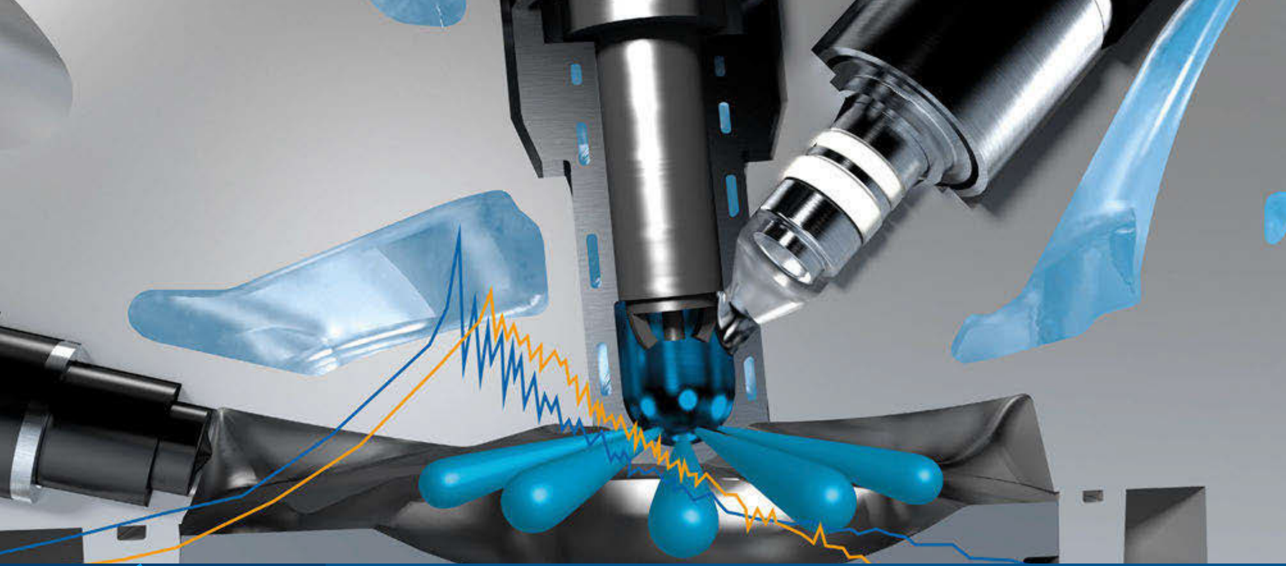

MARC SENS (ED.)

International Conference on Ignition Systems for Gasoline Engines

International Conference on Knocking in Gasoline Engines

expert

IAV

International Conference on Ignition Systems for Gasoline Engines –
International Conference on Knocking in Gasoline Engines

Marc Sens (ED.)

International Conference on Ignition Systems for Gasoline Engines – International Conference on Knocking in Gasoline Engines

expert

Bibliografische Information der Deutschen Nationalbibliothek
Die Deutsche Nationalbibliothek verzeichnet diese Publikation in der Deutschen Nationalbibliografie; detaillierte bibliografische Daten sind im Internet über http://dnb.dnb.de abrufbar.

DOI: https://doi.org/10.24053/9783816985440

– ein Unternehmen der Narr Francke Attempto Verlag GmbH + Co. KG
Dischingerweg 5 · D-72070 Tübingen

Internet: www.expertverlag.de
eMail: info@verlag.expert

printed in Germany

ISBN 978-3-8169-3544-5 (Print)
ISBN 978-3-8169-8544-0 (ePDF)
ISBN 978-3-8169-0112-9 (ePub)

MIX
Papier aus verantwortungsvollen Quellen
FSC® C083411

Inhalt

About Knock and Ignition

Ignition System Basics

Ignition System Basic II

Active Pre Chamber I

Active Pre Chamber II

Knock Detection / Criterion / Control

Pre Ignition / Combustion Phenomena

Pre Chamber III

Knock Detection / Pre Ignition II

Knock in SI-Engines – A continuing challenge for combustion system development

Dr. Frank Altenschmidt, Dr. Eberhard Kraus; Mercedes-Benz AG, Germany

Abstract: Since the invention of gasoline engines, the phenomenon of knocking combustion has been known. Particularly in the first half of the 20th century knock has been promoted by poor fuels, nowadays high power densities of engines and increasingly strict emission legislation, which prohibit substoichiometric operation, are mainly responsible for it. Alleviating measures, which lead to an increase in CO_2-emissions, e. g. decrease of compression ratio, can't be taken due to demanding consumption targets. Engine knock is primarily controlled by the thermal conditions in the combustion chamber. Acceleration of flame speed can be achieved by increasing the charge motion. However, this leads to higher peak temperatures in the combustion chamber, coupled with a higher heat loss to the combustion chamber walls. If the burning rate is too fast, this mechanism can even lead to a deterioration of the knock limit. In order to reduce the gas temperatures in particular during combustion, the use of cooled residual gas is advantageous. This enables significant earlier combustion, leading to comparable gas temperatures and wall heat losses at the knock limit. From this it follows there is a thermal limit for a given combustion chamber configuration that cannot be exceeded without further measures such as cooling or water injection.

Stuttgart, September 2022 Dr. Frank Altenschmidt, Dr. Eberhard Kraus

1 Introduction

Ever since Christian Reithmann was granted a patent for a four-stroke gasoline engine in 1860, it has been impossible to imagine our daily lives without this type of engine. There is probably no other engine that, after 160 years, can be found in every corner of the world and performs its service reliably under such climatically diverse conditions.

However, the demands on reliability, performance, consumption and emissions have changed fundamentally. In the first decades after the invention of the internal combustion engine, the focus of development was on improving mechanical durability. When this reached an acceptable level, more efforts were made to increase the performance of the engines, which also brought the phenomenon of knocking combustion with it. Even though at the beginning of the twentieth century the measuring techniques were far from being available as they are today, the cause of engine knock could be determined. In his textbook “Motorwagen und Fahrzeugmaschinen für flüssige Kraftstoffe” from 1925, Dr.-techn. Arnold Heller [1] describes the process of combustion including the emergence of knocking working cycles. Since pure gasoline was in short supply, especially during the Great Depression after World War I, engine damage due to knocking combustion occurred more frequently because of the poor fuel quality of substitute fuels. The investigations at this time by Thomas Midgley (inventor of tetraethyl lead) and others at General Motors and the findings derived from it led Arnold Heller to the statement that “knocking has completely lost its dangers in practice”. Even though technical progress has been unmistakable since that time, the problem of knocking combustion still represents a limit to the use of high compression ratios at optimum centre of combustion at full load.

