

Experimental and Simulative Friction Analysis of a Fired Passenger Car Diesel Engine with Focus on the Cranktrain

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ABSTRACT

The CO₂ reduction required by legislation represents a major challenge to the OEMs now and in the future. The use of fuel consumption saving potentials of friction-causing engine components can make a significant contribution. Boundary potential aspects of a combustion engine offer a good opportunity for estimating fuel consumption potentials. As a result, the focus of development is placed on components with great saving potentials. Friction investigations using the motored method are still state of the art. The disadvantages using this kind of friction measurement method are incorrect engine operating conditions like cylinder pressure, piston and liner temperatures, piston secondary movement and warm deformations which can lead to incorrect measurement results compared to a fired engine. In the past, two friction measurement methods came up, the so called floating liner method and a motored friction measurement with external charging. Both methods are getting closer to realistic engine load and thermal distortion, nevertheless there are still shortcomings. The floating liner method can only be used with great effort on a single cylinder research engine with cylinder pressure limitation. Using the motored method with external charging, cylinder pressure maxima can be reached, but with incorrect piston and liner temperature and an incorrect phasing of the cylinder pressure curve. In this paper a newly developed friction measurement method which allows a detailed strip down of a fired diesel engine for a complete engine map is used for the measurements. With this method the total friction can be divided into shares of the coolant pump, belt drive, oil pump, balance shafts, cranktrain, high-pressure pump and valve train. The measurement method was tested and verified with a state-of-the-art 3-cylinder diesel engine for passenger cars. The experimental results are presented in this paper. Furthermore the influence of different cylinder liner honings on the cranktrain friction was analyzed and compared in legislation cycle relevant engine operating points.

The second part of the paper deals with friction simulation. The simulation model used is an EHD (elastohydrodynamic) model to simulate the friction in the contact area between piston skirt and cylinder liner using the AVL software Excite Power Unit. The simulation model was parameterized with detailed input data such as surface roughness, piston and cylinder liner geometry, cold and warm deformations of piston skirt and cylinder liner, cylinder pressure and oil data. Using the EHD simulation model, basic relations of various parameters affecting the piston skirt friction can be investigated. Finally a friction optimized combination of piston skirt and cylinder liner geometry was investigated and analyzed in detail. The combination of EHD simulation and experimental results using the new friction measuring method allows a detailed estimation of CO₂ saving potentials resulting from friction reduction measures.

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INTRODUCTION

A modern inline 3-cylinder diesel engine with 1.5 liter cylinder capacity and exhaust gas turbocharging served as the basis for the results and interpretations presented in this publication. The data required for an evaluation of fuel consumption in a dynamic consumption cycle on an engine test bench such as engine speed and the defined torque input for a car chassis dynamometer measurement were used in this work. A subcompact car as currently available on the market was used for these measurements. In Europe the NEDC (New European Driving Cycle) is specified for determining fuel consumption and exhaust emissions. In the following presentation (Figure 1) the NEDC engine operating area is shown in the engine performance map. As can be seen here the NEDC only covers a small portion of the load and engine speed

region with the engine-vehicle-combination as tested. In order to achieve comparability with US legislation the FTP-75 as is usual in the USA is brought into the following diagram.

There are only very slight differences in the medium load and engine speed. The ISO lines shown, represent the portion of fuel consumption, required to overcome the engine friction depending on the operation range. Contingent to the mechanical efficiency a significant increase in the fuel consumption boundary potential through to a low load is recognizable. In order to obtain NEDC relevant statements for stationary friction investigations, a medium NEDC operational point was established, with which the engine examined complies with a rotational speed of 1675 min⁻¹ and a brake mean effective pressure of $BMEP = 2.6$ bar with a boundary

potential of approx. 20 %. For FTP-75 the mean operating point for the engine-vehicle-combination was investigated at $n = 1818 \text{ min}^{-1}$ and $\text{BMEP} = 2.8 \text{ bar}$. In order to obtain a statement relevant for both cycles, a representative load point of $n = 1750 \text{ min}^{-1}$ and $\text{BMEP} = 3 \text{ bar}$ was defined for the investigation.

