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Investigation of compression temperature in highly charged spark-ignition engines

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Abstract: One of the critical parameters for the occurrence of spark-knock and pre-ignition of highly charged spark-ignition (SI) engines is the compression temperature. The investigation of the compression temperature and a better understanding of the combustion behaviour inside the combustion chamber are of great importance for avoiding these phenomena and also for further downsizing of engines. Pure rotational coherent anti-Stokes Raman spectroscopy (RCARS) is an innovative tool providing information on the compression temperature by using only two small line-of-sight optical accesses on the cylinder wall. In this work, RCARS measurements were performed in motored and fired operation, and the results obtained from RCARS spectra have been used in combination with isentropic calculations to improve the accuracy of the compression temperature determination. Studies on the compression temperature by the variation of the intake temperature, the engine load, and the fuel were conducted and the influence of evaporation enthalpy on the compression temperature is discussed.

Keywords: highly charged SI engine, CARS, compression temperature

1 INTRODUCTION

Downsizing of spark-ignition engines in combination with direct fuel injection is an effective method to reduce fuel consumption and the CO₂ emissions of SI engines. As a result of reduced displacement of the engine, the operating points shift to higher loads and the engine efficiency is enhanced. Thus the mean effective pressure has to be increased significantly to achieve more benefits from the

downsizing strategy. However, under full load the harsh thermal conditions can lead to abnormal combustion such as spark-knock or pre-ignition, especially at low engine speed. This kind of abnormal combustion, which is one limiting factor for further downsizing of engines, is highly dependent on the key parameters during the compression stroke and combustion inside the engine chamber, such as temperature, pressure, and chemical composition. Among these parameters the compression temperature is very important since it governs the chemical reactions leading to pre-ignition. Thus it is necessary to study this parameter in much detail. However, the complex interaction between charge cooling by direct injection, the amount and temperature level of the recirculated exhaust gas, and heat

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losses at the cylinder wall make it difficult to perform accurate temperature calculations inside the combustion chamber.

For this purpose, non-invasive laser diagnostics form a useful measurement tool. There are already different examples in the literature demonstrating the potential of laser techniques for temperature measurements at engine-relevant conditions. Grünefeld *et al.* investigated the temperature behaviour in a spray flame and an internal combustion (IC) engine by spatially resolved spontaneous Raman scattering, but the impact of compression temperature on pre-ignition was not discussed in detail and the engine was equipped with large glass windows [1]. Laser-induced fluorescence (LIF) is another possible measurement technique used for temperature imaging [2, 3]. In these applications, model fuels doped with 3-pentanone and triethylamine were applied and the engines were equipped with large glass windows for signal excitation and signal detection. Furthermore, coherent anti-Stokes Raman spectroscopy (CARS) can provide information about the in-cylinder charge temperature. CARS is a coherent, point-wise probing technique for the simultaneous temperature and concentration determination in combustion environments [4–6]. Since a CARS signal can be collected by collection optics with a small numerical aperture, only a small optical access is needed and no large modifications on the engine are required, which offers the possibility to take measurements in nearly production-type engines. Additionally, for the operation of the engine no model fuels are required. Lucht *et al.* performed temperature measurements in an IC engine applying the vibrational CARS technique (VCARS) using *n*-butane as fuel. Their focus was mainly to improve the accuracy of the fitted temperature, and they analysed a number of factors which could impact this accuracy [7]. Several studies have been executed demonstrating the applicability of pure rotational CARS (RCARS) as a useful engine combustion diagnostic technique [8–11]. As the population of the molecular rotational levels is probed, RCARS is comparatively sensitive to temperature changes and valid even for higher pressures because of the large separation of pure-rotational lines and the negligible effect of coherent line mixing, which becomes increasingly important for VCARS at increased pressure. Thus the evaluation of RCARS spectra is much easier than for VCARS. Finally, compared to other temperature measurement techniques, RCARS studies have been proven to be more accurate than VCARS at relatively low temperatures (< 1200 K). The largest deviation between the thermocouple measurements and evaluated RCARS spectra was around 55 K for spectra recorded in an oven

in the temperature range of 300–1600 K, giving an indication of the overall accuracy of the technique even for high temperatures [12, 13].