Particularly in turbocharged series engines, the aim is to use the highest possible compression ratio to achieve good part-load efficiency. However, if this is chosen too high, the losses at knock-limited load points can very quickly cancel out the advantages at part load. Figure 1 shows that very high losses occur in the case of late centres of combustion, which are much higher than profits in the low percentage range due to a high compression ratio.

At the same time, late centres of combustion lead to high exhaust temperatures, which are counterproductive with respect to maximum turbine and catalytic converter temperatures. In the past, therefore, the fuel/air mixture was adjusted substoichiometric to reduce process temperatures, which is no longer permissible today [2].

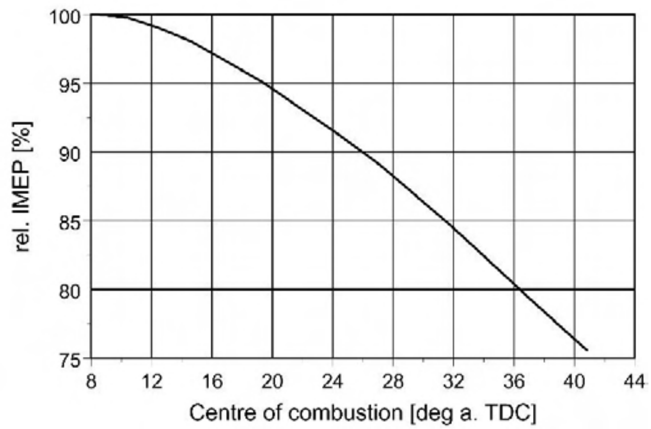

Figure 1: Correlation between COC and IMEP in full load operation

Figure 2 shows a typical engine map with areas of stoichiometric and substoichiometric operation. The area, where enrichment is necessary, is small compared to the whole map. It follows that measures and technologies that enable stoichiometric operation over the entire map may only degrade the other operating range to a very small extent, if at all. Therefore, for example, lowering the compression ratio to improve the knock limit is no option in present times. In contrast, increasing the charge motion to accelerate combustion or using cooled residual gas to improve the centre of combustion and efficiency and thus lower the exhaust gas temperature are interesting measures.

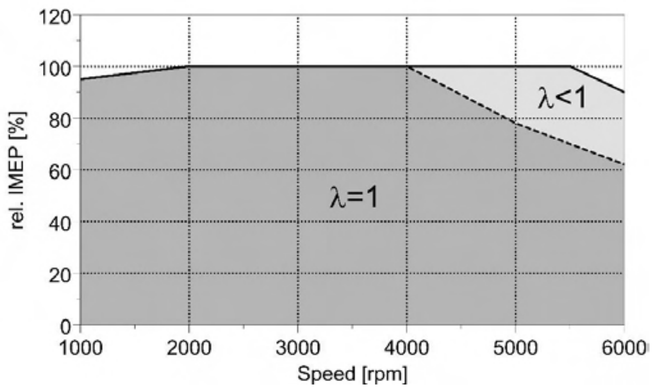

Figure 2: Engine map with λ=1 operation limit

This article examines the influence of charge motion, cooled residual gas or a combination of both on the working process of the gasoline engine.

2 Theoretical Examination

2.1 Fundamentals

With increasing supercharging, knocking combustion is the main obstacle for thermodynamically optimal engine operation. A large number of research projects already initiated on the subject of knocking (e. g. [3], [4]) bear witness to the fact that this topic has not lost its topicality and will not lose it any time soon. Therefore, precise knowledge of the processes in the combustion chamber that lead to knock is vital for further improvement of combustion efficiency especially at full load.

Up to the 80s of the last century, two different theories were developed to explain the origin of engine knock. The detonation theory describes the onset of knocking combustion as an acceleration of the primary flame [5, 6]. The largely recognized self-ignition theory assumes secondary ignition ahead of the primary flame, which together determine the further combustion process [7, 8]. A characteristic feature of the chemical reactions which lead to eventual auto ignition is the ignition delay time. This time interval is strongly dependent on pressure, temperature and the mixture composition [9].

In a variety of publications, it was attempted to capture these very complex interactions with knock criteria [e. g. 3, 10, 11, 12, 13]. However, there is no model which can adequately capture the local mixture states and thermal conditions in the combustion chamber sufficiently to predict engine knock under all circumstances. This shows the needs for further research work regarding improvement of engine knock behaviour.

2.2 Impact of water injection and cooled EGR on the thermodynamic cycle

In order to be able to use stoichiometric mixture composition throughout the entire engine operating map, the gas temperature upstream of the turbocharger turbine and the catalytic converter must not be too high. There are several possibilities to achieve this, all of which in principle rely on the same mechanism. This mechanism is shown in Figure 3 by means of water injection and cooled exhaust gas recirculation.

In case of water injection a liquid medium is injected into the inlet port, usually during the intake stroke. As shown in [14], evaporation takes place during late compression and during the combustion phase. The evaporation cools the gas and increases the mass and heat capacity of the cylinder load. Both lead to an improvement in the knock limit, which additionally lowers the exhaust gas temperature.

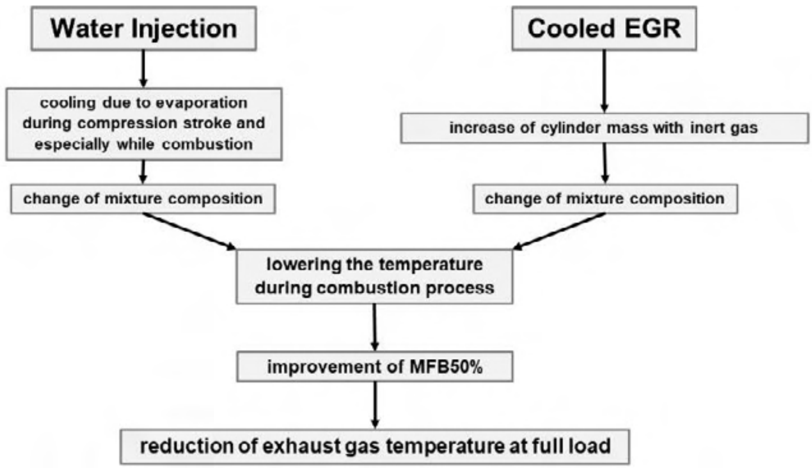

Figure 3: Effect chain of water injection and colled EGR on the SI-engine working cycle

When using cooled residual gas, it is mixed with fresh air during the intake stroke, increasing mass and heat capacity of the cylinder load. Since this is an inert gas, it doesn't participate in the combustion process. However, the released heat is distributed over the now larger mixture mass in the cylinder, resulting in lower process temperatures and thus a better knock limit.