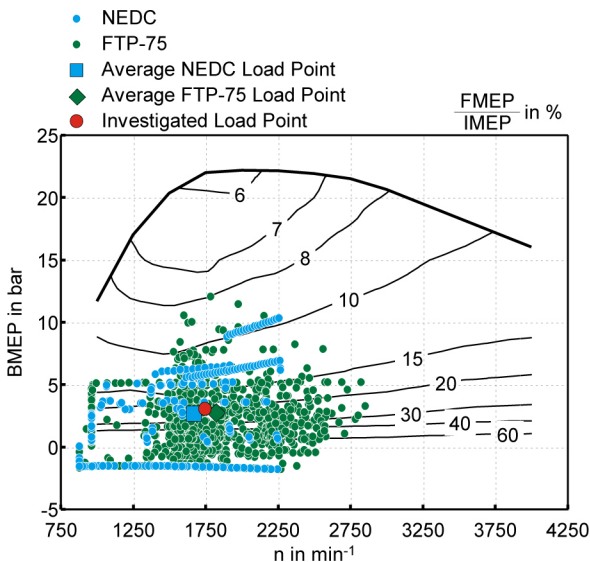


Figure 1. Fuel consumption boundary potential in %.

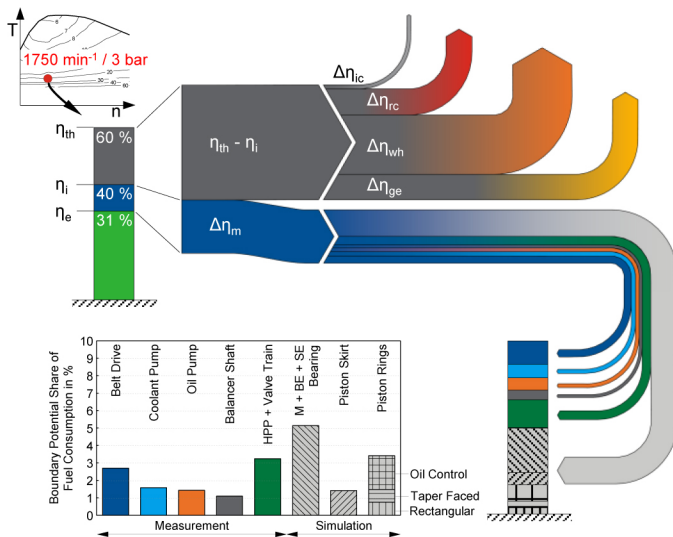


Figure 2. Boundary potential share based on thermodynamic loss analyses.

A loss analysis was made in a first step according to [1] by means of representing the losses of the diesel engine examined in a high level of detail. As shown in Figure 2, an allocation of the mechanical losses for the NEDC and the FTP-75 relevant load points ($1750 \text{ min}^{-1} / 3 \text{ bar}$) was made in addition to separating out the thermodynamic losses. The high resolution level for the separate mechanical losses in coolant pump, belt drive, oil pump, balance shaft, diesel high-pressure pump (HPP) and cranktrain can be recognized. The cranktrain again can be sub-divided into the components crankshaft main bearings (M), connecting-rod bearings (BE, SE), piston pin, piston skirt and piston rings. This precise itemization of the engine

friction is achieved through a combination of fired measurement in the entire engine map area making use of the newly developed measurement method and an EHD simulation.

The sum of the losses shown in Figure 2 must thus result in the previously presented fuel consumption boundary potential of approx. 20 %. The dominant share for the cranktrain can be recognized here, which accounts for approx. 50 % of the total friction in the operating point presented. With the help of this detailed presentation of the friction-causing components potential friction reducing measures can be assessed, or even predicted. In the following the measurement method required for this presentation and the EHD simulation used are described in detail and analyzed.

FRICITION MEASURING METHODS

A precise analysis of the mechanical losses of the engine as also the assessment of possible improvement potentials based on this analysis require an exact metrological registration of the friction loss. The most frequently employed friction measurement methods currently in use are presented in the following [2, 4]. The indicating and strip-down method in particular as used in the current experimental investigations will each be explained in detail. A number of other procedures are available in addition to these measuring methods, but these have lost their significance as a consequence of their limited results quality.

Indicating Method

Using the indicating method the friction losses in fired operation can be established under real load and temperature conditions. Load, engine speed and temperature limitations are not given in this method [3]. Figure 3 shows the structure of the measurement chain (sensors, signal conditioning, data acquisition and signal evaluation) in the course of the friction measurement investigations using the indicating method (measurement specifications see appendix).

