CONTINUOUS SLOW DYNAMIC SLOPE APPROACH FOR STATIONARY BASE INTERNAL COMBUSTION ENGINE MAPPING

by

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Original scientific paper
https://doi.org/10.2298/TSCI190308171M

Engine control optimization, with its always growing complexity, is in permanent focus of engine researchers and developers all over the world. Automotive engines are dominantly used in dynamic conditions, but generally, steady-state operating points are used for building up mathematical models which are later subject to the numerical optimization. For this purpose, a large amount of steady-state regimes needs to be evaluated through experimental work at the engine test stand, which is an extremely time and funds consuming process. Consequently, the methodology for data gathering during engine dynamic excitation could lead to significant savings at the expense of acceptable data accuracy loss. The slow dynamic slope method starting from a stationary operating point was evaluated by several authors in the past. In this paper, slow dynamic slope method with exclusively transient excitation will be presented drawing attention to some of its advantages and drawbacks. The rate of change of engine load as a main control parameter during dynamic test is of great importance for the quality of the final data and for total test duration. In this regard, several tests of different duration were applied for fixed engine speed values to cover engine speed-load usage domain. An approximation of stationary testing results obtained in this way could be used for evaluation of the map gradients and thus as a guideline for additional stationary tests based on design of experiment method.

Key words: *internal combustion engine, dynamic engine testing, engine mapping, slow dynamic slope*

Introduction

Generally speaking, the modern society has a growing need of powertrain systems for transportation purposes. Ultimately, the reduction of carbon footprint and thus reduction of fuel consumption along with the reduction of toxic emission is of primary concern. On the other side, the increase of reliability and drivability are also of great importance. Those requirements never go hand in hand when it comes to powertrain development. As a consequence, these demands have led to a high increase in powertrain complexity and modern engines are equipped with many complex systems, which in turn requires a more sophisticated control and in-depth analysis of engine overall performance.

The conventional procedure for optimizing basic engine control parameters consists of several steps. At first, the legislator defines the driving cycle during which measurements of exhaust gas composition and fuel consumption will be made new European driving cycle,

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worldwide harmonized light vehicles test procedure, and real driving emission. A basic modeling of vehicle dynamics [1, 2] will provide an approximate engine speed and load demand time-series, which will be used to determine the share of the most representative engine operation points. Introducing various ECU's parameters, it became impossible to conduct full-factorial stationary experimentation on a test bench. Design of experiment (DoE) methods are useful tools for reducing needed stationary data sets for building up a mathematical model. Those methods can incorporate pre-knowledge in terms of boundary conditions within N-dimensional space of input control convex hull on local and global basis.

Gathering steady-state measurement data for mathematical modeling and model verification is the next step, which requires an engine test stand with appropriate measuring systems. During stationary experimentation, time dedicated to single operation point is mainly influenced by the period of stabilization prior to data measurement. In some cases, the engagement of several test stands operating in parallel is the only option for performing data collection in a reasonable amount of time, even if DoE features have been applied. This approach will generally provide the most accurate results, but the costs and complexity of further data analysis will be heavily increased. Mathematical model evaluation, numerical optimization, verification of stationary model behavior and extraction of ECU's control maps are all intermediate steps before the final validation of the powertrain system in dynamic conditions.

An important fact regarding automotive powertrain systems is that they are mainly used under dynamic conditions and further development of emission test cycles will surely be going towards even more emphasized dynamic tests. Powertrain systems are generally very complex, and learning more about their dynamic characteristics is of great importance for optimizing the dynamic operation. Consequently, dynamic tests are the logical answer for the identification of dynamic characteristics, but is there a dynamic test that could reduce the time needed for stationary based experiments? A potential answer could be found in the methodology named slow dynamic slope (SDS), which was the subject of the Murakami *et al.* [3] and Leithgob *et al.* [4] research.

The main topic of this paper is the introduction of fully dynamic SDS experiments, and the comparison of system excitation and responses obtained this way with classical approach which was presented by Keuth *et al.* [5] and further analyzed by authors in [6]. Also, some guidelines on potential problems which may occur during SDS engine testing and during data analysis will be given. This paper relies significantly on the authors' pervious research [6], in which additional explanations and theoretical principles can be found.

Theoretical assumptions

In theory, system is linear if the superposition law can be applied and if its stationary response is linear function of system input and system initial condition. Gain of such a system is relation between system stationary input and output. Taking into account system dynamics, time constant is the parameter characterizing the response to a step input of a first-order, linear time-invariant system.

