MARC SENS (ED.)

6th International Conference on Ignition Systems for SI Engines

7th International Conference on Knocking in SI Engines

expert

IAV

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Preface

I would like to take this opportunity to thank my fantastic Scientific Advisory Board, which contributes the following thoughts to our conferences:

“These conferences bring together all the major players and experts in Combustion development sharing technical developments and new technologies to improve the efficiency of the Internal Combustion Engines. The introduction and development of alternative fuels brings up challenges which need new solutions. Such will be presented and discussed, helping OEMs to work with partners to ensure rapid and efficient introduction into production.”

Sandro Pino, Federal Mogul Powertrain, Tenneco Champion

“While electrified technologies will play a pivotal role in the future of transportation and energy, internal combustion engines (ICEs) continue to offer key advantages in many applications. Their function is likely to evolve, incorporating hybrid systems, alternative fuels, and continuous improvements in efficiency and emissions control. Conferences such as 'Ignition Systems for SI Engines & Knocking in SI Engines' provide a platform for global technical experts to exchange innovative ideas for enhancing engine performance. As these discussions and innovations continue, ICEs will remain a critical component in the transition to a more sustainable energy landscape.”

Dr. Kelly Senecal, Convergent Science

“Even two decades after the establishment of this conference series, the topics of 'ignition and combustion' are more relevant than ever. New technologies such as pre-chamber ignition and the use of alternative fuels such as hydrogen, methanol and ammonia open the door to the further development of the internal combustion engine demanded by the society.

This will not succeed without a deeper understanding of the engine combustion process, and these both conferences will make a substantial contribution to this.”

Dr.-Ing. Michael Fischer, Tenneco GmbH

“Reducing greenhouse gases in a short time period can only be reached by enabling all technological tools including a fast ramp-up of renewable fuels like methanol, ammonia or hydrogen. High ignitability causes pre ignition Problems and low ignitability causes ignition challenges. We show at these both conferences how to solve it.”

Dr.-Ing. Olaf Toedter, Karlsruher Institut für Technologie (KIT)

“The mobility sector is in its greatest transition ever. To master this challenge on a global scale, we will need all technical solutions available. Advanced combustion processes and sustainable fuels still provide big benefits for the existing vehicle fleet as well as new vehicles. This conference is the best platform for researchers from all over the world to discuss latest research results for advanced combustion and ignition technologies.”

Dr.-Ing. Michael Fischer, HONDA R&D Europe

“The future will be sustainable by electrification and hybridization, hence renewable energy carriers and electricity will be of utmost importance! But renewable energy won’t be cheap. This means that the efficiency of hybridized powertrain systems has to be higher and further developed! And there, the complete powertrain has its optimization tasks as also combustion efficiency has to be further improved!

These both conferences, organized in one event, addresses the core of ICE efficiency improvement, looking at the most important phenomena like ignition and knocking in the Otto engine!”

Prof. Dr. André Casal Kulzer, Stuttgart University

In line with the statements of the Advisory Board, we look forward to the next edition of the conference in 2026.

Marc Sens, IAV GmbH

Software-Features for optimization of ignition timing

Jakub Kaleta, Danny Jäger, Sascha Gerhardt, Carsten Kluth, Stefan Angermaier, Mukunda Gopal / Robert Bosch Group

Abstract
The continuously enhancement of the efficiency and reduction of emissions of spark-ignition engines is task of system and hardware development on one side, and function and software development on the other side. The increasing stringency of legal requirements has led to a continuous rise in engine complexity in recent years. For instance, significant efficiency improvements under stationary conditions on the test bench have been achieved through variable valve timing and low-pressure exhaust gas recirculation. Notably, the ignition timing itself significantly determines the efficiency of in-cylinder combustion.
The conventional approach for ignition timing or ignition angle calculation, primarily based on calibratable maps depending on relevant physical input signals. But this approach is increasingly limited under real driving conditions, where frequent transient operating conditions occur. This is mainly due to cross-influences of various control parameters, whose actual values often deviate from their (stationary) set positions during transient operation.
By utilizing **Software-Features** developed by the Robert Bosch Group the determination of an optimal ignition timing, across the entire parameter space to further enhance the efficiency of modern, complex engines in real driving, can be improved (see **Fig. 1**):

- Ignition setpoint determination with a data-based model (hereinafter called ***ASC@ECU IgnSp***)
- Ignition Advance by knock control (hereinafter called ***IadKnock***)
- Transient Ignition Setpoint Adaptation (hereinafter called ***TISA***)

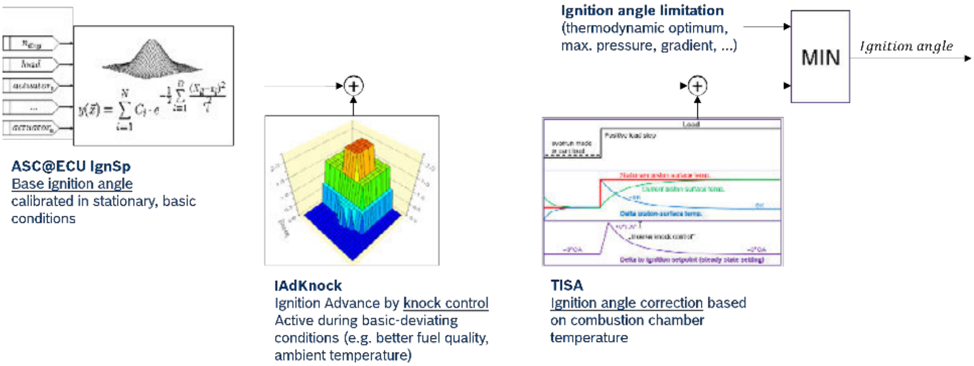

Figure 1: Overview of Software-Features for ignition timing optimization

1 Ignition setpoint determination with a data-based model (ASC@ECU IgnSp)

1.1 Introduction and Motivation

To achieve the highest emission and efficiency standards, while fulfilling high requirements of power density and dynamic qualities, modern internal combustion engines need to be operated as close as possible to the thermodynamic optimum or the knock limit under all operating conditions. At the same time the complexity of the engines is still growing and an increasing number of actuators requires more and more input parameters and dependencies to be taken into account in the ignition angle model. Achieving these goals with conventional ignition angle models based on 1- or 2-dimensional calibration maps and arithmetic calculations is increasingly difficult:

- Nonlinear dependencies between input parameters cannot be modelled with sufficient precision as illustrated by Fig. 2.
- Conventional models are accurate, but mostly under stationary conditions – under dynamic operation, when actuator positions are off the desired values, the models loose accuracy leading to increased fuel consumption and CO_2 emissions
- Adding more parameters and dependencies to the model increases the software development effort, makes the software more complex and less maintainable and also increases the calibration effort