With increasing residual gas fraction in the combustion chamber the oxygen concentration and thereby the flame speed decreases. The question arises whether this should be compensated by an increase of the charge motion generated by the intake port and what influence this has on the combustion cycle as well as the knock limit. These questions will be examined hereinafter in more detail.

3 Experimental Results

3.1 Experimental setup

A research single-cylinder engine was used for the experiments whose combustion chamber geometry is derived from the Mercedes-Benz 6-cylinder engine M256.

Three cylinder heads with different intake ports were used. Figure 4 shows the tumble curves, which were determined by CFD-simulations. The base configuration Tz_1 already has a high tumble at the level of the M256 production engine. Starting from this, the tumble levels are increased by 25% in two steps. With each variant a series with increasing load were measured both with and without cooled EGR, where an HD-AGR system is used. Because the three intake ports have different flow resistances due to the different charge motion levels, all tests were carried out with a constant differential pressure of 200 mbar between the intake and exhaust port to keep the internal residual gas rate almost constant. All series were measured at knock limit and an inlet port temperature of 45°C.

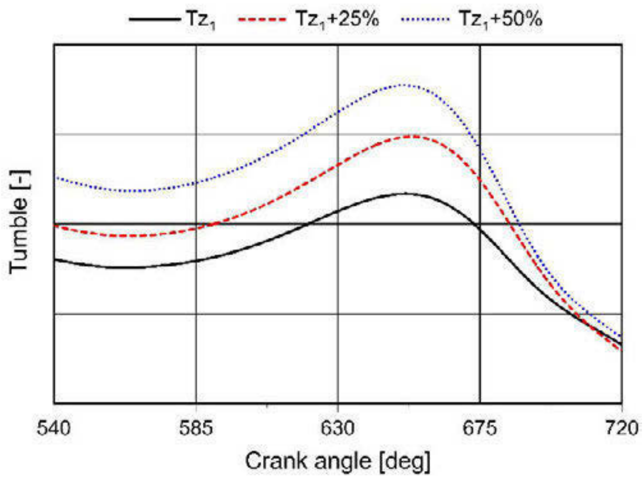

Figure 4: Comparison of the used tumble levels

3.2 Influence of charge motion on the knock limit

Figure 5 shows curves of 50% mass fraction burned (MFB 50%) for all investigated tumble levels at three different engine speeds and increasing load.

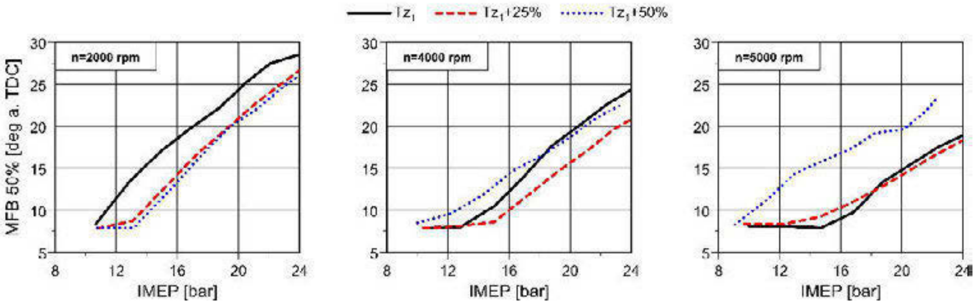

Figure 5: Influence of tumble level on the knock limit for different engine speeds at λ=1

At 2000 rpm the behavior is as expected. The increased charge motion leads to an improved knock limit by 4°CA, with no further improvement with the highest tumble level Tz_1+50%. At 4000 rpm there is still an improvement of the knock limit with variant Tz_1+25% compared to the base port Tz_1, but the variant Tz_1+50% falls back to the level of the base port and is the worst variant at 5000 rpm.

In the following the unexpected results are analyzed in more detail using optical measurements and combustion analysis. Figure 6 shows the distribution of knock events for all intake port variants at engine speed 5000 rpm and IMEP=21 bar. The first two variants Tz_1 and Tz_1+25% have most knock events on the exhaust side, whereas the variant Tz_1+50% on the opposite intake side.