As an example, the first order linear system (LS1) will be analyzed throughout ramp excitation *i.e.* with input signal characterized by constant gradient. If we assume that the system could be defined by its time constant T_1 and gain K, the system equation will be:

$$T_1\dot{y}(t) + y(t) = Ku(t) \tag{1}$$

Applying ramp excitation with a constant gradient β defined as $u(t) = \beta t$, to the LS1, system response equation will have a form given by:

$$y(t) = K\beta t - K\beta T_1\left(1 - \mathrm{e}^{-\frac{t}{T_1}}\right) \tag{2}$$

For a first order linear system at particular time, the difference between system response value and system gain multiplied by system input value will become constant, as shown in following equation:

$$\Delta y(\infty) = \lim_{t\to\infty}\Delta y(t) = \lim_{t\to\infty} - K\beta T_1\left(1 - \mathrm{e}^{-\frac{t}{T_1}}\right) = -K\beta T_1 \tag{3}$$

This fact could be used in the context of system ramp excitation if the system possesses either a relatively small time constant, or has an excitation ramp with a relatively small gradient, so that system output falls within the process disturbances. Another approach to eliminate the response offset, shown in eq. (3), is by implementing an additional system examination ramp using a symmetric ramp with negative gradient.

Unfortunately, the processes within internal combustion (IC) engines cannot be classified as linear and of first order, but for simplicity and further comparison of classical SDS and SDS without stationary operation, an arbitrary LS1 will be analyzed.

In the research [6], a classical SDS was configured as follows:

- At demanded constant engine speed, engine load was set to value in the middle of operational load span and settled until stationary operation.
- For a defined ramp gradient, the engine load was increased to the maximum load, maintained at maximum level for a few seconds and decreased to full motoring with the same, but negative, gradient.
- After reaching the minimum, the engine load was increased again until reaching the starting load value (mean value between full load and full motoring at a particular engine speed).

In fig. 1(a), labeled as SDS(1), such a system excitation and LS1 response are shown. Different SDS tests were set by varying the overall duration of the test (in other words, different ramp gradients) and engine speed, which were maintained constant during whole SDS cycle.

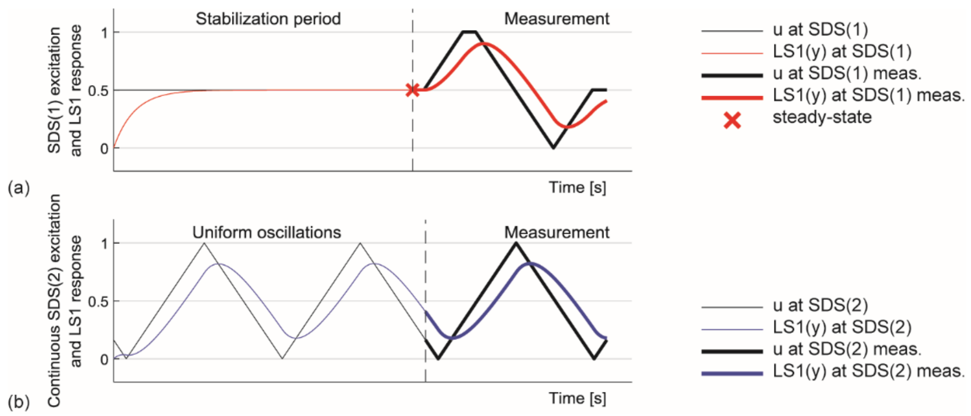

Figure 1. Comparison of system preparation and measurement periods for classical SDS and SDS without stationary operation along with arbitrary LS1 response

In fig. 1(b), continuous SDS system excitation and the LS1 response are shown. For easier comparison, this type of test is labeled as SDS(2). In this case, the system is brought into uniform oscillations. Instead of waiting for the system response to become stationary prior the start of measurement, here we have an option for online monitoring whether the system responses get into repeatable oscillations and if that condition is met, the measurement begins. Certainly, system response deviations from the previous oscillation period needs to be defined. It is not a bad practice to record dynamic measurement for a slightly longer period than the time of full oscillation, as shown in fig. 1(b). This data could be useful for later data validity check. Depending on which operating point the engine was running, generally two to four uniform SDS(2) input periods are enough for all observed parameters to get into oscillations with acceptable deviation.

In fig. 2, LS1 response, y, for two types of dynamic excitations, SDS(1) and SDS(2), as a function of excitation, u, are shown. Also, in the same figure the middle line (ML) of the system response envelopes is shown. In the case of ramp input with infinitely small gradient value, or in the case of LS1 with zero response offset, the area inside of envelope will become equal to zero and thus the system response would lie on the regression line. In that case, results will also coincide with line matching stationary excitation response of LS1 with gain equal to K = 1, as in this example.