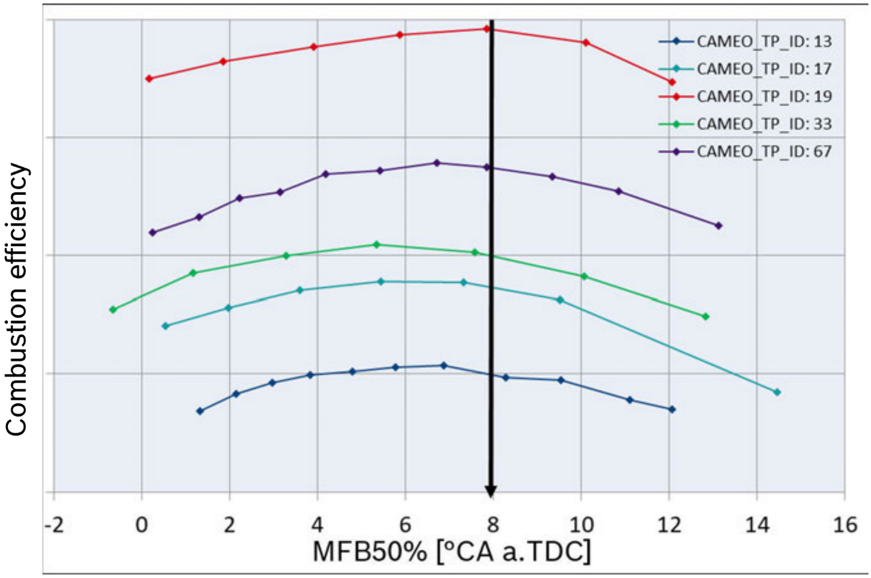

Figure 2: The center of combustion (MFB50, 50 % of **m**ass **f**raction **b**urned) needed for maximal combustion efficiency varies depending on the operating conditions as does the combustion velocity. The resulting dependency between the ignition timing needed to achieve this optimal center of combustion and the operating conditions is nonlinear and not fully described by a conventional map-based approach.

To address these points parts of the conventional ignition angle calculation chain can be replaced with a **data-based Gaussian Process Regression model** trained on engine test bench data.

For better convenience the Robert Bosch Group provides a standardized workflow and a corresponding Toolsuite stretching from designing the model and planning the necessary measurements to deployment on the engine control unit (ECU). This workflow is referred to an **A**dvanced **S**imulation and **C**alibration on the **e**ngine **c**ontrol **u**nit (ASC@ECU) and is supported by the ETAS ASCMO calibration software.

1.2 Gaussian Process Regression (GPR)

GPR is a nonparametric regression method, meaning that no explicit function structure is assumed. Rather, the model is constructed from basic assumptions about the smoothness of the unknown true function and about the measurement. Combined with measurement data this results in a probability distribution for model output. That means each model evaluation not only produces a predicted value but also gives a measure of confidence as shown in **Fig. 3**.

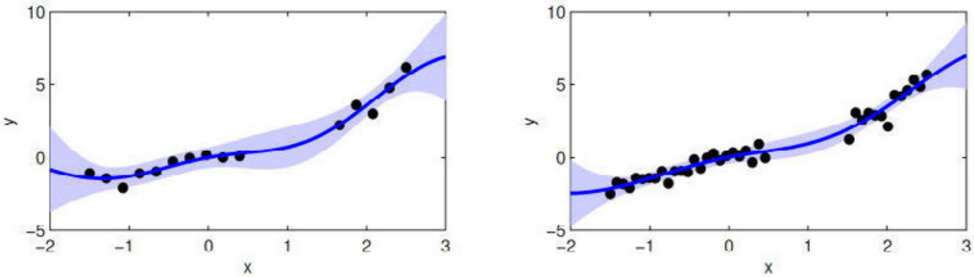

Figure 3: The confidence interval of the GPR model prediction is smaller if more data points are used (right figure) compared to sparser data (left figure).

The black dots represent measurement data, generated from an unknown true function plus an unknown amount of noise. The model prediction generated by the GPR method is shown as a blue curve. The shaded area shows the 1-sigma confidence interval about the model prediction. In the right panel, more data points were used than in the left panel, resulting in smaller confidence intervals of the model prediction. For x values around the value of 1, a gap in data was assumed. This results in smaller model confidence in this region. The same is true for x values outside of the region covered by training data (beginning and end of x axis).

Another key feature resulting from the probabilistic approach is automatic handling of the tradeoff between model complexity and generalization, which addresses the problem of overfitting.

The formula for the GPR model predictions given by:

$$v = k^{*\,T}\left(K + \sigma_n^2 I\right)^{-1}\mathbf{y} \tag{1}$$

Where K is the covariance matrix,

$$K = \begin{bmatrix} k(x_1, x_1) & k(x_1, x_2) & \dots \\ k(x_2, x_1) & \ddots & \dots \\ \vdots & \vdots & k(x_N, x_N) \end{bmatrix} \tag{2}$$

incorporating the squared exponential kernel function.

$$k\left(x_i, x_j\right) = \sigma_f^2 \bullet e^{-\frac{1}{2}\left(\sum_d \left(\frac{(x_{i,d} - x_{j,d})^2}{l_d^2}\right)\right)} \tag{3}$$

The vector $k(u)$ contains the covariances between the test point and the n training points.

$$k^{*T} = \left[k(x_1, u) \ \dots \ k(x_N, u)\right] \tag{4}$$

Here u is the input position where the model is evaluated, N is the number of training points with $x = \left[x_1 \ \dots \ x_N\right]$ (d-dimensional) and $y = \left[y_1 \ \dots \ y_N\right]$. Furthermore, l, σ_f, σ_n are additional model parameters (hyperparameters), which are determined during model training, see (Rasmussen & Williams, 2006) for details.

1.3 ASC@ECU workflow from functional concept to ECU deployment

1.3.1 Functional structure

Likewise to the conventional approach the functional concept is derived from the project specific requirements. The main goals are to select relevant input variables and design a suitable functional structure. Although a full data model calculating the final ignition angle directly from a set of input variables could be used, most real applications of ASC@ECU use a **hybrid model**. This structure combines a data-based model with conventional calculations based on calibration maps and arithmetic operations. This has several benefits:

- More simple dependencies without significant cross-influence of other relevant input variables can still be considered in the classical map- and curve-based approach.
- By reducing the model input parameters, the number of required training data / measurement points decreases.
- offset calculations for special conditions and proven in use can be retained (e.g. ambient compensation or switching from HOM to HSP injection mode)

The ASC based ignition angle calculation uses a model with typically four to six input variables like engine speed, cylinder air charge and camshaft positions, but other project specific variables like the air-fuel-ratio or EGR-rate could be added. The model is trained to calculate a calculation base for the best possible ignition angle under stationary basic conditions and then other influences like the intake air or coolant temperature are added similarly as in the conventional software.

1.3.2 Data acquisition

The data needed for model training is usually obtained during a measurement campaign on the engine testbench. The ASCMO software can be used to design an optimized measurement campaign (DoE – Design of Experiments). Depending on the number and constraints of the input parameters on the one side and the desired model precision on the other side, ASCMO automatically creates a list of necessary measurement points optimally covering the multidimensional feature space according to a Sobol Sequence.

On the other hand, the ASC model training can be also done based on existing data from previous measurements or calibration campaigns to achieve good results in an early project phase. In a later stage the model quality can be improved by adding additional data to areas with lower precision.

1.3.3 Model training

Once the measurements are available the ASCMO software is used to automatically train a GPR-model for the best possible ignition angle. The software provides convenient tools to assess the model quality and optimize the results, e.g. by identifying outliers.

By using advanced AI (Artificial Intelligence) methods from the field of machine learning, it is possible to accurately model as well as analyze and optimize the behavior of complex systems based on a relatively small set of measurement data.