Here, only a brief introduction into the RCARS technique will be presented; more details can be found elsewhere, for example in Eckbreth [14]. In RCARS, S-branch rotational Raman transitions are excited resonantly by the difference between the frequency of the pump beam ω_{pump} and the Stokes beam ω_{Stokes} , as shown in Fig. 1. A narrowband probe laser ω_{probe} is scattered from the excited Raman coherence, and the signal with the RCARS frequency ω_{CARS} is emitted as a coherent beam which is Raman shifted to the anti-Stokes side of the probe. Thus interferences from fluorescence processes are generally not a problem because the CARS signal is blue-shifted. Temperature and concentration information are encoded in the relative intensities of different rotational lines in the RCARS spectra. In the applied dual-broadband (DB) approach a single broadband dye laser serves as both pump beam and Stokes beam [15, 16]. Not only is the complete RCARS spectrum generated in a single shot, but the effect of dye laser mode fluctuations is minimized and higher accuracy and precision can be achieved by this DB-RCARS technique.

To obtain more accurate information out of the RCARS measurements, a calculated temperature curve is fitted to the RCARS results. With this curve, an exact compression start and end temperature can be defined and a better accuracy is achieved. In the first step the compression temperature is calculated based on the isentropic theory [17]

$$T = T_0 \cdot \left(\frac{V_0}{V_1} \right)^{\kappa-1} \quad (1)$$

Fig. 1 Energy levels of RCARS transitions

where V_0 and V_1 are the combustion chamber volume at the beginning and the end of the compression. Both variables are determined by the engine geometry and the piston position. The isentropic coefficient κ is ascertained by the specific heat capacity of the in-cylinder components [18–20], which are air, fuel, and residual gas. The exact mass amount of each component is obtained with a thermodynamic gas exchange analysis. A more detailed description about this analysis can be found elsewhere [17, 21, 22]. The calculation was carried out with the computer program ‘GT Power’ [23] by using intake, cylinder, and exhaust pressure, exhaust temperature, geometric engine data, and information about the fuel as already known parameters. The remaining variable in equation (1) is the compression start temperature T_0 at bottom dead centre (BDC). For this purpose the RCARS measurements could not be used directly because the standard deviation due to cycle-to-cycle variations, beam steering, and degradation of phase matching induced by temperature gradients and engine vibrations was about 25 K. During engine operation without firing, a CARS standard deviation of 17 K was achieved. In contrast to this, at laboratory conditions a standard deviation of 10–12 K is typical for this temperature range [12, 24]. Therefore the curve of the calculated isentropic temperature is fitted to several RCARS temperature measurements taken at different crank angles during the compression stroke. The start temperature was varied until the temperature difference between CARS measurements and the isentropic calculation reaches a minimum. A better accuracy of temperature determination can be achieved by combining a series of RCARS measurements with the isentropic calculation than by the RCARS measurement results alone, because more experimental information about the temperature profile was available.

In order to investigate the performance of highly charged spark-ignition combustion under realistic engine conditions, the dependence of the compression temperature on different intake temperatures, engine loads, and fuels are studied in this work, and the RCARS results are compared to isentropic calculations.

2 HIGHLY CHARGED SI ENGINE

The engine used in this study was the Volkswagen 1.4l-TSI spark-ignition four-cylinder engine. It was designed with single continuously variable cam phasers for the intake camshaft and twin charger

technology [25, 26]. It provides intake air pressure up to 2.5 bar by an in-line supercharger and turbo charger. A stoichiometrical direct fuel injection with fuel pressure up to 150 bar was used for this engine. More details of the engine are shown in Table 1.

Each cylinder of the engine was equipped with one pressure transducer. Additional pressure and temperature transducers were placed close to the intake and the exhaust system of the fourth cylinder. Cylinder four was chosen for the RCARS temperature measurement because of its easiest accessibility. A pair of line-of-sight optical accesses was implemented in the top part of the engine block. The optical accesses in the engine block are shown in Fig. 2. Two flat fused silica windows, 8 mm in diameter, were respectively bonded into the optical access and could easily be detached for cleaning or renewing. Two aluminium O-rings were used on both sides of each silica window to reduce the vibrational damage from the running engine. Thus, the diameter of the effective area on the optical window for the laser beam passing through was only 6 mm. In addition, one side of each fused silica window was coated with an anti-reflective layer for the 532 nm laser beam, and the uncoated side faced the combustion chamber when the window was mounted into the optical access.