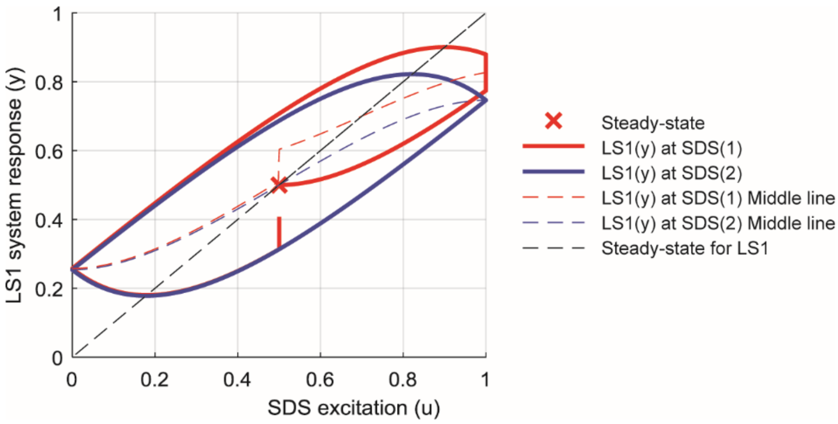

Figure 2. Arbitrary LS1 system response as a result of different excitations, SDS(1) and SDS(2) (*for color image see journal web site*)

The main idea behind gathering information about stationary system response based on the analysis of dynamic SDS data is by evaluating the ML of the system response envelope. The advantage of SDS(1) test is that an absolutely accurate value of system stationary response is present at the beginning of the test. On the other hand, this stationarity introduces discontinuity of the SDS(1) ML. The second potential issue lies within the asymmetric excitation regarding the upper and lower input limits. The benefit of the upper input holding is that physical quantities with great thermal inertia are provided enough time to overcome their significant time constants. Regarding the lower limit, an input delay is omitted because of practical reasons. If the test is configured in such a way that the sweeping of engine load goes to zero or negative torque values, there is great concern of getting into fuel cutoff regimes. In that case, thermal fluxes will be drastically violated because of combustion absence, and the engine's responses nonlinearity will become significant.

The difference between the regression line (stationary LS1 input/output for K = 1) and ML for different envelope shapes determined by LS1 time constant or SDS ramp slope is shown in fig. 3. As it is noticed, SDS(1) approach will always provide certain discontinuity at the mentioned difference line in system excitation domain, compared with SDS(2) excitation sequencing.

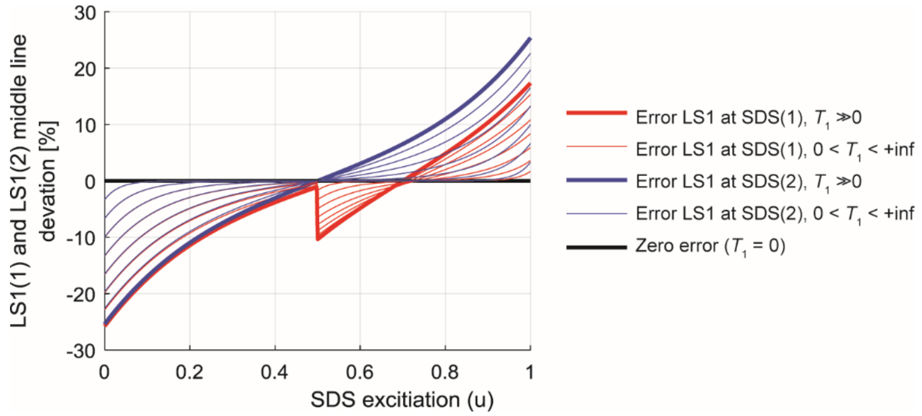

Figure 3. Difference of LS1 regression line and envelope ML for different lengths of SDS(1) and SDS(2) tests (*for color image see journal web site*)

Experimental installation

Experimentation was conducted on an automotive diesel engine PSA DV4TD 8HT coupled with a high performance dynamic AC dynamometer. Basic information of the engine and dynamometer is shown in tab. 1. During tests, the OEM engine control unit was used, so that there was no concern about violation of system boundaries [7] during setting up a demand values of engine operation points. The on-board diagnostics link was used for additional check of the engine proper functionality. All engine effective parameters were measured in