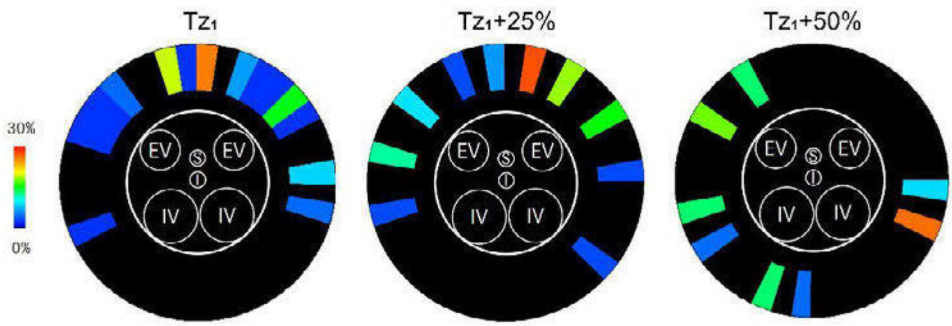

Figure 6: Distribution of knock events at n=5000 rpm, IMEP=21 bar and λ=1

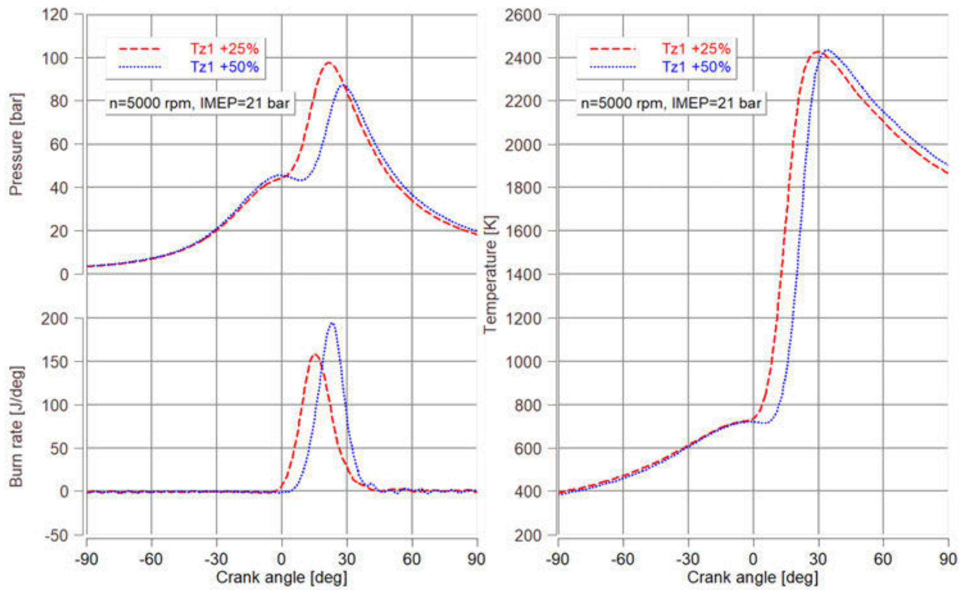

Figure 7: Cylinder pressure, burn rate and gas temperature at n=5000 rpm, IMEP=18 bar and λ=1 for two different tumble levels

From this one could deduce that there is a hot spot on the exhaust side, e. g. at the exhaust valves, for the two variants with lower charge motion level. But this would also have a negative influence on the variant with the highest charge motion. To further investigate this, the flame propagation was visualized for the intake port Tz_1+25% at a slightly reduced load using high-speed endoscopy. The analysis showed a fast flame movement towards intake side of most cycles, which explains the knock events at the exhaust side. Thus the intake port Tz_1+50% has not only the highest tumble level, but also leads to a different flame propagation.

Figure 7 shows cylinder pressure, burn rate and gas temperature for the intake ports Tz_1+25% und Tz_1+50% at 5000 rpm and IMEP=21 bar. Due to a later MFB 50% the peak pressure of variant Tz_1+50% is lower than of variant Tz_1+25%, but the peak temperatures are almost identical due to the higher flame speed of variant Tz_1+50%. Since the two cylinder heads differ only in the inlet channels and knocking is significantly influenced by the gas temperatures, the results shown here could indicate a thermal limit of the combustion chamber. This will be the focus of chapter 3.4.

3.3 Influence of cooled EGR on knock limit, exhaust gas temperature

Due to the high power density of modern gasoline engines, sufficient cooling in the vehicle is definite a challenge. If cooled residual gas is used, it increases the need for additional cooling. Therefore, the maximum residual gas rate is restricted in real vehicle operation. Since this is not a problem on test benches, the investigations were carried out with the maximum possible residual gas rates.

Figure 8 shows the change of MFB 50% with increasing residual gas rate (EGR) applied for different speeds at a load of IMEP=18 bar. For all variants, an improvement of the MFB 50% can be observed with increasing residual gas rate, but its amount depends significantly on the speed. In particular, the behaviour of the variant Tz_1+50% with the worst knock limit (see Figure 5) is remarkable, which shows the biggest improvement in MFB 50% upwards n=4000 rpm compared to the other two variants.

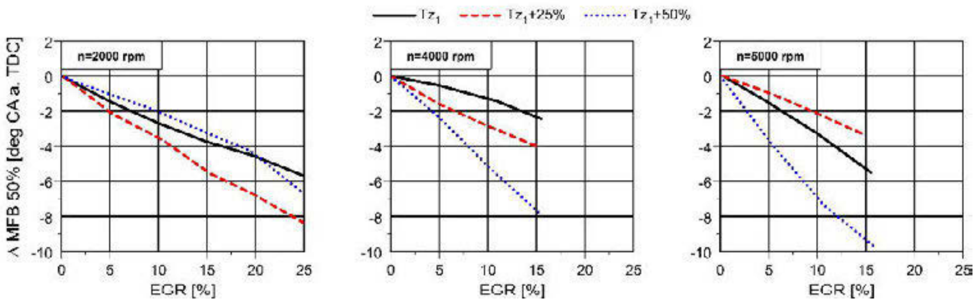

Figure 8: Effect of EGR on the MFB 50% for different speeds, IMEP=18 bar and λ=1

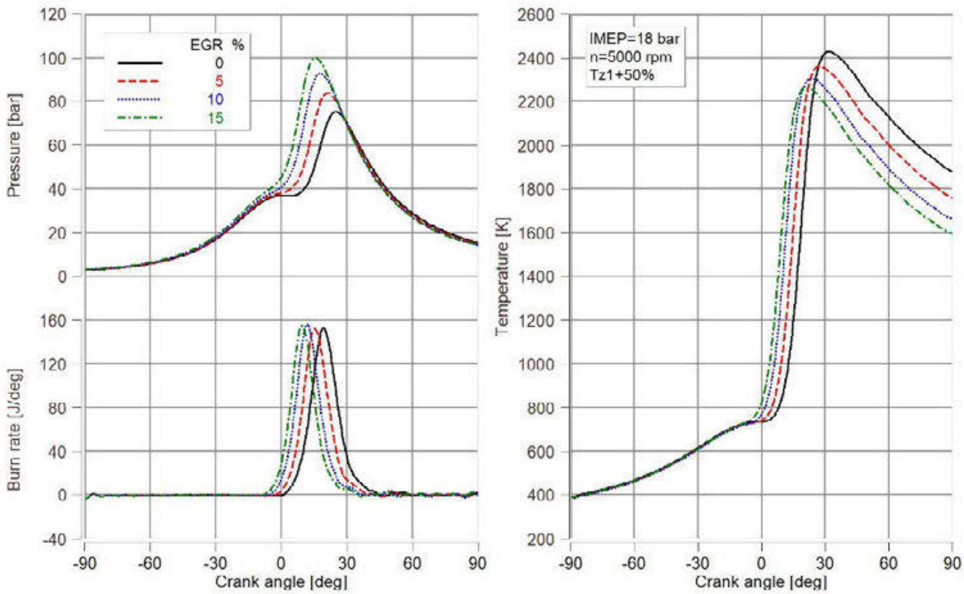

Figure 9: Cylinder pressure, burn rate and gas temperature at n=5000 rpm, IMEP=18 bar and λ=1 for different residual gas contents, tumble level Tz_1+50%

Figure 9 shows for four different residual gas rates the cylinder pressure, heat release rate and mass mean temperature curve at n=5000 rpm and IMEP=18 bar for intake port Tz_1+50%. Although the peak pressure in the combustion chamber increases significantly with rising residual gas mass and earlier MFB 50%, the maximum mass mean temperature decreases and accordingly the heat flow into the combustion chamber wall and its temperature.