1.3.4 ECU deployment

Almost all Bosch MDG1 ECUs feature a dedicated Hardware unit (AMU – **A**dvanced **M**odelling **U**nit) optimized for the evaluation of exponential functions. The AMU is used to evaluate ASC models in real time. This would be not possible on the normal ECU hardware, as the evaluation of the exponential functions would take too long.

The ASC model needs to be compressed with ASCMO for the usage in the ECU. This is done by resampling the fully trained model with a significantly lower number of artificial data points with minimal losses in model accuracy.

1.4 Demonstration of potential based on real measurements

The ASC@ECU-based (hereinafter called ASC-based) ignition angle model is already implemented in several systems with which the benefit of the approach could be depicted. In **Fig. 4** it is depicted that the model deviation for ignition angle for maximum engine efficiency of the conventional-based model is almost seven times higher than the model deviation of the ASC-based model. This means that the ignition timing is more precise and closer to the optimum leading at the end to a lower fuel consumption. The data was captured on test bench with an engine with 2.0-liter engine displacement, 4 cylinders, fully variable valve train, direct injection in homogeneous mode.

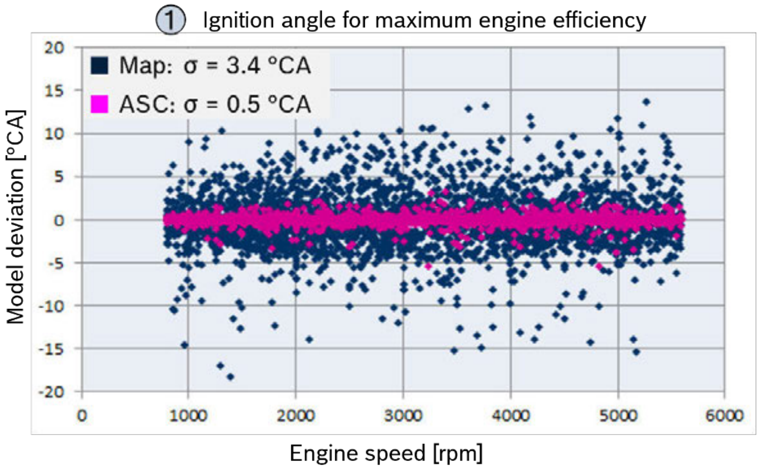

Figure 4: Model deviation for ignition angle for maximum engine efficiency

The ASC-based model shows benefits in stationary, but also in dynamic driving situations in which (for example) the air charge, engine speed and camshaft positions are changing dynamically.

The measurement data for **Fig. 5** to **Fig. 7** are derived from a powertrain test bench with a 1.5-liter engine, 4 cylinders, fully variable valve train, external low pressure exhaust gas recirculation and direct injection. WLTC stands for **W**orldwide harmonized **L**ight vehicles **T**est **C**ycle and represents a standardized test-cycle for light-duty vehicles for determining exhaust emissions (pollutant and CO_2 emissions) and fuel consumption.

Fig. 5 depicts a WLTC sequence with partially knock-limited operating conditions. The black lines depict the measurement [1] with conventional ignition angle calculation, while the green lines represent the behavior with the ASC@ECU approach [2]. The following measured parameters are depicted: Engine speed, relative air charge in the cylinder, low-pressure exhaust gas recirculation rate, MFB50, mean knock intervention, and the ignition angle difference between both approaches calculated from measurement [2]. A more precise ignition angle calculation with ASC@ECU results in efficiency advantages, evident in the more favorable center of combustion positions (MFB50).

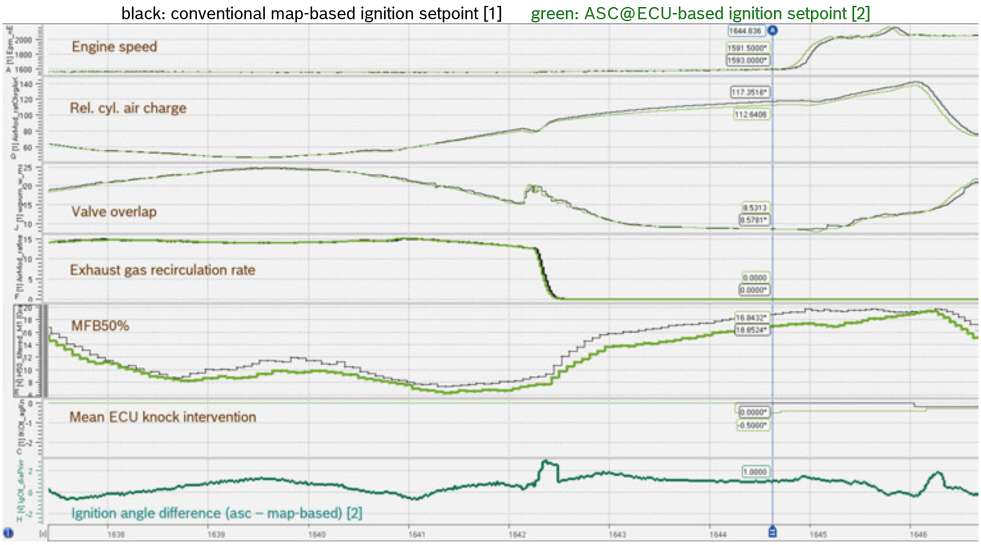

Figure 5: WLTC sequence with conventional map-based compared to ASC-based ignition angle calculation

Fig. 6 is illustrating for a WLTC-cycle that the ignition angle determined by the ASC-model is closer to the optimum, especially for lower or medium relative cylinder air charges.

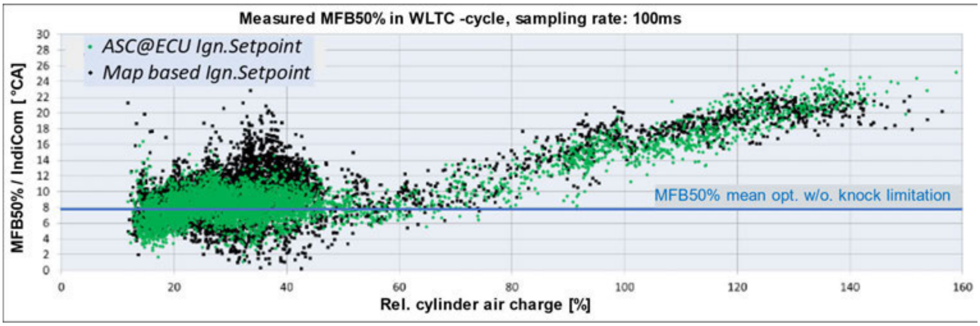

Figure 6: Measured MFB50 in WLTC-cycle. Conventional-based (black dots) versus ASC-based (green dots) ignition angle setpoints. Sampling rate: 100ms

The usage of an ASC-based ignition angle determination leads to a lower fuel consumption what can be depicted by the indicated specific fuel consumption (ISFC) in WLTC-cycle (see **Fig. 7**). It shows that especially for low and medium air charge the fuel consumption is lower with the ASC-based model.