Table 1. Engine, dynamometer and test cell main features

Engine	PSA DV4TD 8HT	Dynamometer	Rotronics, ATB Schorch
Manufacturer	PSA group	Max. braking torque	700 Nm
Model	DV4TD 8HT	Max. braking power	300 kW at 10000 min^{-1}
Type	4 cylinder inline, 4 stroke CI, 2 valves per cylinder; turbocharged, non-intercooled	Torque sensor	HBM T40 2kNm, 0.05% acc.
		Intake and exhaust pressure sensors	IHTM, 0-5bar, ±0.1% FSO
Bore/Stroke	73.7 mm/84.0 mm	Cylinder pressure indication sensors	AVL GM12D, max 200 bar, 15 pC/bar, linearity ±0.3%
Rated power	40 kW at 4000 min^{-1}		
Rated torque	130 Nm	Temperature TC	LFTC-KA, type K, ±2.2% FS
Fuel injection system	Common rail, Siemens 8HT	Temperature RTD	RTDLF Pt100B, class B, ≤0.8 °C at 100 °C
Turbocharger	KP35 (3K-BW)		

time domain using NI PXI platform with appropriate in-house developed NI LabVIEW application. Thanks to the modularity of the acquisition system, multifunctional NI PXI 6229 and NI PXI 6123 cards were used for the main data acquisition.

Engine indication was performed using the AVL IndiMaster module, advanced AVL IndiCom and the AVL Concerto software for indication data evaluation. Cutting-edge AVL Micro IFEM Piezo charge amplifiers were used, alongside the AVL GM12D (200 bar range. ±0.3% FSO) pressure indicating sensors and a high-resolution incremental encoder AVL 365C (resolution up to 0.1 CA). System automation and test sequencing was performed using the intelligent AVL CAMEO software [8] connected *via* Modbus to the dynamometer control rack. All dynamic tests were configured in a way that after transitioning from idle operation, engine speed and engine load were controlled in closed loop using ramp sweeps of different durations. The signal for acquisition start was predefined within AVL CAMEO, which greatly simplified data processing and time synchronization of measured channels in time and angular domain. An additional part of the experimental installation was the fuel consumption measuring unit AVL 733, and fuel temperature control module AVL 753. The engine was equipped with additional temperature measuring points, especially for intake, exhaust and turbocharger unit. Basic installation components and connections are shown in fig. 4. The measurement results of several engine variables will be presented as an example. Quantities with different time constants are deliberately chosen and an elementary description of those channels is given in tab. 2.

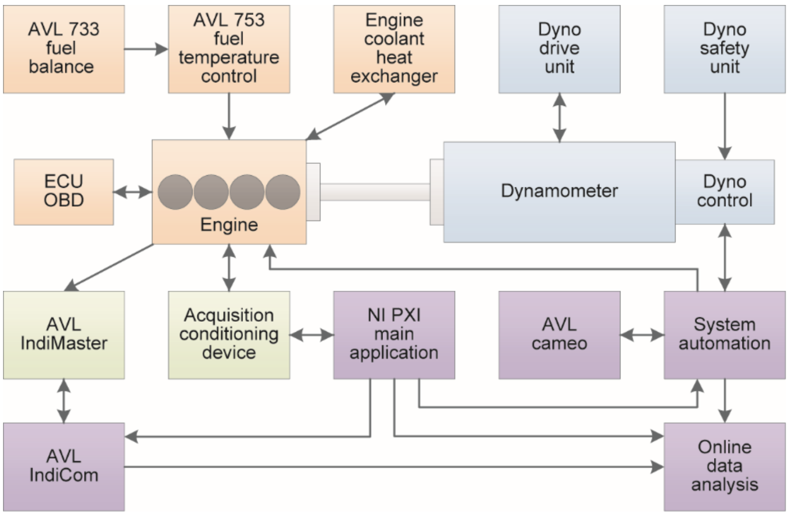

Figure 4. Engine test cell components and general dataflow of system automation at the ICED lab

Table 2. Description of measured channels used in the following diagrams

Channel	Description	Measurement type	Response
PT1	Turbine inlet exhaust gas pressure	Directly, piezoresistive sensor	Very fast
TT1	Turbine inlet exhaust gas temperature	Directly, TC	Slow
TT2	Turbine outlet exhaust gas temperature	Directly, TC	Slow
BSFC	Brake Specific Fuel Consumption	Indirectly calculated	Fast

Experiment plan

Before implementing any dynamic test, the engine was examined in detail at steady-state operating points in-between engine speed and load operational limits. Thanks to an option for engine motoring, torque set-point went from full motoring up to full load for eleven different engine speeds (from 950 min^{-1} to 3900). Approximately, 250 stationary points were examined for determining the engine base stationary characteristics. Data collected in this way will be used for comparison with results gathered by the implementation of SDS methodology.

The continuous SDS series of experiments were configured in the following way:

– During each test, engine speed were held constant. Engine speeds of 1500, 2000, 2500, and 3000 min^{-1} were evaluated.
– For each specific engine speed, SDS(2) sequences were configured by varying ramp durations. The input rising and falling ramps were set for total time of 120 seconds up to 600 seconds with steps of 60 seconds for a single engine speed. Additional 20 seconds were recorded to verify input/output envelope enclosure. In that manner, SDS(2) tests were named as 140, 260, 380, 500, and 620 seconds.