The compression ring of the piston was not allowed to reach the level of optical access, and therefore the conrod length had to be reduced by 9.6 mm. In order to achieve the original position of the piston, the fire land was elongated for 9.6 mm. All the modifications mentioned lead to a compression ratio of 9.3 which was comparable with the value of 9.5 for the original engine. Due to the elongated fire land the emissions of hydrocarbons and nitrogen oxides were about 10 per cent higher than for the engine without modification. No more modifications were made on the engine and all of the other measurement parameters of the engine agreed with an unmodified engine. That means the situation of the real running engine could be simulated quite well in the present studies. During operation, the CARS signal passed through the engine until the piston reached -35°CA after top dead centre (ATDC). While the crank angle is between -35 to 35°CA ATDC the optical accesses are blocked because during this period the piston is at the top of the combustion chamber.

3 RCARS SETUP AND MEASUREMENTS

The laser system, illustrated in Fig. 3, consists of a Nd:YAG laser with wavelength of 532 nm and

Table 1 Specifications of the investigated engine

Valve number per cylinder	4
Displacement	1390 cm ³
Bore	76.5 mm
Stroke	75.6 mm
Compression ratio	9.3
Conrod length	130.4 mm
Fuel injection system	Direct fuel injection
Engine controller	Bosch MED 9.5.5
Fuel quality	Gasoline RON 95

Fig. 2 CAD model of the optical accesses in the engine block

a broadband dye laser. The laser dye was 4-dicyano-methylene-2-methyl-6-(*p*-dimethylaminos-tyryl)-4H-pyran (DCM) which was dissolved in ethanol, and the wavelength of this dye laser was centred at 630 nm. The initial Nd:YAG laser pulse

was split into two beams: one was used for the RCARS process and the other one for pumping the dye laser. The dye laser beam was then also split into two beams and focused together with the Nd:YAG laser by a lens with 400 mm focal length. Laser polarizations were defined by Glan–Taylor prisms to be vertical for all the three beams. A planar BOXCARS phase-matching geometry was chosen in order to apply RCARS in confined optical access of the engine. A beam-view system was used to monitor and optimize the beam crossing in real time and to analyse the performance of each beam near the focal-plane [27]. The RCARS signal emerges from the overlapping region of the three laser beams in a laser-like manner. The probe volume is actually an elongated ellipsoid and is normally approximated by a cylinder of length L and diameter d . In the present experiment this cylinder size was approximately 2 mm in length and around 100 μm in diameter [27]. Maximum laser pulse energies of 4 mJ for each of the three beams were used to avoid window damage and laser-induced breakdown.

The RCARS signal was separated from the other three beams by a series of dichroic mirrors and beam dumps and then focused into a spectrometer by a lens with 100 mm focal length. The spectrometer was equipped with one grating of 1200 grooves/mm whose focal length was 0.55 m and the theoretical resolution was 0.03 nm. A knife edge was placed at the exit plane of the spectrometer to filter elastically scattered stray light [28]. The spectrally dispersed signal was finally imaged onto a CCD camera with a chip size of 1600×1200 pixels and $7.4 \times 7.4 \mu\text{m}$ per pixel size. To minimize camera noise, a binning of 2 was applied in the wavelength direction and the data read out was from a region

Fig. 3 Schematic of RCARS setup (BS: beam splitter; T: telescope; GP: Glan–Taylor prisms; DC: dichroic coupler; BD: beam dump)

of interest with only 10–20 pixel rows in width, where the RCARS signal appeared. No camera binning in the second direction was used in order to retain full dynamic range. The complete setup was built up on two conventional optical breadboards in the engine test facility; thus a vibrational isolation was not necessary. Because of the harsh conditions surrounding the engine when it is operating, a remote controller was used to run the experiments in another room beside the engine test bed for safety considerations.

The engine was operated at a constant engine speed of 1800 r/min. The laser pulse frequency was 30 Hz. An electronic trigger box was used to transform the crank angle degree (°CA) signal from the crankshaft position transmitter into a signal with 30 Hz repetition which was used to externally trigger the laser and camera. As a result the consecutive measurements were performed at the desired degree of crank angle in the compression stroke and 360° shifted in the gas changing cycle. Therefore for each measurement point in the compression stroke from BDC (−180 °CA) to TDC (0 °CA), 200 spectra of single-shot laser pulses were recorded.