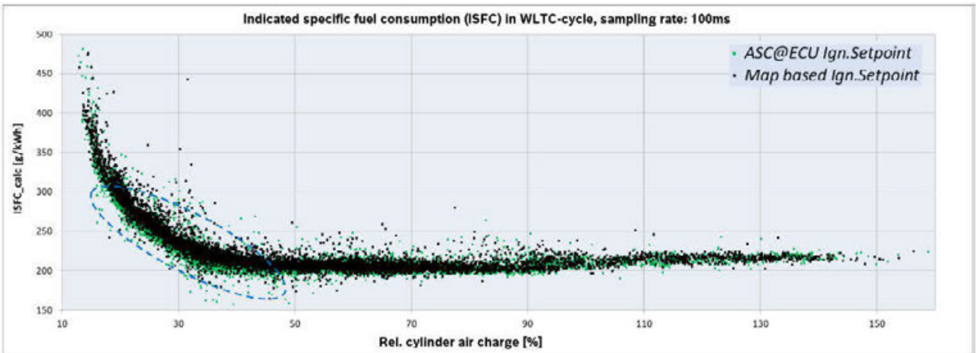

Figure 7: Indicated specific fuel consumption (ISFC) in WLTC-cycle. Conventional-based (black dots) versus ASC-based (green dots) ignition angle determination

1.5 Summary of key benefits

The ASC@ECU approach for the ignition setpoint determination has shown several key benefits over the conventional-based approach:

Reduction of fuel consumption and thus **CO_2 emissions by greater than 1 %** due to an overall improvement of ignition angle accuracy.

Reduced knocking risk under dynamic engine operating conditions.

Robustness of the ignition angle model against calibration (setpoint) changes late in the project and improved software maintainability due to standardized functional design.

Reduced software development effort for the introduction of new input parameters or new complex dependencies.

Reduced calibration effort and nevertheless an increased degrees of freedom in the parameter optimization of the parameters influencing the ignition timing.

The data models behave like maps at the edges of the parameter space. Unwanted extrapolation does not occur outside of these definable values.

Visualization can be carried out in the usual way by plotting maps with the ASCMO tool.

Applicable also for other engines, like hydrogen internal combustion engines.

2 Ignition Advance by Knock control (IadKnock)

The features ***TISA*** and ***ASC@ECU IgnSp*** can be combined with an ignition advance algorithm based on knock detection. Target of this algorithm is to operate the engine at the knock border or the thermodynamically optimal set point. The algorithm considers the center of combustion, the acceptable mean or maximum cylinder pressure and the knock frequency.

It is based on a holistic knock detection and control approach which evolves the classic knock detection further to a real optimizer of the ignition timing under all operation conditions, compensating environmental influences, as well as system tolerances and ageing effects of the engine.

The algorithm and further information are presented in detail in the paper (Benzinger & Biehl, 2022).

3 Transient Ignition Setpoint Adaptation (TISA)

3.1 Introduction

Not only under (quasi-) stationary engine operating conditions or combustion chamber temperature conditions is it important to pre-control the ignition angle optimally. Transient, dynamic operating conditions can represent a significant, fuel consumption-relevant proportion depending on the powertrain concept and driving profile. Especially during acceleration phases, in which the engine must be operated at higher loads far from the MFB50-optimum due to knock-limited ignition angle, there is often untapped potential for further CO_2 reduction. Coming from part-load or coasting operation, it can take several seconds for the combustion chamber temperature to reach stationary level after a load jump. Since the combustion chamber temperature has a major influence on mixture formation and tendency to knock, it is obvious that the ignition angle suitable for stationary conditions cannot represent the efficiency optimum for non-stationary conditions. Depending on the temperature difference to the thermally stable condition, the knock limit shifts towards earlier MFB50 with significantly better efficiencies. With the feature ***TISA*** this dependency is considered by advancing the ignition angle.

The concept is not novel. A patent from Mitsubishi dating back to 1992 describes an ignition angle calculation that considers the current cylinder wall temperature (USA Patentnr. 5150300, 1992). See excerpt in **Fig. 8**.

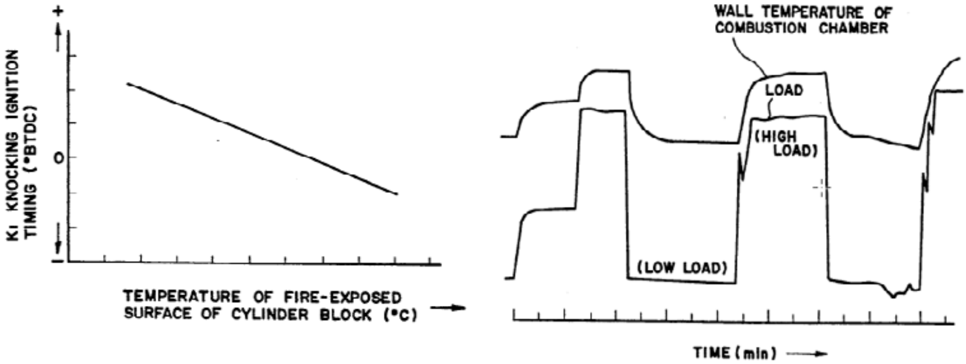

Figure 8: Extract from patent US5150300

At Robert Bosch Group, piston surface temperature is used as an equivalent to the combustion chamber temperature, instead of the cylinder wall temperature. The difference to the respective steady-state piston surface temperature is the reference variable for ignition setpoint correction. Over the years, valuable experience has been gained in the application of a corresponding temperature model for injection angle correction (TSA: **T**ransient **S**OI-**A**daptation) in various series projects. SOI stands for **S**tart **O**f **I**njection. Here, a temporary SOI correction prevents or reduces fuel wetting on the colder piston surface at the beginning of a load jump, thereby reducing particle formation during subsequent combustion. Ideally, both software features (TSA & TISA) are combined for the best possible reduction in particle emissions and fuel consumption during dynamic engine operation.

3.2 Functional Approach

Fig. 9 schematically illustrates the functionality of TISA. The suitable spark advance is promptly considered depending on the engine speed, cylinder air charge, and temperature difference. If there is a (very short term) dynamic intervention of the knock control (in retarded direction), this can be considered, and TISA is enabled with a slight delay. Depending on the gradually decreasing piston surface temperature difference, the positive ignition angle offset is stepwise reduced. In the event of detected knock control interventions, the spark advance can be prematurely reduced or terminated on a cylinder-individual basis, depending on the intensity of the knock control intervention. The speed of this ignition angle adjustment must be limited to the rate at which the air system can compensate for the corresponding change in desired air charge. This will prevent any unwanted noticeable torque fluctuations.