This significant improvement in knock limit is also evident in the case of increasing load with and without residual gas, which is illustrated in Figure 10. The dependency of the MFB 50% on the load for the best variant Tz_1+25% and the worst variant Tz_1+50% are compared with and without residual gas for the speeds n=2000, 4000 & 5000 rpm. While at 2000 rpm MFB 50% and therefore the knock limit are almost the same, the situation changes significantly at 4000 rpm. In both cases without residual gas (as shown in Figure 5), the intake port Tz_1+50% has a significantly worse knock limit than the port Tz_1+25%. After residual gas is added, the knock limit of the intake port Tz_1+50% catches up and is comparable to the other intake port.

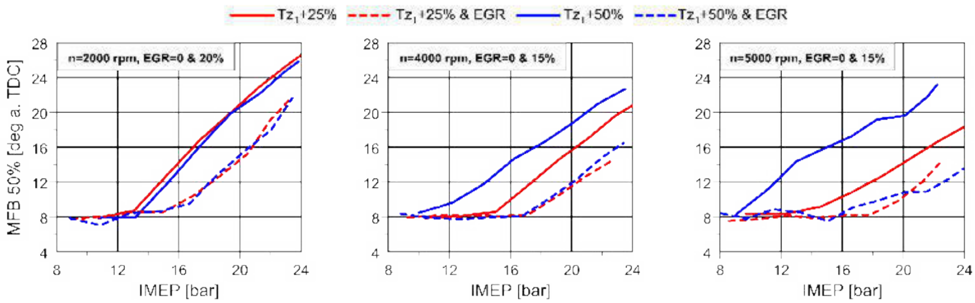

Figure 10: MFB 50% at increasing loads with and without cooled EGR, λ=1

Since the variant Tz_1+50% has the greatest changes of the knock limit, in Figure 11 the pressure and heat release rate as well as the mass mean temperature and the wall heat flow at n=5000 rpm and IMEP=20 bar are displayed. Due to the higher cylinder mass and the significant earlier MFB 50% the peak pressure in the case with 15% residual gas is higher. Surprisingly, because of the earlier knock limit the burning rate is almost as fast as without EGR. The mass mean temperatures is lower due to the increased cylinder mass, despite of the approximately 12°CA better MFB 50%. Although there are considerable differences in the mixture composition and the combustion process, the total heat loss (Qw) to the walls towards the end of the expansion stroke is almost the same.

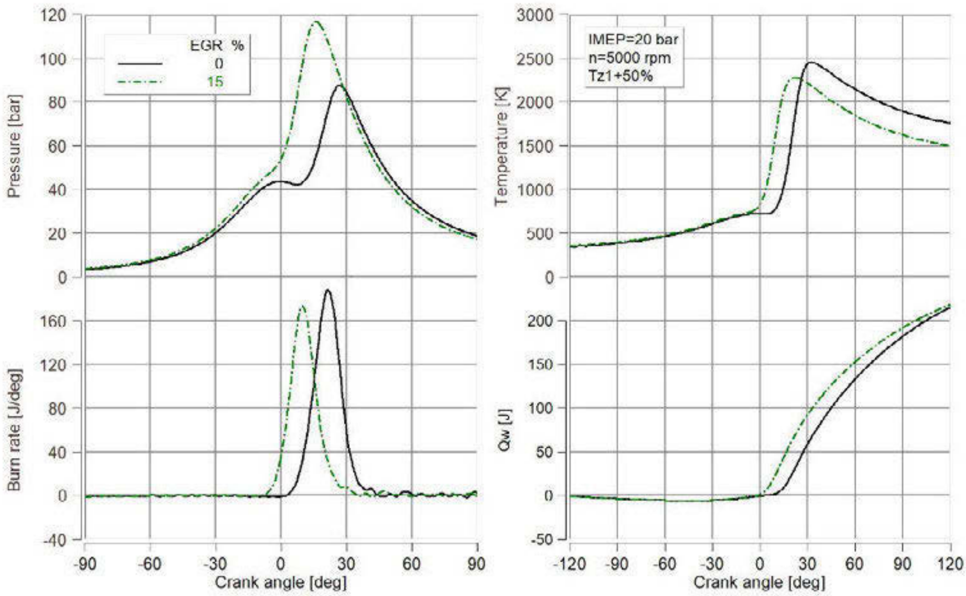

Figure 11: Combustion analysis for n=5000 rpm, IMEP=20 bar, λ=1, EGR=0 & 15%

In most papers on knock criterions [e.g. 3, 10], the temperature in the unburnt zone plays a key role. In addition, high charge motion levels are considered advantageous for good knock limits. However, the results presented her suggest the benefits are not unlimited.

3.4 Existence of a thermal limit

As shown in Figure 7, the higher flame speed of the intake port Tz_1+50% results in a comparable maximum mass mean temperature despite the significantly later knock limit spark advance. For a better understanding concerning the thermal gas conditions in the combustion chamber, the results from a two-zone calculation for the load variation presented in Figure 10 are analysed. Figure 12 shows the maximum temperature in the burnt zone, Figure 13 in the unburnt zone.

As expected, the maximum temperature in the burnt zone can be significantly reduced with the addition of cooled residual gas. However, it is striking that at n=2000 rpm the temperatures in the case with no EGR are almost identical and at higher speeds slightly lower for the maximum tumble stage. On the contrary, the values for all speeds with cooled residual gas lie very close together.