Before the start of the measurements, the cylinder was filled with argon through the spark plug hole with an adapter tube to record a non-resonant excitation profile. The experimental spectra were corrected for the bandwidth profile of the broadband dye laser by dividing the measured spectra of pure argon. As argon has no rotational transitions, the spectrum is entirely composed of the non-resonant signal [29]. The temperature and concentration evaluation from RCARS spectra were based on a computer program that compares the experimental spectra with a spectrum from a pre-calculated library by the use of a least-squares contour-fitting procedure [30]. This pre-calculated library was determined by varying temperature and concentration and non-resonant susceptibility. The mean pressure measured at the corresponding degree of crank angle by the mounted pressure transducer was taken as an input value for the RCARS spectra fitting.

4 RESULTS AND DISCUSSION

Measurements were conducted for variations of the parameters, intake temperature, engine load, and fuel. For all measurement points the spark timing was set to the knock limit. The start of injection (SOI) was always at −300 °CA ATDC, and the air number lambda (λ), which is the reciprocal of the

equivalence ratio ϕ , was also kept constant with $\lambda = \phi = 1$. Since spark-knock occurs at the edge of the combustion chamber and pre-ignition appears stochastically distributed inside the combustion chamber, there are two general possibilities for optimization by the improvement of the local and of the global conditions in the combustion chamber, which are temperature, pressure, and chemical composition. Both approaches could reduce the probability of occurrence of these two phenomena. In the present study, the CARS technique was used to study and improve the global temperature conditions, thus the probe volume was located 9 mm below the cylinder head gasket and in a central position of the cylinder.

4.1 Variation of intake temperature

The variation of intake temperature is mainly used to characterize the standard deviation and the accuracy of the measurement of the CARS technique during the fired operation. It could also demonstrate that with isentropic calculation the accuracy could be improved. In this investigation the engine was operated in a stable stationary operation at a constant engine load of 60 per cent with gasoline RON 95. The compression temperatures with two different intake temperatures, 285 and 305 K, recorded by the temperature transducer at the intake manifold, were studied.

In Fig. 4, the separated open symbols are the mean values obtained from 200 single-shot RCARS spectra with standard deviation represented by the error bars, while the isentropic calculations are fitted to the RCARS results being presented by curves. Blue colour displays the lower intake temperature and red denotes the higher one. The overall standard deviation of RCARS fitting results in these measurements was about 25 K because of cycle-to-cycle variations, beam steering, and degradation of phase matching induced by temperature gradients and engine vibrations. Additionally, when standard gasoline fuel was used in the engine, the contamination on the optical access windows led to increased stray light levels that allowed measurements in fired motor conditions only with a low signal/noise-ratio (SNR). Due to this standard deviation of CARS, it was difficult to distinguish the difference between the compression temperatures in these two different intake temperature cases. However, a clearer view of the compression temperature difference could be obtained and a much more accurate result was achieved by fitting the RCARS results with isentropic calculations as shown

Fig. 4 Comparison of temperatures being calculated by isentropic equations and from the RCARS spectra at different crank angles

by the curve which provides the standard deviation below 10 K.

It can be seen from Fig. 4 that for both intake temperatures the air/fuel mixture was heated up by around 35 K from the intake manifold to BDC (180 °CA), where the calculated temperature with isentropic fitting for 305 K intake temperature was 342 and 320 K for an intake temperature of 285 K. A start temperature difference of 22 K between two different intake temperatures was found, which was in very good agreement with the value of 20 K read from the temperature transducer. Nonetheless, the start temperature difference at BDC was expected to be slightly smaller than 20 K because for lower intake temperature the heat transfer from the surface of the engine cylinder to the cooler air charge is larger. However, this could not be observed, probably because the heat transfer was very small. The reason could also be that it was within the accuracy limitation of the RCARS technique in combination with the isentropic calculations. A remarkable deviation of 38 K between the two curves was observed at TDC, where the compression temperature was 702 and 664 K for 305 and 285 K intake temperature, respectively. This noted temperature difference at TDC is almost double the value at the initial BDC. According to equation (1), for lower start temperature the temperature gradient was also smaller during the compression. This leads to an increasing temperature difference between the two cases at the end of the compression stroke.