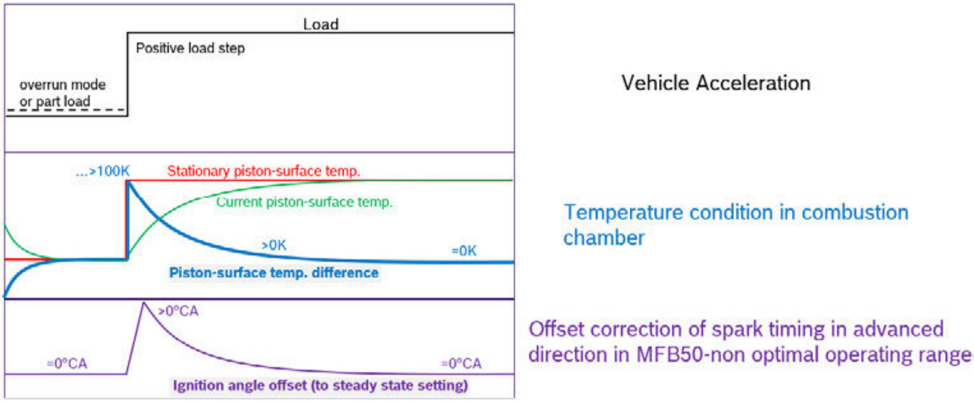

Figure 9: Schematic representation of the functionality of TISA

In combination with an advanced ignition angle correction for variable environmental or fuel conditions (engine temperature, intake air temperature, humidity, fuel octane rating), it can be achieved that the ignition timing is set as close as possible to the knock limit in order to optimize the fuel efficiency for different conditions.

In this context, it becomes clear that simultaneously acting ignition angle offset corrections (in both directions) must be coordinated with each other. Different physical influences must be considered separately in the ECU software to ensure optimal system behavior under all conditions. A known example of this is the intake air temperature, which, like the combustion chamber or piston surface temperature, has a significant influence on the shifting of the knock limit.

At a higher level, there is the functional option to apply a spark advance limit that is dependent on operating point and calibration parameters. This limit can, for example, represent the respective thermodynamic ignition angle optimum, a maximum permissible peak cylinder-pressure, or a maximum tolerable combustion pressure gradient. When using the ***ASC@ECU*** approach for the stationary base ignition angle, this data model-based advance limit is already included. Alternatively, a conventional functional structure (map-/curve-based) can be used. All these options are available in the platform software of the Robert Bosch Group.

3.3 Piston surface temperature model

The piston surface temperature is provided by an appropriate model in the engine control unit (ECU). The model itself will not be further discussed here. To calibrate the model, a suitable reference signal is required.

The measurement of piston surface reference temperature is indirectly carried out using an infrared temperature measurement system from *FOS Messtechnik GmbH* (**Fig. 10**). For this purpose, a special spark plug with optical access is installed in a suitable cylinder. The signal recording can be crank angle-resolved using an existing indication system.

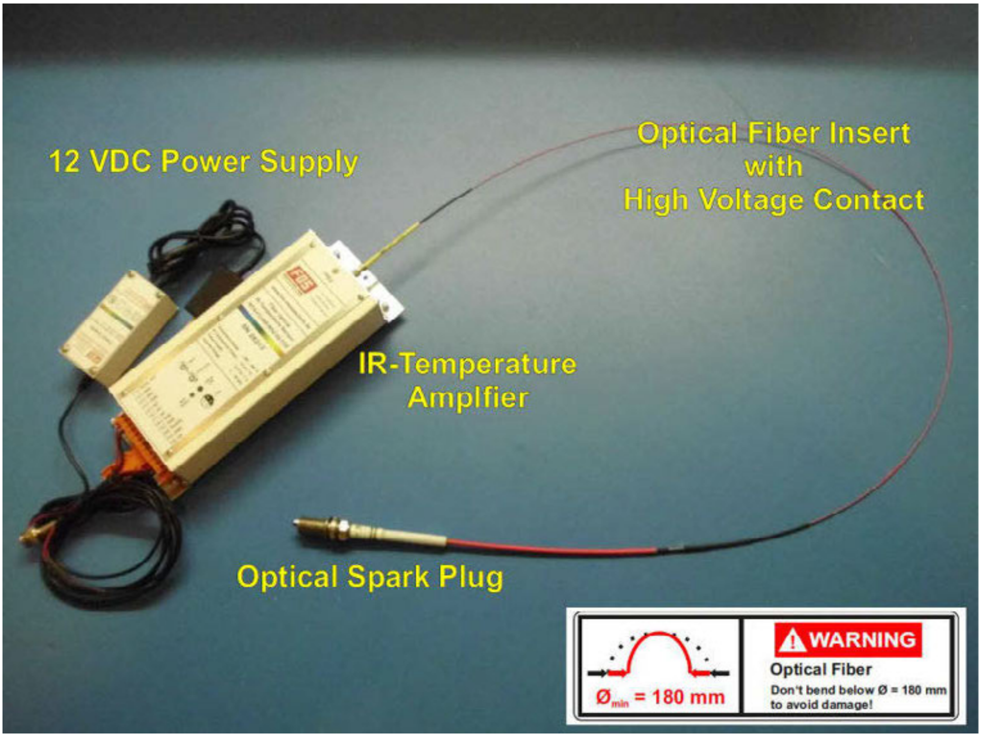

Figure 10: Measuring system used for piston surface temperature

3.4 Proof of concept for piston surface temperature model

Regardless of the starting condition a positive load jump into the knock-limited range occurs, the actual temperature profile must be accurately mapped. Only then can a suitable spark advance be determined based on the temperature difference between the stationary and current state. Without this, efficient calibration of ignition angle adjustment for unsteady states becomes challenging. An underestimated piston surface temperature causes more difficulties than an overestimated temperature. If the piston surface is hotter than the model value suggests, the calculated ignition angle may be too early, leading to unwanted heavy knocking interventions on several cylinders at the beginning of the load jump.

On the other hand, the ECU knock detection also offers the possibility to validate and adjust the piston surface temperature model if necessary. **Fig. 11** shows a load jump sequence performed on the engine test bench at constant speed (3000 rpm), constant start load (1) and constant target load (3). The ignition angle for this target load was exemplarily advanced by 3°CA compared to the stationary. In this test, the initial value of the piston surface temperature has been varied by different durations of the **F**uel **C**ut **O**ff phase (2), hereinafter called FCO. The expectation was that the first knock events always occur above a certain threshold of the modelled surface piston temperature.

All engine measurements shown below were performed on a 2.0L direct injection, turbocharged engine with variable intake and exhaust camshaft.

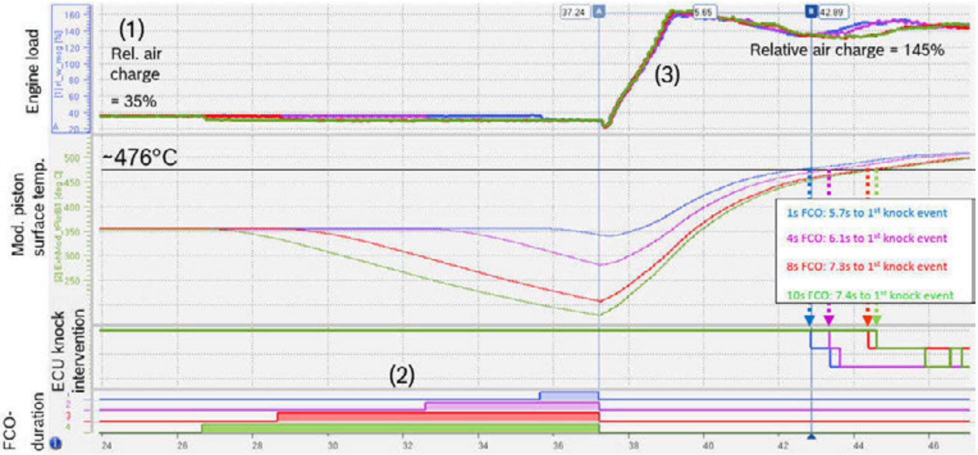

Figure 11: Variation of FCO duration before a load jump into knock limited area at constant engine speed (3000 rpm)

As expected, the first knock events provoked by the constant ignition angle offset always occurred at a very similar modelled piston surface temperature (475-478°C), which was reached after different durations at defined high load condition, depending on the preceding overrun time. This general behavior was also observed in other load jump sequences, demonstrating the fundamental applicability of the TISA functionality.