Overall, 20 dynamic runs were executed. Before each measurement, the engine load was varied at least two times by means of predefined SDS cycle ramp gradient in order to ensure repeatable oscillations of measured values.

Special attention was given to data post-processing in terms of event synchronization and filtering. During dynamic engine sweeps, this step is very important due to the inability to repeat or to prolong the measurement. All channels, one at a time, were processed with custom parameterized Savitzky-Golay filter because of its great ability to smooth the data with reduced possibility of destroying the data, especially information related to sudden changes of signal value. Another recommendation for data filtering in terms of noise reduction and preservation of data dynamics is by use of recurrent dynamic non-linear autoregressive neural network with exogenous inputs (NARX).

The time series of engine torque demand (D) and actual values (A), alongside TT1 and PT1 signals for SDS(1) sequence are shown in fig. 5. Temperature and pressure readings are chosen to emphasize difference between signals measured using different techniques. Also these readings are influenced by thermal inertia of the system and by different time constants of used sensors. Those two readings are presented as a function of system input in fig. 6,

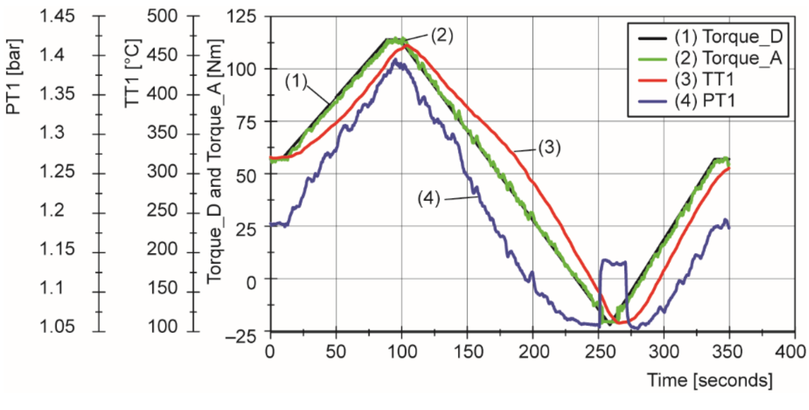

Figure 5. The SDS(1) engine torque demand at 1500 min^{-1} and 350 seconds run time; actual torque, TT1 and PT1 in the time domain

alongside appropriate ML and stationary measured values. The ML are linearly trimmed taking into account the slight inequality of absolute system excitation gradients during physical realization of the experiment.

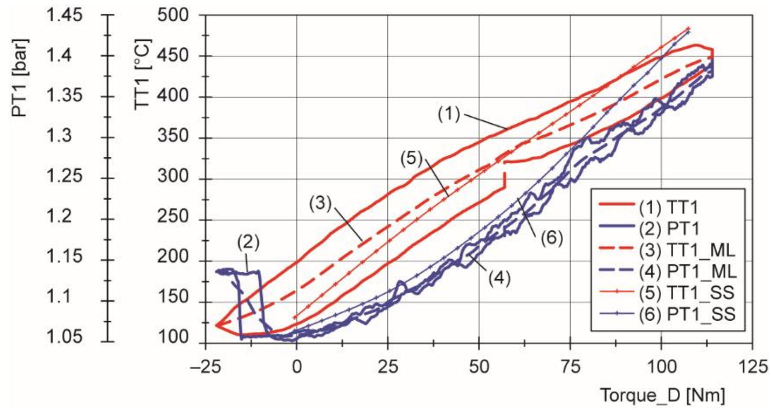

Figure 6. The TT1 and PT1 system responses; ML and steady-state also included as a function of SDS(1) system excitation for total of 350 seconds

Temperature envelope and ML deviations in the zones apart from the starting stationary point are noticeable, but the general trend is as expected. Comparing with TT1, the pressure traces have a remarkably smaller envelope area due to the faster reaction of the measuring device. Also, in the lower region of engine load, the ML and SS line have a relatively good matching except in the region of negative engine load in which fuel cutoff occurred, which is also noticeable as an exhaust pressure increase in fig. 5 due to EGR valve closing. For a similar input gradient value as in previous figures, SDS(2) test results for the same parameters and engine speed are shown in figs. 7 and 8.