4.2 Variation of engine load

Self-ignition phenomena such as pre-ignition [31, 32] which occurs at low speed and high load is heavily dependent on temperature, pressure, chemical composition, and spark time. The compression temperature is the dominating one among all these parameters and in the present study it was investigated at a variation of the load for 50, 70, 80, and 90 per cent. During the valve overlapping, an air flow was formed because of the pressure gradient between the intake port and the exhaust port, which helps to scavenge the gases of combustion in the cylinder (scavenging effect). This process keeps the amount of residual gas for 70, 80, and 90 per cent load below 1 per cent, and only for 50 per cent load was it a little bit larger than 1 per cent. The conditions of the engine and the results for the compression start and end temperature are summarized in Table 2. The SOI was kept constant at -300 °CA ATDC, while the end of injection (EOI) was drifting with increasing load due to a larger required amount of fuel.

The mean values of the single-pulse RCARS temperature results (symbols with error bar) and the isentropic calculations (lines) in the compression stroke under different engine load are shown in Fig. 5. The calculated isentropic curves fit well to the measured RCARS temperatures except for some points approaching TDC. All start temperatures at BDC were located in the range from 328 to 333 K. Comparing with the value of 290 K recorded by the temperature transducer, the mixture was heated up

Table 2 Conditions and results for the variation of load at a constant speed of 1800 r/min

Load (%)	P_{Intake} (bar)	T_{Intake} (K)	Spark timing ($^{\circ}\text{CA ATDC}$)	$\lambda(-)$	T_{Exhaust} (K)	T_{BDC} (K)	T_{TDC} (K)	SOI ($^{\circ}\text{CA ATDC}$)	EOI ($^{\circ}\text{CA ATDC}$)
50	1.12	290	-8	1.0	788	332	679	-300	-273
70	1.63	290	-2	1.0	914	328	665	-300	-256
80	1.81	290	1	1.0	943	333	670	-300	-249
90	2.01	289	5	1.0	983	329	661	-300	-241

Fig. 5 Temperatures calculated by RCARS and by isotropic theory with different engine load

by around 40 K from intake port to BDC for all different loads. It was assumed that the temperature at the end of the compression stroke would increase with higher load. However, the obtained results showed an unexpected result, indicating that the compression end temperature was not affected by an increasing load. Some facts contribute to this phenomenon. The first one is that the parameters except engine load such as air/fuel-ratio and residual gas amount inside the cylinder were kept almost the same. Second, the surface temperature of the combustion chamber increased with higher load, which leads to a larger heat transfer from the surface to the air/fuel mixture. But at the same time, in order to maintain a higher load, more air/fuel mixture will be fed into the cylinder. The higher heat transfer will be diluted by the larger amount of air/fuel mixture. Furthermore, the theoretical isotropic temperature calculation has three influencing factors: start temperature, compression ratio, and isotropic coefficient. The start temperature (T_{Intake}) and the compression ratio, are constant for this variation of load. The isotropic coefficient is also not dependent on the load

(cylinder pressure) if the temperature is below 1500 K [33] and in the present study the compression temperature is definitely below this value. Thus the compression temperatures are comparable for all four different loads.

The RCARS measurements and the isotropic calculation of the temperature during the compression stroke combined with the theoretical considerations confirmed that the load has obviously no influence on the compression temperature. This means that pre-ignition will occur if a certain temperature threshold is achieved in the combustion chamber; however, this temperature threshold is not increasing with higher loads. Thus in the case of pre-ignition, besides temperature the influencing factors mentioned above such as pressure, chemical composition, and the spark timing could play an even more important role than load on the generation of pre-ignition.

4.3 Variation of fuel

Another important influencing factor on the compression temperature is the vaporization enthalpy,

Table 3 Fuel characteristics

Fuel (-)	Evaporation enthalpy h_{EV} (J/kg)	L_{ST} (-)
Gasoline RON 95	420 000	14.6
Isooctane 95%	308 000	15.13
E85	831 400	9.86

which is required to overcome the interactions between molecules in a liquid state during evaporation [34]. An energy balance based on a different evaporation enthalpy is used to estimate the reduction of temperature ΔT_{EV} during evaporation in the compression process. The expression of the temperature cooling is

$$\Delta T_{EV} = \frac{Q_{EV}}{m \cdot c_p} \quad (2)$$

where Q_{EV} is the evaporation heat, calculated from the evaporated fuel mass m_F and the evaporation enthalpy h_{EV} for a given fuel. The overall mass m consisting of fuel mass m_F and air mass m_A can be calculated from

