free shaft end where the rotation angle sensor was positioned, the curves shown in Figure 5 for idling and full load were obtained. As can be seen, the maximum torsion of more than 1.5° CA appears along the entire length of the crankshaft at full load. The amplitude of the torsional oscillation reduces as the load declines. Observing the algebraic sign for relative torsion, it is evident that, by and large, the flywheel side lags behind the front end. Also depicted in Figure 5 is the simulation result of the total torsion of the crankshaft at full load: this exhibits excellent congruence with the measured results, not only in the absolute magnitude of torsion but also in the time curve over crank angle.

3.2 Measured results for GETDC

To structure the depiction of the GETDC values measured with the optical TDC sensor more clearly, Figure 6 shows separate presentations reflecting the engine's vee configuration, as Cylinder Bench 1 (cylinders 1 to 12) and Cylinder Bench 2 (cylinders 13 to 24). Once the crank angle encoder (reference basis) was installed on the front end in immediate proximity to the crankpin of cylinders 12 and 24, there was – as expected – no TDC offset caused by torsion on these cylinders. However, the greater the distance from the front end of the engine, the greater the measured GETDC offset becomes.

Observations show that the angles of torsion determined by the measurement technology increase progressively in the direction of the cylinders nearest to the flywheel. This behavior is clearly visible on both cylinder banks; the absolute angles of torsion on Bench 1 are generally somewhat greater than those on Bench 2. The reasons for this are structural, and they result from the mass relationships in the crank mechanism and the engine's firing order. The reduction in the TDC offset with decreasing load is also reflected clearly in the measurement data.

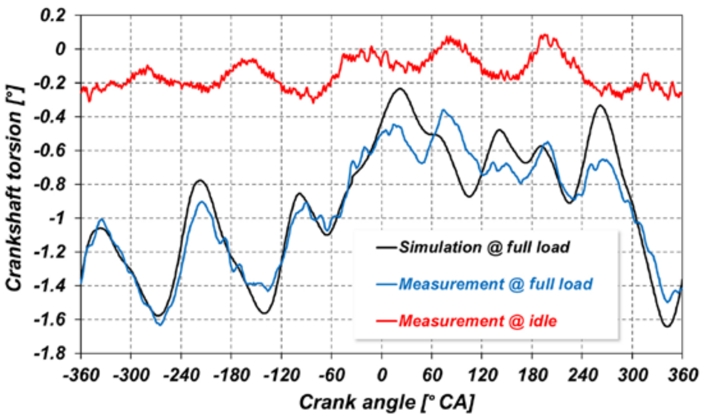

Figure 5: Measured and calculated crankshaft torsion on the flywheel side, with respect to the front end at 1,500 min^{-1}

4. Validation of crank mechanism simulation

A comparison of measured and calculated torsion across the entire length of the crankshaft was already shown in Figure 5. In addition, the measured results from the optical TDC sensor depicted in Figure 6 were applied to validate the calculated crankshaft torsion. Figure 7 shows this comparison for the full load point. Here, too, simulation was capable of reproducing the measured data with a precision of better than ±0.2° CA on all 24 cylinders.