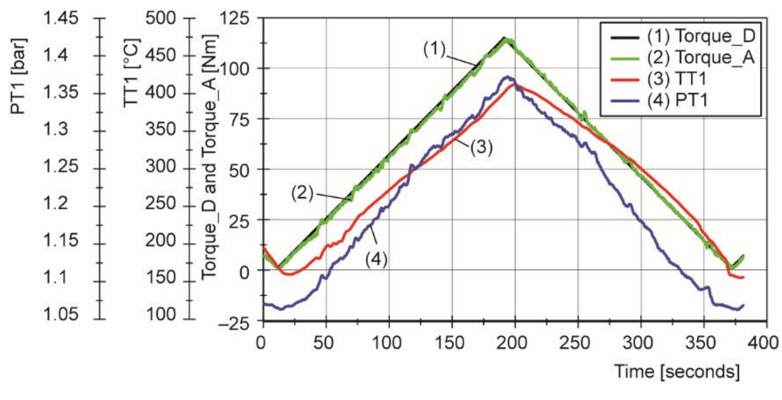

Figure 7. The SDS(2) engine torque demand at 1500 min^{-1} and 380 seconds run time; actual torque, TT1 and PT1 in the time domain

In the case of SDS(2) tests, the region of potential fuel cutoff was deliberately avoided, which, as a consequence, has a lack of data in the negative torque demand in fig. 8. It can be seen that the ML generally has a smoother shape for TT1 and PT1, but because

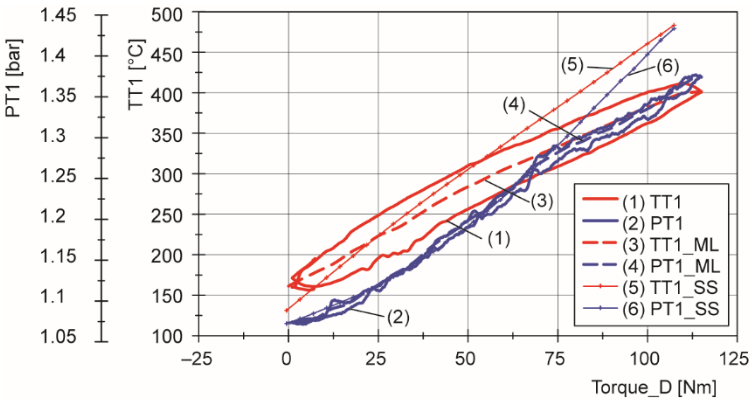

Figure 8. The TT1 and PT1 system responses; ML and steady-state also included as a function of SDS(2) system excitation for total of 380 seconds

of the omitted stationary point at the beginning of the test, and the non-existence of input hold at the maximum load, there was not enough time for the engine global temperature level to become similar in values to those that exist during stationary experimentation. This deficiency could be overcome by increasing the overall time of the SDS(2) test, and thus lowering the value of the excitation ramp gradient. As an example, in fig. 9, envelopes, ML and steady-state lines of turbine outlet temperatures (TT2) for different durations of SDS(2) test sequence are shown. It is noticed that by increasing the test time, the difference between steady state and dynamically measured data becomes smaller as in the LS1 example in figs. 2 and 3.

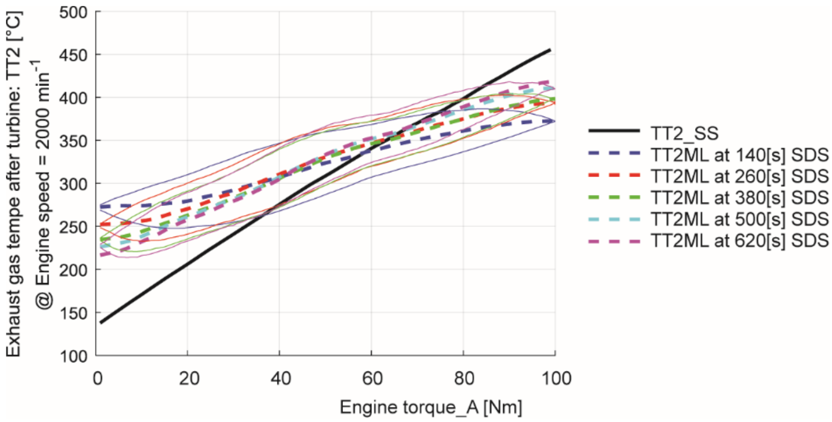

Figure 9. The TT2 steady-state, SDS(2) envelopes and ML at 2000 min^{-1} engine speed for different durations of test input sequences

Estimation of optimal SDS test duration

With the aim of determining a relation between acceptable results accuracy and the needed total SDS test time, all test results were compared. During analysis, consideration of any statistical parameter that uses data of system input and output must be performed carefully because of system non-linearity and thus, change of the output amplitudes as test duration increase. As a goodness quantification of SDS results, the approximation of simplified parameter named standard deviation of difference (SoD) is used, which is calculated as:

$$\mathrm{SoD} = std\left[norm(u) - norm(y)\right] \tag{4}$$

where u and y are functions of measured channel, engine speed and SDS type and length. As an example in fig. 10(a), SoD for TT1 is shown alongside fig. 10(b), where the envelope area of the same measurement channel is presented. In those figures, normalized values are used because absolute values do not have physical interpretation.