3.5 Demonstration of potential based on real measurements

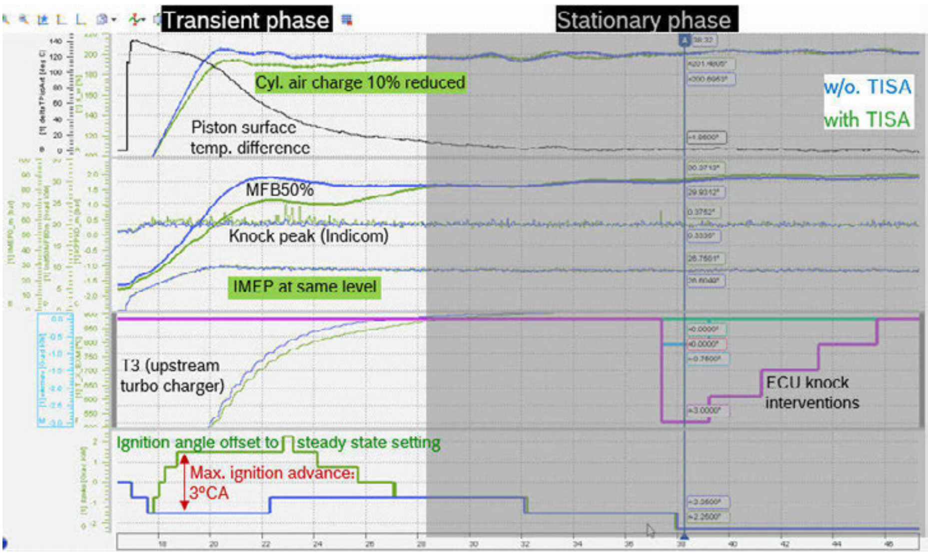

Figure 12: Load jump into full load at constant 2500 rpm after FCO phase; 2.0L GDI, TC, VVT engine

Fig. 12 exemplifies a load jump performed on the engine test bench from the FCO-operation to full load at a constant engine speed (2500 rpm). The ignition angle was adjusted by a maximum of 3 degrees advancing in the knock-limited range compared to the stationary optimum. The TISA intervention (green) was enabled for approximately 9 seconds. For comparison, all signals without TISA are shown in blue. By advancing the MFB50, the cylinder charge required to achieve the desired torque was significantly reduced, resulting in a 5 % decrease in fuel consumption during this transient phase. In addition, the exhaust gas temperature could be lowered by approximately 35K.

A second example (**Fig. 13**) illustrates a sequence of the ADAC highway cycle BAB130. The activation conditions for TISA are frequently met through repeated deceleration and subsequent acceleration to 130 kph. The computed difference of the piston surface temperature (in blue), the TISA ignition angle intervention (in light green), and the relative fuel mass with (in green) and without TISA (in black) are depicted. The required fuel quantity could be significantly reduced during the TISA activation periods with comparable indicated mean pressure (IMEP), resulting in an overall 1.7 % reduction in CO_2 emissions through a maximum ignition timing advance of only 3°CA.

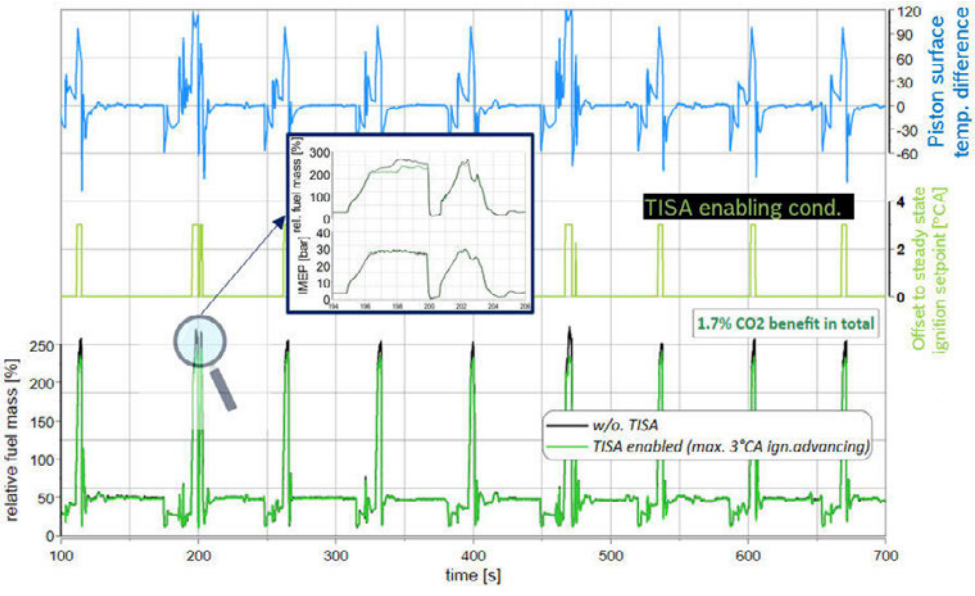

Figure 13: Sequence of BAB130 test cycle, potential fuel consumption reduction through a max. spark advance of only 3°CA

Not only fuel consumption and thus CO_2 emissions can be reduced during dynamic engine operation through TISA, but also the particle number (PN) of the raw exhaust gas can be decreased. As the engine can be operated at better ignition angle efficiency during load jumps, the cylinder charge and thus the injection quantity can be reduced. This often has a positive effect on soot particle number reduction, especially during dynamic engine operation. Example with a 20 % reduced relative cylinder filling see **Fig. 14**.

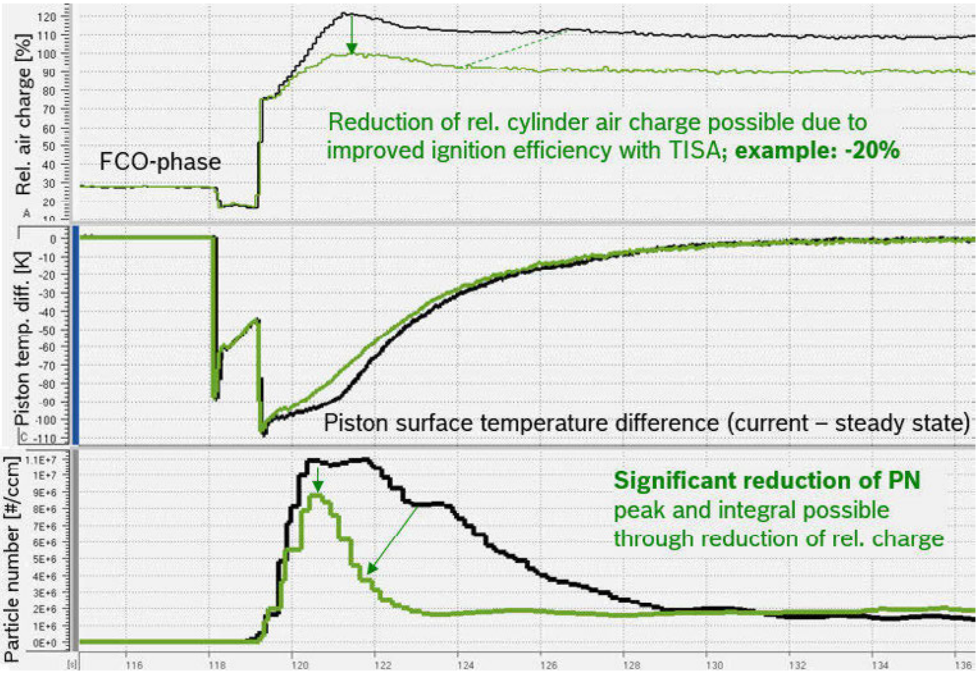

Figure 14: Potential of PN-reduction during load jump at 1750 rpm through reduction of relative cylinder charge by 20 %