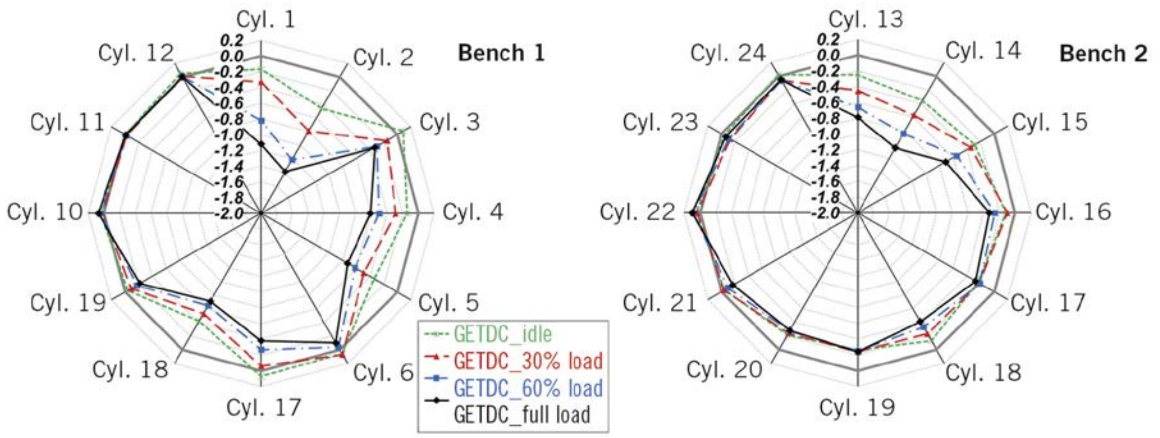

Figure 6: Relative angle position of the GETDC for different loads with respect to the front end at 1,500 min^{-1}, optical TDC measurement system

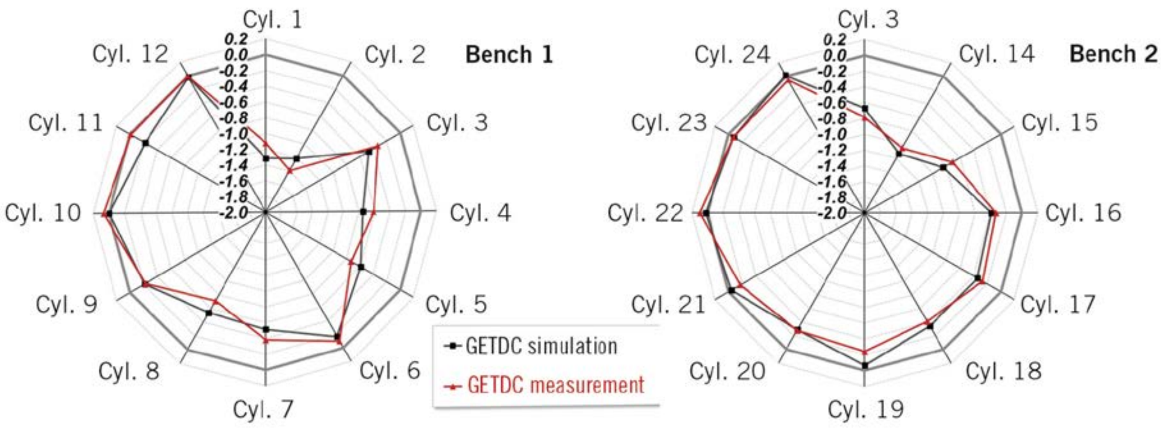

Figure 7: Comparison of determined angle of torsion in GETDC with respect to the front end at 1,500 min^{-1} FULL LOAD; simulation vs. optical TDC measurement system

5. Determination of key indication values

To allow depiction of the effects of including crankshaft torsion in cylinder pressure diagnostics, the LEC-CORA (zero-dimensional engine process simulation and analysis tool) software developed at the Large Engines Competence Center (LEC) of the Graz University of Technology had first to be modified and/or extended. Potential was created to overlay the crank angle-dependent torsion curves for individual cylinders as $\Delta\varphi_{(\varphi)}$ of the crank angle basis. This resulted in a changed volume function which formed the basis for the engine process analysis described below.

Three different methods were investigated in the context of the subsequent comparison of results:

Method 1: baseline
Engine process analysis with the measured raw combustion analysis data, on the basis of the cylinder-specific TDC determination over the thermodynamic loss angle performed on the test bench. As the full engine could not be run in non-fired operation, the non-fired cylinder pressure required for the procedure was determined in a coast-down test. The range of engine speed taken into account extended from nominal rated speed to 80% of nominal rated speed.

Method 2: fully corrected volume function
The verified torsion curves from the crankshaft simulation were used for this method. The torsion at the individual cylinders, described above as $\Delta\varphi_{(\varphi)}$, was taken into account in the engine process analysis.

Method 3: cylinder-specific TDC correction
Due to the high degree of effort needed for method 2, a version was investigated that represents the determination by measurement technology of cylinder-specific ITDC. For this purpose, the corresponding values were obtained from the simulated torsion curves of the individual cylinders at ITDC and used as ITDC offset values in the engine process analysis.

The volume functions resulting from the three different procedures can be seen in Figure 8. The "fully corrected volume function" and "cylinder-specific TDC correction" variants are shown ten times their actual size.