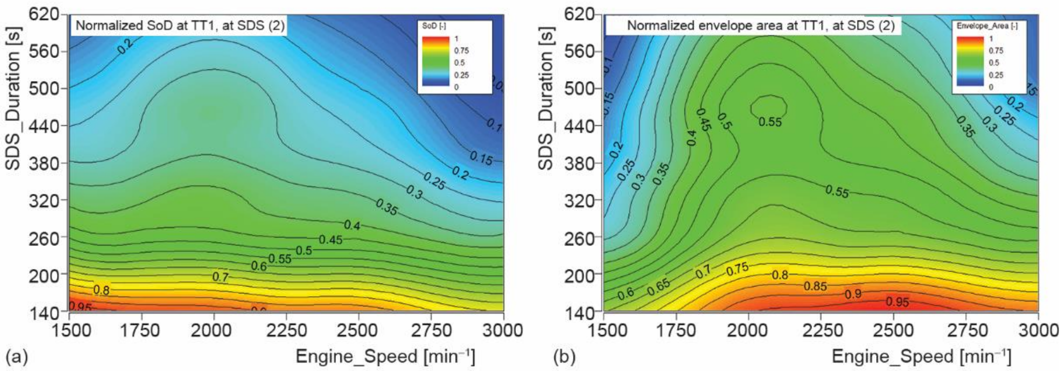

Figure 10. Normalized SoD, (a) and envelope area, (b) for TT1 as a function of engine speed and SDS(2) test duration (*for color image see journal web site*)

It is noticed that with the increase of measurement time, on the whole engine speed range, system response during rising and falling ramps become more similar, even for slow response variable, such as TT1. Also, with increasing engine speed, SDS measurement time could be reduced to match deviations on lower speed ranges, as indicated by the global trend in fig. 10(a). Although it is very suggestive, the use of envelope area analysis is not a proof of matching stationary and SDS-obtained data. Final results will be undeniably better for longer tests, but acceptable results could be obtained for fast response signals such as indication parameters (IMEP, $P_{\max}$, $AP_{\max}$), pressure measurements and fuel consumption measurement. Signals with higher response offsets, such temperatures or exhaust gas composition and opacity, need to be evaluated throughout longer SDS tests and only in the middle range of the excitation span. For extreme values of engine load, it is recommended to perform additional steady state measurements.

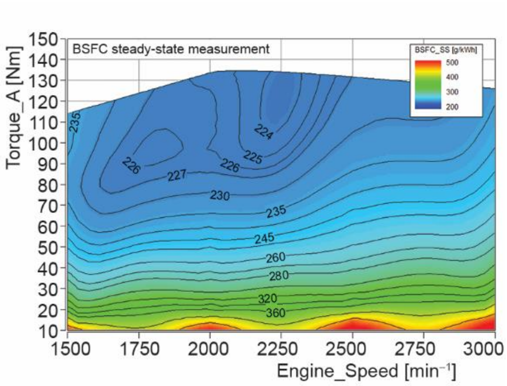

Figure 11. The BSFC for stationary engine operation
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In fig. 11, brake specific fuel consumption (BSFC) is shown for engine stationary operation in engine speed and load boundaries reached by the SDS(2) tests. This data is used for relative differences calculation between the BSFC estimated using the SDS sequences of a certain duration (140, 380, and 620 seconds), as shown in fig. 12. The fastest test has BSFC deviations of up to 5%. Test with duration of 380 seconds measurement time (at a particular engine speed) has deviation of up to 3.5% for the majority of the characteristic diagram. The slowest SDS test shows the best results with deviation of less than 2% for the majority of engine speed/load range, which is a relatively accurate result. Despite all benefits, slow dynamic slope methodology has some disadvantages, as listed:

– involvement of sophisticated hardware and software for dynamic testing,
– a lot more measurement data that needs to be evaluated and analyzed,
– limited accuracy of final results, and
– some measurements practically useless (temperatures).