3.6 Summary of the benefits and positive side effects

With the ***TISA*** software feature at the Robert Bosch Group, it is possible to close the gap between stationary ignition setpoint calibration in the knock-limited operating area and the consideration of actual temperature conditions inside the combustion chamber during dynamic engine operation. In summary, the advantages are as follows:

- Improved fuel efficiency in non MFB50-optimal operating area means a considerable reduction of CO_2 emission under real driving condition.
- Potential to reduce raw emissions (PN, CO).
- Improved engine response and performance in dynamic driving.
- Enhanced drivability in the low-end torque range; torque build-up is more reliant on ignition angle efficiency than on cylinder air charge build-up, resulting in a more harmonious acceleration.
- More stable and robust detection of low octane (poor fuel quality) for ECU knock control.

4 Conclusion

In the chapters before it has been demonstrated that there is still potential to increase the efficiency of modern complex spark-ignited combustion engines significantly just by adding and calibrating **innovative software features for ignition control**. These features have been validated in several projects. The validation has shown benefits of the features individually but also in the combination. These and further software solutions and services are provided by the Robert Bosch Group.

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On the Origin of Pre-Ignition inside a Pre-Chamber Spark Plug – Gas Analysis

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Abstract

Pre-chamber spark plugs are widely used to increase the efficiency of internal combustion engines in order to meet future CO_2 emission regulations. Unfortunately, pre-ignition at higher load operation points in combination with pre-chamber ignition systems has been reported in the past [1]. Two main assumptions for pre-ignition are (1) pre-ignition to be triggered either by hot pre-chamber surface components or (2) by hot and still reactive residual gases present inside the pre-chamber from previous combustion cycles. Assumption (1) was examined in a previous publication [2] and not found to be the leading cause of pre-ignition. However, with the help of optical pre-chamber spark plugs, it was discovered that the origin of the pre-ignition is located in the upper part (around the insulator) of the pre-chamber. For this reason, the gas composition in the upper part of the pre-chamber is to be examined in detail and the CFD calculations thereby validated.

The objective of this study is to analyze the residual gas composition inside a pre-chamber spark plug while running in an engine. For this purpose, a heavy-duty natural gas engine, that is operated with a M14 pre-chamber spark plug, will be run purposely up to a point where pre-ignition occurs. A small gas probe is to be sampled from the internal pre-chamber spark plug volume at various times during the combustion cycle and gas analytical measurements are to be performed. The gas is extracted from the area identified in the previous publication as the source of the pre-ignition. The sampling is controlled by a fast-switching valve so that the sampling time and duration can be selected precisely. In order to achieve the necessary sensitivity for measurement of the low sampling volume flow, the sample is diluted with nitrogen in a controlled manner.

In addition to carbon dioxide, unburned methane was also detected in the pre-chamber spark plug throughout the entire combustion cycle. These findings corroborate the hypotheses proposed in the previous publication, namely that the upper area of the pre-chamber spark plug only undergoes combustion at a relatively late stage, which allows for the presence of unburned gases and radicals that can lead to pre-ignition. However, further measurements are necessary to obtain a comprehensive analysis.

1 Scope and Motivation

The automotive industry has been engaged in research into pre-chamber combustion technology with the objective of enhancing the efficiency of gasoline combustion engines for a number of years now. The technology can be classified into two concepts: passive and active pre-chamber spark plug combustion. In passive pre-chamber spark plug technology, the fuel is mixed with air inside the main combustion chamber (engine cylinder) and then pushed into the pre-chamber during compression. In contrast, in active systems, fuel is introduced directly into the pre-chamber. The concept of passive pre-chamber combustion is regarded as the cost-optimal solution in the medium term due to its reduced complexity in comparison to the active pre-chamber system. For an overview of the literature on passive pre-chamber combustion, please refer to [3, 4].

The utilisation of passive pre-chamber spark plugs in series production presents a number of advantages, including the potential for faster combustion and lean-burn capability. However, the necessity for their functionality in challenging marginal areas must be considered. Such conditions include the operation under high loads and the heating of the three-way-catalyst (operating the engine under low loads with delayed combustion center positions). At elevated loads, the potential operating range of pre-chamber plugs can be highly constrained, such that even a minor alteration in the ignition timing, by a single degree of crank angle, can result in the initiation of strong, consecutive pre-ignitions. It is only possible to control these pre-ignition series by cutting off the fuel supply.

The two primary assumptions regarding pre-ignition are as follows:

(1) pre-ignition can be triggered either by hot pre-chamber surface components or
(2) by hot and still reactive residual gases present inside the pre-chamber from previous combustion cycles.

In a previous paper [2], the hypothesis that component temperatures are the cause of pre-ignition was tested with the aid of thermal measuring spark plugs, which yielded results that did not support this hypothesis. However, investigations utilising optical pre-chamber spark plugs have demonstrated, that the source of the pre-ignition phenomenon is situated in the upper region of the pre-chamber spark plug. The aforementioned volume is insufficiently scavenged in the pre-chamber spark plug under examination. It is conceivable that combustion residues are not adequately flushed out, which could result in pre-ignition during the subsequent cycle. To validate this thesis and the preceding CFD calculations on scavenging, it is necessary to extract and analyze the gas in the upper area of the spark plug while the engine runs in high load close to pre-ignition operating point. Preceding CFD calculations indicated the presence of a prolonged combustion flame in the region of interest, well after top dead center.

The objective of this study is to establish a measurement system for the analysis of gases in the upper region of a pre-chamber spark plug, with the aim of identifying the source of pre-ignitions that occur under conditions of increased load.

The following working hypotheses have been formulated:

1. Unburned and reactive gases from the main combustion chamber are pushed into the pre-chamber volume.
2. The unburned/reactive gases from inside the pre-chamber remain within the system until the beginning of the subsequent cycle.

Both hypotheses assume that the compression of the remaining reactive gases and the fresh methane which is pushed into the pre-chamber leads to self-ignition.

2 Engine and Pre-Chamber Description

2.1 Engine and Test Bench Characteristics and Operating Point

The engine utilized is a single-cylinder medium-duty diesel engine (type BR2000) manufactured by MTU. The engine has been modified to function as an intake manifold-injected spark-ignited engine. In place of the diesel injector, a spark plug is positioned within the cylinder head, and the compression ratio is adjusted to 12.5:1 through the modification of the piston. The fuel utilized is natural gas drawn from the local natural gas network.