The influences on IMEP and energy balance are depicted below in Figure 8 for the evaluation of the quality of cylinder pressure diagnostics on the basis of the methods investigated.

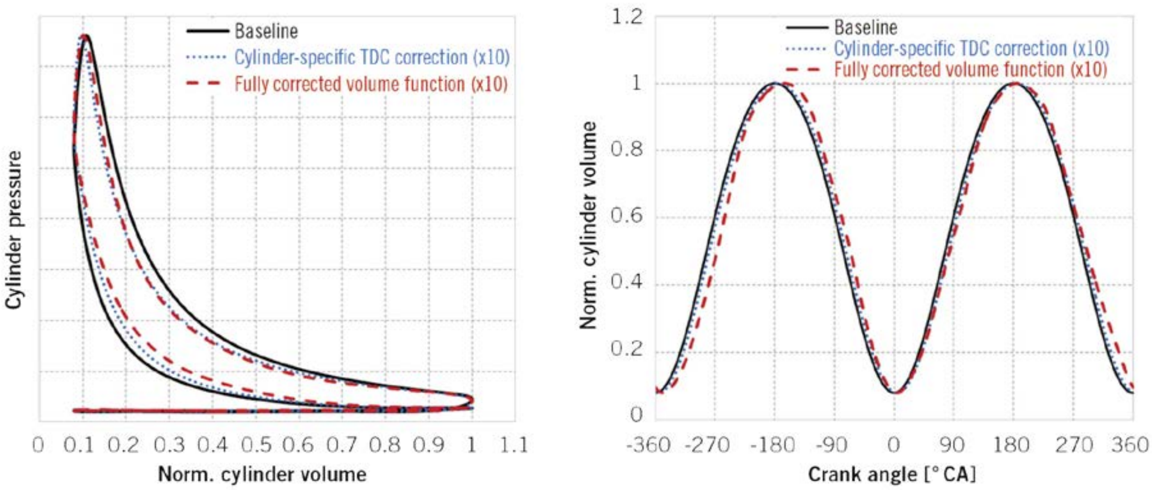

Figure 8: Resulting volume functions: "baseline", "fully corrected volume function" and "cylinder-specific TDC correction"

6. Results

In order to judge the quality of cylinder pressure diagnostics based on the examined variants, their influences on IMEP and energy balance are shown as follows.

6.1 Influence of crankshaft torsion on IMEP

An initial interpretation of the comparison shown in Figure 9 is that the fluctuation range of IMEPs determined by the use of the raw measurement data (baseline) is comparatively large. One factor that certainly impacts this fluctuation range is the specification of the charge mass for the individual cylinders. As only the entire mass can be measured on the test bench, the masses were distributed in equal parts over the individual cylinders. This assumption, which does not reflect reality, was followed for all the methods analyzed.

If the volume function is corrected in accordance with Method 2 (fully corrected volume function), it will result in a distinctly more homogeneous distribution of the cylinder-specific IMEP values. This also applies to Method 3, when the calculated torsion error at ITDC (cylinder-specific TDC correction) is used. This suggests that good results can be achieved with this method, at least with this test object. Depicted on the far right of the diagram is the cylinder mean for the three methods. Here, deviations of a non-negligible magnitude of 1 to 2% arise between the baseline method and the virtually identical results from the two other methods. Translated into FMEP, this deviation leads to an error in the range of 25 to 30%.

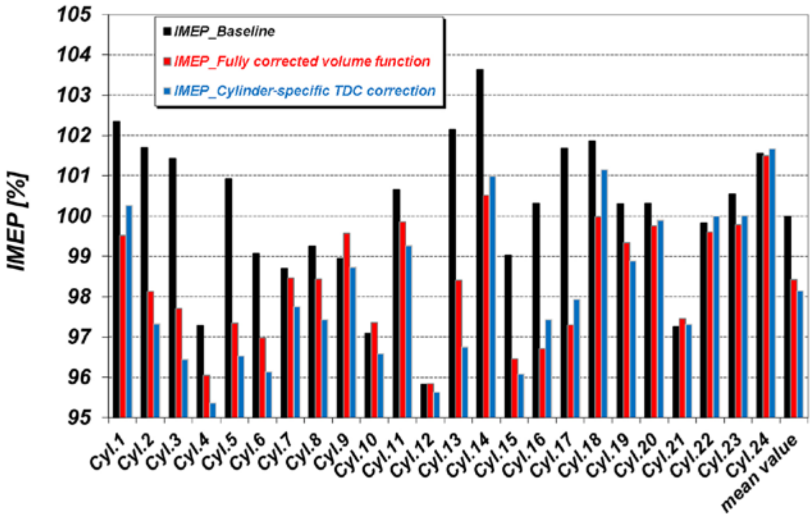

Figure 9: Calculated IMEP, taking account of differently corrected volume functions at 1,500 min^{-1} full load