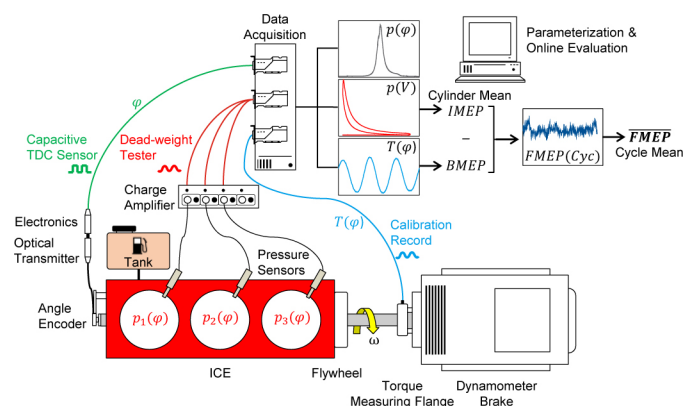


Figure 3. Engine test bench with cylinder pressure indication system.

The cylinder pressures were captured with high temporal resolution for the pressure indication results using piezoelectric pressure sensors screwed into the glow plug bore holes and downstream charge amplification. The allocation of the pressure signals for the crankshaft position was carried out using an angle encoder attached to the free end of the crankshaft. Here the precise determination of the top dead center had a decisive role in the preliminary work for the

measurements. The integration of the measured cylinder pressure curves throughout a work cycle gives the internal work done, which applied to the piston displacement gives the (IMEP) indicated mean effective pressure [1].

$$IMEP = \frac{\oint p dV}{V_d} \quad (1)$$

The effective torque is measured crank angle resolved by a torque measuring flange on the drive shaft between the flywheel and the dynamometer brake. The brake mean effective pressure (BMEP) for a four stroke engine is given according to the following equation:

$$BMEP = \frac{4\pi T}{V_d} \quad (2)$$

The friction mean effective pressure (FMEP) is calculated from the difference between the indicated and the brake mean effective pressure.

$$FMEP = IMEP - BMEP \quad (3)$$

Due to the subtraction of two similarly dimensioned and tolerance affected measured values the indicating method demands a very high measurement and reproduction precision from both, IMEP and BMEP [1]. Measurement uncertainties occur primarily as a result of potential measurement errors in the pressure indication results as also in the torque measurement, errors in TDC determination, errors in the thermodynamic evaluation, unequal distributions in case of multi-cylinder engines and cyclical pressure and torque fluctuations. Over and above this the course of the measuring process itself has a decisive influence on the measurement quality and the reproducibility of the results and for this reason it is described in more detail in the section "Measuring Process". The high measuring technology standard required for the indicating method consists in minimizing the faults described and also in the influencing variables. In order to maintain the high precision requirements for establishing friction a procedure has been established in the course of this work (application - calibration - TDC determination - processing - evaluation) for the measurement of fired friction maps. Mean effective pressure fluctuations (cylinder charge, injection quantity) between the separate cylinders were taken into account through high-pressure indicating of all cylinders and subsequent averaging the friction mean effective pressure. The calibration of the entire pressure measurement chain (pressure sensor, charge amplifier, measuring cable) was done using a pressure compensator (dead-weight tester). The torque measuring chain is tested on the one hand by a comparison of the zero torque in the unloaded shaft status and on the other by connecting an electrical calibration signal produced on the measuring flange ($\pm 50\%$ nominal torque). Since even small angular errors can result in the pressure and volume progression to major errors in the friction mean effective pressure [7], the determining of the top dead center position is made with the greatest precision and with complex method using a capacitive TDC sensor [4]. The sensor is assembled on the cylinder head instead of the injector (cylinder 1) closest to the angle encoder. For the TDC allocation the engine is motored in an operationally hot

state at a mean engine speed (2000 min^{-1}). Any heat expansion influences of the cranktrain should thus be taken into account in this process. What is referred to as the thermodynamic loss angle is used for plausibility checking and additional checking of the trigger point before and after the measurement. The thermodynamic loss angle gives the difference between maximum pressure and top dead center caused by wall heat and leakage losses in motored operation [1]. In addition to the calibration of the measuring system as also the angle allocation for the signals the simultaneous further processing of the two mean pressure values (indicated, effective) is of significance for a correct subtraction due to the cyclic pressure and torque fluctuations. The resulting cyclic friction pressure is measured over a period of 25 sec to rule out errors and is then saved as the result for the operating point under observation.