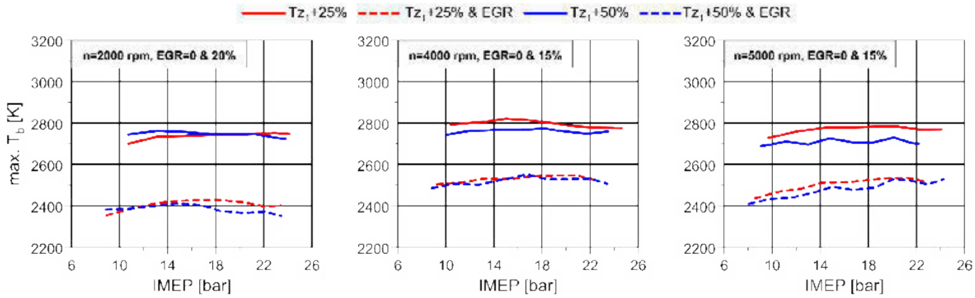

Figure 12: Maximum temperature in burnt zone with and without cooled EGR, λ=1

Considering the temperatures in the unburnt zone, which are used in all common knock models, there are comparable values for all variants with and without added residual gas within the model accuracy. This is particularly remarkable at 5000 rpm for the Tz_1+50% variant, as here at the highest load the MFB 50% is approximately 12°CA different from the other variants.

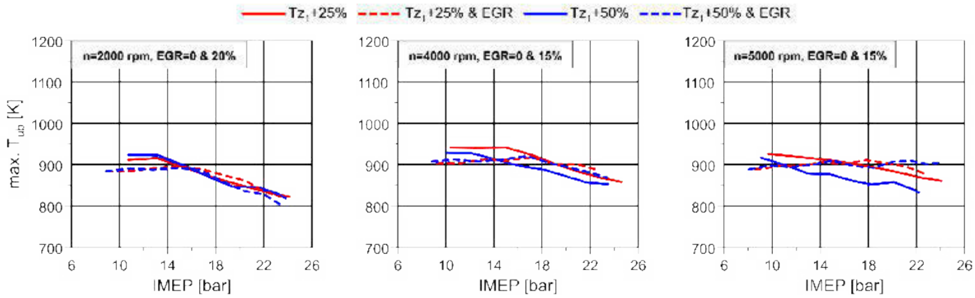

Figure 13: Maximum temperature in unburnt zone with and without cooled EGR, λ=1

Since the examined cylinder heads differ only in the intake ports and not in the combustion chamber geometry or in the water jacket, it is shown that a combustion chamber configuration has a thermal limit. If additional cooling measures are no option, an improvement in the knock limit spark advance can only be achieved with significant lower process temperatures, whereby the thermal limit here assumed is not changed. The use of cooled residual gas is therefore a very effective method to improve the knock limit. A thoroughly designed EGR-system is one of the few technologies which not only contributes to an increase in thermodynamic efficiency in full load operation and enables stoichiometric fuel-air mixture in the entire engine operation map, but also improves the fuel consumption in part load.

4 Summary and outlook

The presented studies of three different charge motion levels have shown that the overall system “combustion chamber” must be carefully designed. The occurrence of engine knock is determined by a variety of parameters that influence each other. In addition to a good mixture formation, which can be achieved by an adequate charge motion level, the thermal condition during the combustion process is crucial. Lower charge motion causes longer combustion duration and lower heat losses to the combustion chamber walls. Higher charge motion with faster combustion speed at the same MFB 50% lead to higher thermal losses to the combustion chamber walls compared to lower charge motions. If the MFB 50% is earlier, the thermal losses increase even further. By adding cooled residual gas, the mean gas temperature during combustion can be significantly reduced. The results presented here show that the maximum gas temperatures in the unburned zone at knock limit spark advance for all tested charge motion levels are approximately at the same height, regardless of the charge motion and amount of residual gas. It can be concluded that there is an individual thermal limit for a given combustion chamber configuration. To shift this limit towards a better knock limit, the process temperature during combustion must be lowered. Since the maximum possible residual gas rate is restricted due to the cooling capacity in a vehicle, additional measures must be taken. This could be the use of water injection, improved charge air cooling or combustion chamber cooling. Even if it was not explicitly shown in this article, the use of cooled residual gas represents a very effective, if not even the most effective measure to improve the knock limit compared to the other mentioned technologies.

5 References

Heller, A.: Motorenwagen und Fahrzeugmaschinen für flüssigen Brennstoff, Erster Band: Motoren und Zubehör, Springer Verlag 1925

https://ec.europa.eu/germany/news/20181218-co2-grenzwerte-autos_de

Worret, R.: FVV-Vorhaben Nr. 700 “Klopfkriterium”, IfKM Universität Karlsruhe, 2001

Rothe, M.: FVV-Vorhaben Nr. 816 “Extremklopfer”, IfKM Universität Karlsruhe, 2005

Curry, S.: A-Three-Dimensional Study of Flame Propagation in a Spark Ignition Engine, SAE Trans. 71, 1963

Sokolik, A.: Self-Ignition, Flame and Detonation in Gases, Israel Program for Scientific Translation Ltd., Jerusalem, 1963

Ganser, J.: Untersuchungen zum Einfluss der Brennraumströmung auf die klopfende Verbrennung, Dissertation RWTH Aachen, 1994

Stiebels, B.: Flammenausbreitung bei klopfender Verbrennung, Dissertation RWTH Aachen, 1997

Warnatz, J. et al.: Verbrennung, Springer Verlag 2. Auflage, 1996

Franzke, D. E.: Beitrag zur Ermittlung eines Klopfkriteriums der ottomotorischen Verbrennung und Vorausberechnung der Klopfgrenze, Lehrstuhl für Verbrennungskraftmaschinen und Kraftfahrzeuge, TU München, 1981

Binder, S.: Implementierung und Verifikation verschiedener Klopfkriterien auf Basis vorhandener Indizierdaten, Diplomarbeit, IVK Universität Stuttgart, 2003

Marchi, A. et al.: Neuer Ansatz zur Berechnung des Klopfens bei Ottomotoren, Ricardo Deutschland GmbH, MTZ 03/2015

Mally, M. et al.: Klopfen bei Volllast-Abgasrückführung, RWTH Aachen, MTZ 02/2018

Altenschmidt, F. et al.: Water-Port-Injection at SI-Engines: Effects and Challenges, 17th Conference "The Working Process of the Internal Combustion Engine", Graz, 2019

Basic investigations on the cause of initial pre-ignition in a constant volume combustion cell

Jan Reimer, Institut für Kolbenmaschinen, Karlsruher Institut für Technologie(KIT), Mitra Zabihigivi (KIT), Ina Volz (Mercedes Benz AG), Jürgen Pfeil (KIT), Frank Altenschmidt (Mercedes Benz AG), Thomas Koch (KIT)

Abstract: This paper investigates the effect of lubricant oil and the additives on low-speed pre-ignition (LSPI). The abnormal combustion phenomenon of LSPI is still an important issue in the development of modern internal combustion engines. Despite numerous studies to understand the potential sources for low-speed pre-ignition, this phenomenon has not yet been fundamentally understood.