The utilization of a medium-duty engine allows for a detailed examination of pre-ignition phenomena due to its inherent durability. Consequently, the engine can be operated for a certain duration in areas, where pre-igniting combustion occurs.

The operating point was selected to replicate the conditions of the previous test series conducted by Rosenthal et al. [1] in 2018, ensuring high reproducibility. The following table illustrates the operating point at which pre-ignitions can be generated in a reproducible manner.

Speed / rpm	1500
Mean indicated pressure / bar	12.3
MFB50 / °CA aTDCf	≤ 8
Energy from fuel per cycle / kJ/cycle	5.9
Lambda / -	1.45

Tab. 1: Operating point with occurring pre-ignitions

The engine is operated on a modern engine test bench, which allows for the setting of reproducible operating conditions. The fuel gas is subjected to continuous analysis by a gas chromatograph in order to enable the compensation for fluctuating calorific values, thus allowing the establishment of a reproducible operating point. The implementation of automatic, pressure-based safety monitoring ensures that short-term operation in pre-ignition conditions can be conducted in a safe manner.

2.2 Pre-Chamber Spark Plug

The passive pre-chamber spark plug utilized in this investigation was developed for medium-sized gas engines in which M14 pre-chamber spark plugs can be employed. A distinctive attribute of this passive pre-chamber spark plug is the incorporation of a ring-shaped ground electrode, comprising a flat disk with a central aperture containing a ring carrier with precious metal. In conjunction with an elongated and protruding center electrode situated within the center hole of the ground electrode disc, a ring-shaped spark gap measuring approximately 0.3 mm is formed. This electrode configuration was designed with the objective of ensuring a long service life, and a center electrode with a copper core allows rapid heat dissipation. The ground electrode disk is positioned between the end of the shell and the pre-chamber cap. It is attached using laser welding. Three supplementary kidney-shaped apertures are incorporated into the design of the ground electrode to facilitate enhanced gas exchange between the lower and upper sections of the pre-chamber volume. Prior research [1,2] has indicated, that the aforementioned holes are inadequate for the complete scavenging of the pre-chamber spark plug, and instead serve to impose an aerodynamic resistance.

Furthermore, the pre-chamber cap is equipped with four radially arranged holes, each with a diameter of 1.2 mm. The center lines of the holes are oriented in a direction aligned with the central electrode tip. Additionally, the core nose of the insulator and the shell ridge on which the insulator rests are also situated in the upper part of the pre-chamber. The total volume of the pre-chamber is 813 mm^3, with 322 mm^3 between the inner surface of the cap and the ground electrode ring (lower section) and 491 mm^3 behind the ground electrode ring (upper section). For further information regarding the spark plug characteristics, refer to [2].

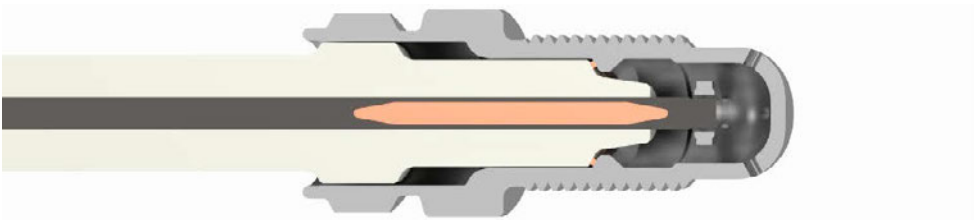

Fig. 1: Pre-chamber spark plug

3 Measurement System for Gas Anaylsis

3.1 Gas-Extraction Pre-Chamber Characteristics

In order to obtain reliable measurement results as close as possible to the original series system, the gas extraction channel has to be installed into a series M14 spark plug with only minimal changes to the geometry of the pre-chamber volume.

Furthermore, the extraction process must be precisely controllable in order to facilitate the extraction of gas at a defined point in the combustion cycle. This necessitates the use of a fast-switching valve. To avoid any distortion of the gas composition, it is essential to maintain the shortest possible length and volume of the line between the pre-chamber and the valve. However, the diameter has not be too small, as otherwise the friction losses will impede the extraction process. In order to satisfy these requirements, a capillary with an internal diameter of 1.8 mm was attached to the body of an M14 pre-chamber spark plug by laser welding. Holes of 0.8 mm diameter are drilled through the spark plug body terminating in the upper volume surrounding the core nose. Due to packaging limitations of the cylinder head, the gas sampling valve cannot be positioned in closer proximity than ~9 cm to the sampling point in the pre-chamber.

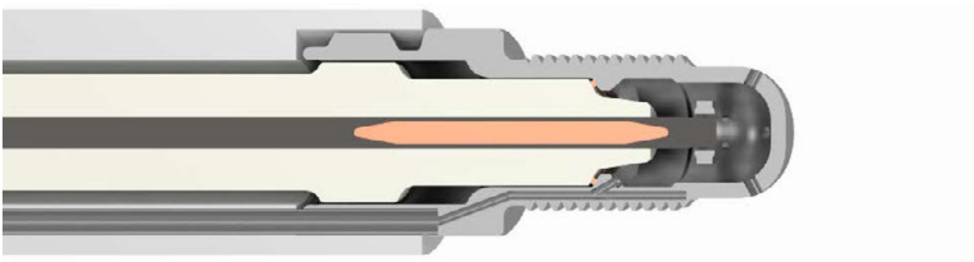

Fig. 2: Gas-extraction pre-chamber spark plug

To ensure the validity of the operating conditions, it is necessary to extract only a minimal quantity of gas, as otherwise the spark plug would be actively scavenged, thereby falsifying the operating conditions. As a preliminary estimation, approximately 1 % of the pre-chamber volume, or 8 mm³, should be removed per cycle. Assuming an extraction pressure of 30 bar and a motor speed of 1500 rpm, the resulting extraction flow is 180 ml/min. To maintain the maximum 1 % volume extraction, the extraction duration must be adjusted depending on the pressure at the time of extraction. The aforementioned sample gas flow is insufficient for the purposes of exhaust gas analysis. Accordingly, the sample gas is diluted with nitrogen. A dilution section has been implemented for this purpose, which regulates a nitrogen volume flow to which the sample gas is added. The total flow of sample gas and nitrogen is subsequently measured and controlled. The dilution is quantified using the following formula:

$$f_D = \frac{\dot{V}_{tot}}{\dot{V}_{PC}}$$

where f_D is the factor of dilution, $\dot{V}_{tot}$ is the total extraction gas flow and $\dot{V}_{PC}$ the gas flow extracted from the pre-chamber spark plug.

Subsequently, the diluted sample gas is fed into a FTIR (Fourier Transform Infrared Spectroscopy) spectrometer for analysis, with a focus on identifying and quantifying 28 specific components typical for CNG combustion. This analysis is conducted and documented at a rate of 5 Hz on a continuous basis. To obtain the concentration of the components in the undiluted gas, the individual concentrations are multiplied by the dilution factor. It is important to ensure that the gas transit time from the dilution section to the FTIR is considered. Using repetitive combustion cycles and pressure signals, the gas analysis measurements are projected onto the crank angle of the engine.