Strip-Down Method

In the strip-down method the separate friction-causing components and auxiliary units (oil and coolant pump) of the engine for friction evaluation are stepwise disassembled and when necessary replaced by external conditioning units, see Table 1. The measurement cycle starting with a complete engine, is carried out for each strip-step and as far as possible under the same boundary conditions in regard to coolant and oil temperature. The mechanical loss for the disassembled engine component examined is calculated by the differentiation of measured FMEP for the previous (k-1) and the current strip-step (k).

$$FMEP_{\text{Component}} = FMEP_{k-1} - FMEP_k \quad (4)$$

The main focus is on producing comparable operating conditions to avoid any falsification of the measurement results as a result of different operating temperatures. This is realized by external conditioning units for the oil and the coolant circuit, as explained in more detail in the section "Measuring Process". Furthermore these conditioning systems provide the opportunity for carrying out friction evaluations at various media temperatures and pressure levels. The friction measurement can in principle be made in either motored or fired engine operation using the indicating method. In the case of a motored measurement the engine is driven by a dynamometer and the drag torque (effective torque) is measured at a specific engine speed. Without a correction by means of an additional high-pressure indication, the gas exchange losses will be allocated in principle to the mechanical losses [5]. The most significant limitation of the motored method compared to real operation is that firing does not take place and lower temperatures and pressure loads on the components result from this. In Figure 4 the measured cylinder pressure curves as also the estimated mean piston temperatures for a motored, low load and high load operation point are compared in each case at an engine speed of 2000 min^{-1} . It is clearly recognizable here that even in the low load area there is a virtual doubling of the peak pressure and the piston temperature in comparison to the motored operation. In addition there is a change in the position of the maximum cylinder pressure ($\Delta^\circ\text{CA}$) and as a consequence the position of maximum piston normal force in fired operation.

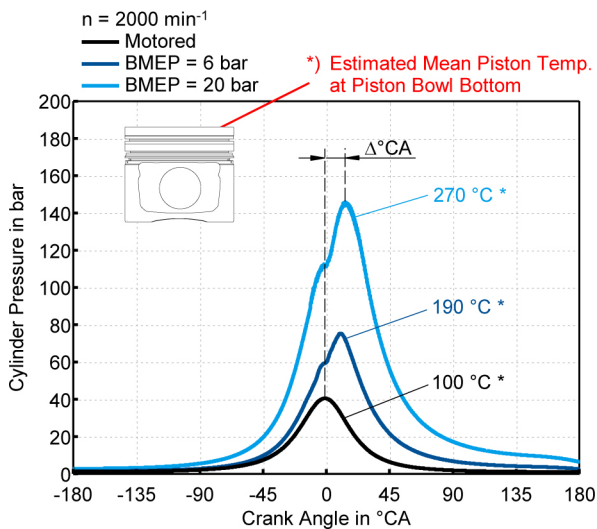


Figure 4. Cylinder pressure curve of different load points.

In a classic motored measurement the load-dependent friction influences (bearing loads, cylinder distortions) are thus not correctly presented, with the result that evaluation errors can occur for the valve or cranktrain friction. This problem is clearly shown in Figure 5 through a comparison of the friction from a fired measurement to the indicating method as also various motored measuring methods (load change activated, cylindrical camshaft, open cylinder head) for two “friction optimized” engines in relation to a basis variant. In the course of the investigations made in [6], it could be shown that in dependence on the used motored method there were significant deviations compared to the fired indication method.

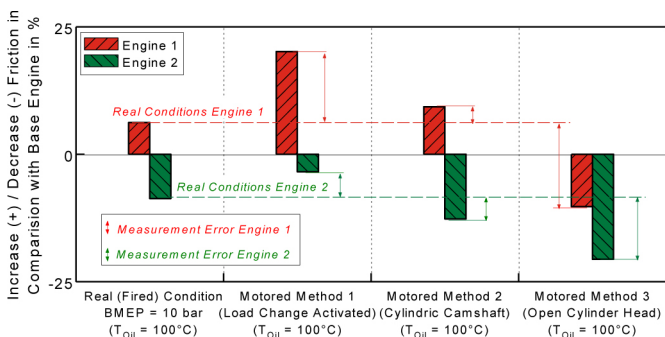


Figure 5. Comparison of friction mean effective pressures (different methods) [6].