Therefore, this paper provides a test procedure to investigate the necessary thermodynamic conditions leading to LSPI. A major goal of this study was to reduce the complexity in a gasoline engine arising from the interaction of diverse processes typically taking place in an engine. Therefore, the experiments were conducted by using a newly developed constant volume combustion chamber as a centerpiece of a completely new testbed. Additionally, a device to generate a defined small amount of tempered oil droplets in the size range of microns into the combustion cell was designed and used for this study.

This experimental approach focuses on lubricant oils as a potential trigger for pre-ignition. Therefore, three oils with different contents of Calcium and Magnesium were used to study ignition delay times and self-ignition temperatures of oil under various engine-like pressure conditions. Furthermore, to separate possible evaporation effects of the liquid oil droplets, the different oil samples were also admixed to the base fuel. The test results showed that in the Calcium range of 0.025% to 0.075%, the Calcium detergents impact LSPI activity as promoters. In addition, from the length of the ignition delay time for oil droplets in the hot air, it could be concluded that an oil droplet in the engine combustion chamber cannot initiate a pre-ignition in the same working cycle. However, an oil droplet, deposited on a hot component in the engine may lead to a pre-ignition in a further working cycle.

1 Introduction

In general, modern turbocharged gasoline engines with direct injection are increasingly showing anomalies during the combustion process. These anomalies may be caused by Low-speed pre-ignition (LSPI), which is characterized by an ignition before the actual spark ignition timing. Low-speed pre-ignition (LSPI) is an undesirable combustion phenomenon and it potentially leads to extreme knocking events, which can cause serious engine damage. Consequently, exploring possible sources and the mechanism of this type of abnormal combustion in internal combustion engines plays a significant role in the development of modern gasoline engines.

Since pre-ignition seems to be a serious challenge for improving the performance of gasoline engines, numerous researches have been conducted to clarify the mechanism of this phenomenon and to propose solutions to eliminate LSPI. Over the last several years exploring a proper solution for LSPI has been a serious technical challenge for automotive companies. Due to this fact, there have been major efforts to establish a systematic methodology for exploring and evaluating potential sources of LSPI events. Because of the risk of engine damage after a pre-ignition event, investigations to reduce LSPI frequency have attracted considerable attention. However, despite several high-quality studies looking to understand the mechanism behind the LSPI and searching for the main origin of pre-ignition events, this phenomenon needs still more researches in order to explain the mechanism and the probable causes.

While some early publications focused on engine design and operation to reduce LSPI frequency, recent publications have brought other probable causes of pre-ignition into focus because LSPI should be explained by a combination of engine-related, oil-related, and fuel-related mechanisms interacting in an engine [1–3]. Jatana et al. [2] , for instance, have performed engine testing with three single component fuels in order to understand the effect of fuel properties on LSPI. The aim of this study was to investigate the dependence of fuel distillation on Low-speed pre-ignition (LSPI).

On the other hand, in several works lubricant oil has been identified as the main origin of pre-ignition [4–8]. These studies have attributed LSPI events in gasoline engines to the auto-ignition of lubricant oil droplets in the combustion chamber. Considering the results of these publications, one of the most probable explanation for the occurrence of pre-ignition in a gasoline engine is that the oil droplets, released from the cylinder liner, can become a source of LSPI [5, 6, 9]. Furthermore, in several studies, composition of lubricant oil and the oil additives are believed to be the major contributing factors in the occurrence of LSPI events in the engine [7, 10].

Morikawa et al. [11] indicated that the oil properties will affect the tendency of LSPI. Three different oil properties such as cetan number, distillation characteristics, and Calcium additive were examined in this work as critical points in pre-ignition. Dahnz et al. [6] carried out experimental investigations using a test engine accompanied by numerical simulations to assess possible causes for pre-ignition. The results have identified that oil dilution greatly affects the amount of released oil droplets and hence the pre-ignition frequency. Okada et al. [12] have applied visualization approach to perform optical investigations and to clarify the mechanism of pre-ignition in the engine. Ritchie et al. [13] presented a statistical analysis by assessing data collected from six different engines. Furthermore, tests were conducted on various lubricant components to determine their effects on LSPI. The effects of Calcium and Magnesium detergents on LSPI were investigated. They found out that in contrast to the effect of Calcium, an increase in Magnesium concentrations has no effect on the occurrence of LSPI. Hence, LSPI results for mixtures of Calcium and Magnesium are directly proportional to the amount of Calcium. Takeuchi [14] has performed several engine tests with a prototype turbocharged DI-SI engine to investigate the influence of the oil additives on pre-ignition. This study confirmed that the type of oil and additives have an intensive impact on LSPI. Results indicated that Calcium detergent has a contributory effect on a pre-ignition event and higher Calcium content can benefit the pre-ignition rate. In addition, several other studies indicated that increasing levels of Calcium lead to an increase in the LSPI rate [4, 15]. However, a close examination of the impact of Calcium additives is lacking and this point requires further in-depth investigations to be sufficiently clarified. Exploring and understanding the dependence of pre-ignition on Calcium detergents is a major goal of this paper. Therefore, a fundamental evaluation of the impact of Calcium on LSPI activity has been undertaken in this work.

The vast majority of publications describing lubricant impacts on LSPI only consider engine tests to assess the impact of different factors on this phenomenon [5–7, 11, 16]. Most of these investigations have been carried out inside an engine, where many other different factors can influence the mechanism of pre-ignition. For this reason, a detailed investigation of LSPI by engine tests is not possible due to the complexity coming from several processes taking place simultaneously in the engine. Therefore, the results of experiments depend strongly on engine variables. Once these several influencing variables are reduced, it is possible to get a more accurate understanding of the process. Therefore, the investigations are to be carried out with some simplifications through delimiting the influencing variables of the engine. Consequently, in this study the investigations are not conducted in a combustion engine, but in a constant volume combustion cell. It was decided to utilize a combustion cell, which is newly designed and developed at KIT. The detailed characteristics of combustion chamber will be explained in the following sections of this paper.