6.2 Influence of crankshaft torsion on energy balance

As shown in Figure 10, it can be seen from the comparison of the energy balances determined that, even in this evaluation, the corrected methods exhibit a clear improvement in their average values (mean value). The result of a virtually closed energy balance is very promising. A study of the individual results from the “fully corrected volume function" and "cylinder-specific TDC correction" methods reveals a different, cylinder bank-specific behavior. The energy balances in Bench 1 for the "cylinder-specific TDC correction" method result in a consistently lower percentage rate than with the "fully corrected volume function" method, although the opposite picture is to be seen on Bench 2. Applied to the entire engine, the smallest range of fluctuations in the results occurs with the "fully corrected volume function" method.

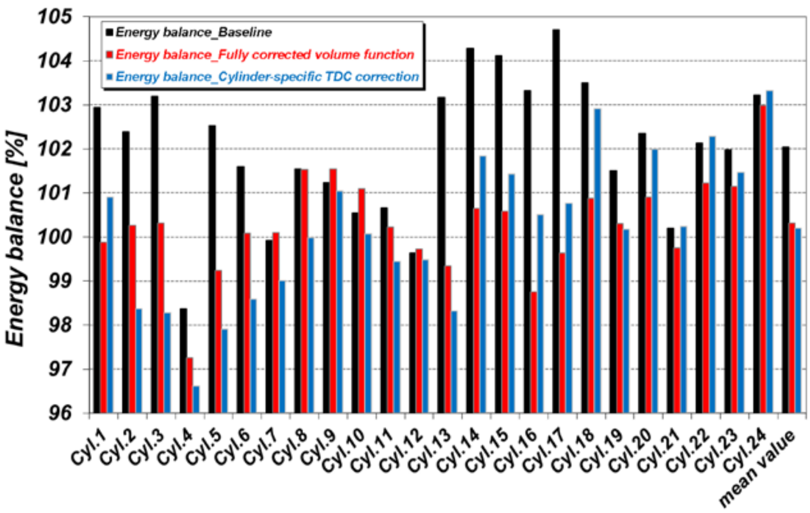

Figure 10: Energy balance for individual cylinders, taking account of differently corrected volume functions at 1,500 min⁻¹, full load

7. Summary

In this article, it was possible to demonstrate that crankshaft torsion is relevant for cylinder pressure diagnostics in large engines and that it can be incorporated into their analysis. Of prime importance is the dependency of crankshaft torsion on given operating points (load and speed).

The basis for the inclusion of crankshaft torsion in cylinder pressure diagnostics is the crank angle-dependent torsion of the individual crankpins, as determined in the simulation of the crank mechanism. The torsion angles can be validated by measuring the total torsion of the crankshaft between the flywheel and its front end. Values for the cylinder-specific torsion angle can be acquired using measuring technology, including the possibility of optical measurement of the piston position in gas exchange during fired engine operation.

If validated results for cylinder-specific crankshaft torsion are available, then a corrected volume function for cylinder pressure diagnostics can be calculated. In this case, the volume function (depending on crank angle) for each individual cylinder is calculated in the maximum expression and used in the subsequent engine process calculation.

If individual cylinder torsion can be determined at ITDC, then a simplified correction method can be applied. This means that the angle of torsion can be defined directly in the combustion analysis system, and corrected even as the measurement is being performed.

A third possibility is provided by by thermodynamic TDC adaptation in the engine process calculation. This requires excellent coordination of gas exchange simulation to determine the required charge mass at the individual cylinders.

measurement of the piston position in gas exchange of a magnitude of around 30% in FMEP and around 2% in energy balance are determined.

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