Furthermore the high-pressure pump losses in motored operation could only be inadequately established as a result of the lack of an injection process (comparatively low rail-pressure). For auxiliary units such as the oil and coolant pump, which show largely speed-dependent losses, the motored method offers an adequate level of precision.

Motored Method with External Charging

The motored method with external charging is a special variant of classic motored measurement. Through the use of external charging it is possible to create peak pressures in the combustion chamber, such as those prevalent in fired operation. The method of external

pressurization of the piston for friction measurement in motored operation is already familiar from earlier publication [8]. The advantages from the representation of the real peak pressure, however, still stand in the shadow of the very significant disadvantages resulting from the lack of a temperature load from firing and the incorrect cylinder pressure gradient in comparison to fired operation. Detailed analyses of motored friction measuring with external charging were made in [15].

Floating Liner Method

The floating liner method makes possible the direct measurement of the friction forces on the cylinder liner resulting from the piston group in fired operation. First experiments with this method go back to [9]. The force is established using force measurement sensors, on which there is a floating suspension supported cylinder liner. By means of this principle the friction force curve in the stroke direction can be recorded in high resolution. Due to the very high level of constructional and measuring technical effort required, the floating liner method is used exclusively for single-cylinder engines. As a result of the design the cylinder distortions that occur deviate in comparison to a real complete engine [3]. Further to these issues the maximum possible peak pressures are limited by the force neutral seal that is needed between the liner and the cylinder head [10].

Each of the measuring methods presented has its areas of application and the measurement quality and informational validity have been further prompted by and parallel to rising demand.

The following chapters present the results of measurements with a newly developed measuring method. This measurement method allows a very high level of detail to be achieved in the identifying and separating out the various friction causing engine components and it combines the strip-down method with a fired indicating friction measurement.

MEASUREMENT METHOD OF CRANKTRAIN FRICTION ON FIRED ENGINE

Using this newly developed friction measuring method it has been made possible to carry out a strip-down measurement in fired engine operation for the entire engine performance map. The great advantage compared with the previously described friction measurement processes is on the one hand in the real operational conditions given by the fired operation and on the other hand by the possibility provided for carrying out the investigations on multi-cylinder engines. The main focus here lies on the separated friction recording on the cranktrain. The fired measurement of the cranktrain friction represents the greatest challenge and this can be measured separately from the complete engine through the use of a specially developed friction measurement method. This method makes possible a separation of the crankshaft torque from the valve train torque and can be integrated directly to the engine on the test bench. As a consequence of the measurement setup the friction of the valve train and high-pressure pump is given as a sum in the results presented for the strip-down investigations. In addition a separation of the valve train and the high-pressure pump is planned for the future.

MEASURING PROCESS

Required Workflow for High Measurement Quality and Reproducibility

A systematic procedure in the measuring process is absolutely essential in addition to the preliminary work described in chapter "Measuring Methods" (calibration of pressure and torque measurement chain, TDC assignment) in order to ensure a high measurement quality for the measurement method used. This defined measuring process was used for all the measurements made and its value could be confirmed by the very good reproducibility of the measurement results.

At the start of friction investigations on a test engine the first task is to compile the complete engine characteristic map. The engine has to be externally power supplied instead of the engine generator which allows to eliminate the charge controller influences. Using the friction map that have been compiled it is possible to make statements about the load sensitivity of the complete engine. In a subsequent work the friction-influencing data is established, this is required for the subsequent fired strip-down measurements. In particular the recorded data of the complete engine map, such as oil and coolant temperature, oil pressure, EGR rates and engine-specific combustion data. The measurement data serve on the one hand as input parameters for the required oil and coolant conditioning unit for the setting of precisely the same temperature and pressure conditions as with the entire engine and on the other hand for checking the engine-specific application data. As a result of the EGR rate on the position of half mass fraction burned (MFB50) and thus indirectly on the position of the maximum piston normal force, a cross-influence on the friction at the piston group is also given here.