The experiments and results presented in this paper are to be understood as a following project to the previous paper presented in 5th International Conference of Knocking in Gasoline Engines [17] and also as a preliminary step to the upcoming experiments in the future. While the general features of the testbed and the measurement system were described in the previous publication comprehensively, this paper focuses on the experimental methodology and results. In addition, some newly added components and features are also presented in this paper.

2 Testbed development and setup

This paper is a follow-up study and the second part of a paper series presenting the further developments of the testbed, methodology, and results of the project. In the first part of this series of papers [17] the general requirements, challenges, and features of the testbed were discussed. Additionally, a detailed description of the design and dimensioning of the combustion chamber, as well as the measurement setup, were presented. The current paper provides first a general overview of the combustion chamber system. It will be explained, how the combustion chamber has been further developed and which other new components have been added to the system. Then in the following sections, it focuses on test procedure and results.

Following the aim of studying the conditions leading to pre-ignition, a combustion chamber was developed to be utilized for the experiments. In order to reduce the system complexity, the engine was replaced with a combustion chamber, since the complexity of the engine makes detailed analysis difficult. This provides a great possibility to perform fundamental experiments to clarify the necessary conditions for the occurrence of pre-ignition accurately. Consequently, a constant volume combustion chamber was manufactured for optical investigations of pre-ignition at high pressures and high temperatures.

Different parts of the system are described in Fig. 1. The system consists of the pre-conditioning chamber, the combustion chamber with optical accesses, fuel injector, and a newly developed oil dosing device. In addition, the fuel and air management systems serve to supply air and fuel during experiments. The fuel management system can provide an injection pressure of 200 bar using an accumulator serving as a pressure reservoir. Furthermore, several valves and sensors are utilized to implement the experiments. The positions of different pressure sensors (blue) and temperature sensors (green) are described in Fig. 1. High-pressure cut-off valves are utilized to control the flow through the system and also to prevent the damaging of other components.

At the beginning of each experiment, air flows from the air management system through valve V1 into the pre-conditioning chamber. After some procedures in the pre-conditioning chamber, which are explained in detail in section 3, air or the gaseous mixture flows through valve V2.1 into the combustion chamber. Then, combustion takes place in the combustion chamber. The combustion chamber has been developed for a maximum gas pressure of 350 bar and a maximum gas temperature of 500 °C. The combination of high pressures and high temperatures has been the main challenge for the design, dimensioning, and safety of the combustion chamber.

In order to provide the same preliminary conditions for all experiments, it must be possible to release the burned gas from the chamber completely and to flush it. Therefore, after the combustion is done completely, valve V2.2 opens and makes it possible for the air in the pre-conditioning chamber to release into the environment. Valve V3 has a similar function and serves the burned gases to exit from the combustion chamber. Applying a switching valve makes it possible to flush the system by air or nitrogen.

In addition to the cut-off valves, there is also a pressure control valve, which serves as an element for precise adjustment of the pressure in the system. Furthermore, since pressure peaks of higher than 350 bar can occur during the experiments, a pressure relief valve is utilized as a security measure in the system. The relief valve opens when the pressure in the combustion chamber exceeds 350 bar. Although the relief valve is used to prevent pressure peaks over 350 bar in the combustion chamber, the dimensioning of the chamber is based on much higher pressures in the order of magnitude of 1000 bar.

Controlling and adjusting of the different actuators and valves as well as the recording of the pressure and temperature values are performed by a modular CompactRIO System from National Instruments. The pressures inside the two chambers are recorded by an indicating system with a frequency of 100 kHz.

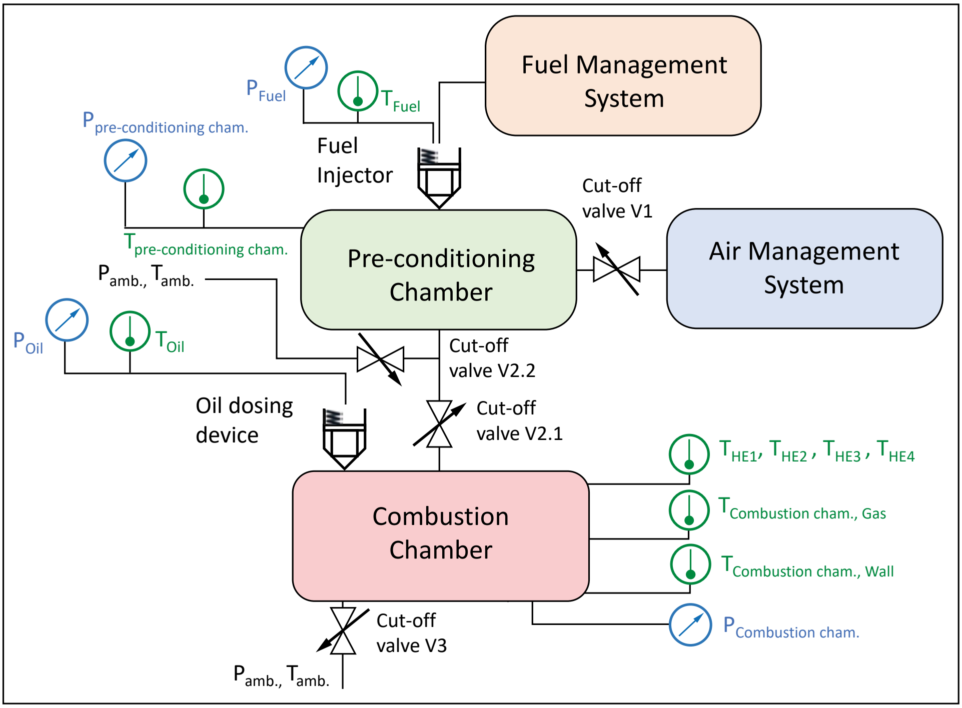

Fig.1: Overview of the developed Testbed [17]

The combustion chamber with its four accesses is shown in Fig. 2. The design and positioning of the different accesses in the chamber required precise calculations in order to satisfy the demanding requirements. The combustion chamber was conditioned by regulated wall heating, depending on the temperature requirement of each experiment. For the reason of reproducibility, a homogeneous temperature field in the gas phase inside the combustion chamber was indispensable.