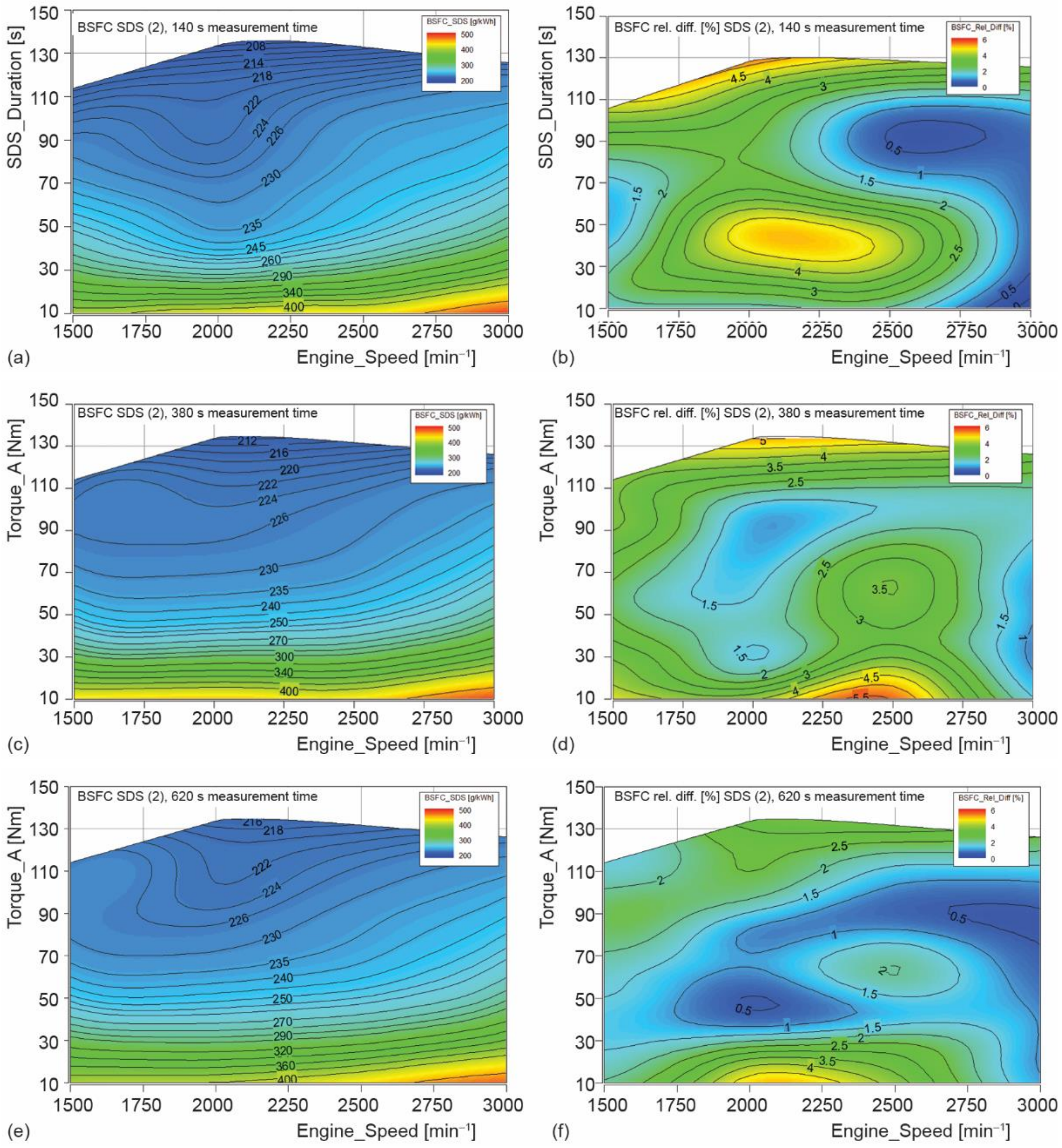

Figure 12. The SDS(2) estimated BSFC and relative differences compared with steady-state data for three lengths of SDS cycle (short, medium and long SDS duration)
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Conclusions

Dynamic testing of IC engine is an inevitable part of the its development process, especially during exhaust composition, durability and drivability optimization. Shortening time needed for approximate steady-state data collection is an imperative during development of engine mathematical models, and SDS methodology is one option of doing so. It should be noted that cumulative dynamic testing time needed for building up characteristic charts, such as in fig. 12, were roughly 30, 80, and 120 minutes, respectively, due to pre-measurement ramp excitation.

Although there are shortcomings, this method has a great potential because combined slow dynamic slopes of engine speed and load could save even more time at the engine test bench.

Nomenclature

$AP_{\max}$ – angular position of peak pressure
$P_{\max}$ – cycle peak pressure

Abbreviations

BSFC – brake specific fuel consumption
DoE – design of experiment
ECU – engine control unit
FSO – full scale output
IMEP – indicated mean effective pressure
RTD – resistance temperature detectors
SDS – slow dynamic slope

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Paper submitted: March 8, 2019
Paper revised: May 1, 2019
Paper accepted: May 3, 2019