In the course of the investigations that were carried out, a so called hysteresis effect could be demonstrated in the measurement of the load sweeps. Figure 6 shows how a test series on the load variation for a low load point (BMEP = 0.5 bar) through to full load (BMEP ~ 22 bar) and back again was measured.

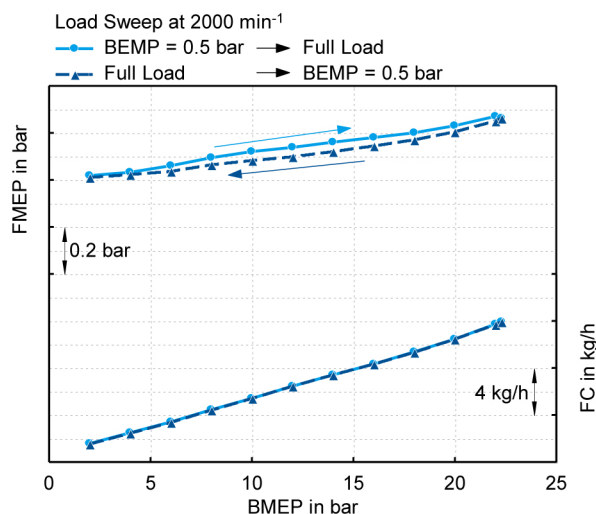


Figure 6. Hysteresis effect.

In the course of this the engine was already running warm until reaching the starting point operational temperature and subsequently a dwell time of 10 min for each measuring point was maintained, in order to be able to rule out the influences of oil or component temperatures. It can be recognized here that in dependence on the direction of loading (load variation direction) the hysteresis has a significant influence on the measured friction values. On a comparison of the measured fuel consumption as also other measured values that gave virtually the same results independent of the direction of measurement, measurement errors can be concluded in regard to the indication. The occurrence of a load change and temperature drift of the uncooled pressure quartz can be pointed to as possible causes of the hysteresis effect [4].

A further test for establishing the optimal measuring process is presented in Figure 7. In order to analyze the influence of component temperatures on friction in sudden load changes from one measuring point to the next, temperature measuring points were implemented on the cranktrain (cylinder liner, crankshaft main bearings). The optimal settling time of the operating points for compiling the engine friction characteristic maps could be established as a result.

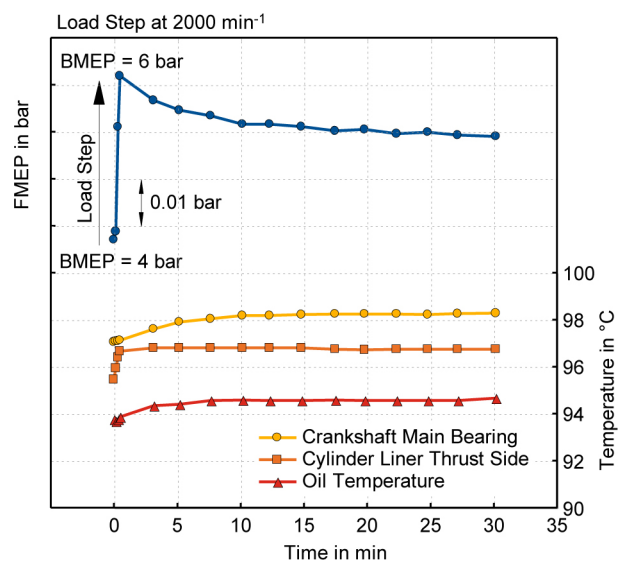


Figure 7. Load step at 2000 min⁻¹.

A load step from 4 to 6 bar brake mean effective pressure at an engine speed of 2000 min⁻¹ is shown as an example. An abrupt rise in friction at the precise time of the load step can be seen here. With a continuation of the measuring time the friction declines again down to a constant value at approx. 10 - 15 min. The measured temperature of the cylinder liner reaches a constant value after approx. 3 min. By contrast with this the oil temperature and the temperature in the crankshaft main bearing have a settling time of virtually 10 min.

On the basis of the knowledge acquired, it was possible to develop an optimized measuring program for the subsequent characteristic maps and load variation investigations. The optimized characteristic map measuring process is shown in Figure 8. An engine map measurement, independent of the strip-down step, was started with a load variation at the minimal speed and load to be measured. On reaching full load the next speed was set at minimal engine load and