$$m = m_F + m_A = m_F + m_F \cdot \lambda \cdot L_{ST} = m_F(\lambda \cdot L_{ST} + 1) \quad (3)$$

where λ is the air number which is the ratio of actual air/fuel-ratio to that one for stoichiometry for a given mixture, and L_{ST} is the stoichiometric fuel rate. The specific heat capacity c_p for the air/fuel mixture is obtained in the same way as the isentropic coefficient is determined, choosing an intake temperature of 340 K and a pressure of 1 bar

for this calculation. From the equations (2) and (3), the temperature cooling ΔT_{EV} is calculated by

$$\Delta T_{EV} = \frac{m_F \cdot h_{EV}}{m_F \cdot (\lambda \cdot L_{ST} + 1) \cdot c_p} = \frac{h_{EV}}{(\lambda \cdot L_{ST} + 1) \cdot c_p} \quad (4)$$

The evaporation enthalpy h_{EV} and the stoichiometric fuel rate L_{ST} for gasoline RON 95, isooctane 95 per cent (isooctane 95 per cent, *n*-heptane 5 per cent), and E85 (ethanol 85 per cent, gasoline 15 per cent) are listed in Table 3 [17].

With all this information the temperature cooling as a function of different λ values can be calculated theoretically; Fig. 6 shows the effect of the fuel on the compression temperature. It can be seen that a dramatic temperature cooling due to evaporation was found for E85. For $\lambda = 1$, E85 shows a theoretical temperature cooling of 71 K, while the corresponding value for isooctane 95 per cent is only 17, which is due to the smaller evaporation enthalpy of isooctane. The difference on the temperature cooling between these two fuels is up to 54 K. At the same λ , the temperature cooling in the compression stroke for gasoline RON 95 is 26 K, a little bit larger than that of isooctane 95 per cent.

Because E85 has the largest temperature cooling effect whereas isooctane 95 per cent has the smallest value, the temperature behaviour at different crank angles in the compression stroke are compared in Fig. 7 for both of them. The measurements were conducted at a constant engine load of 50 per cent and for an engine operation at $\lambda = 1$. For both fuels, the intake temperatures are the same. Again, the RCARS temperature results are displayed by

Fig. 6 Theoretical temperature cooling in consequence of evaporation enthalpy for different fuels in dependence of lambda (λ)

Fig. 7 Temperatures as a function of crank angle for two different fuels at a load of 50 per cent

open symbols with error bars together with isentropic calculations denoted by lines.

Figure 7 shows that, although the intake temperature of these two fuels is the same, the start temperature at BDC is different due to the different evaporation enthalpy of both fuels. The difference of isentropic temperature at the beginning of the compression stroke is 27 K. This value is much smaller than 54 K, which is the difference of the theoretical temperature cooling between these two fuels shown in Fig. 6. This distinction can be explained by two possible reasons. The first one is that, fuel is only partially evaporated at the position of BDC for the SOI of $-300^{\circ}\text{CA ATDC}$. Additionally, the heat transfer from the cylinder liner and cylinder head to the air/fuel mixture increases, which reduces the theoretical temperature cooling. A strong effect of partial evaporation is especially obvious for E85 due to its high evaporation enthalpy. However, the difference of isentropic temperature for both fuels is about 70 K at the end of compression, where the fuel is completely evaporated.

5 CONCLUSIONS

In the present study, measurements were taken of those quantities which have a high impact on the compression temperature in a highly charged SI engine, namely intake temperature, engine load, and fuel composition, while the SOI was kept

constant at $-300^{\circ}\text{CA ATDC}$. Dual-pump RCARS was used to achieve an accurate temperature estimation. As it is a point-wise probing technique, only one pair of two small optical accesses needed to be implemented in the cylinder block, and no large modification of the engine was required.

The isentropic calculations are based on the results obtained from RCARS spectra, and a much smaller inaccuracy for the temperature estimation could be provided by this combination. An impression of the accuracy improvement was demonstrated by different intake temperatures while other parameters were kept constant.

Another factor expected to influence compression temperature is the engine load. However, the temperature curves obtained for the variation of the engine load during the compression stroke were very similar. This is because the increased heat transfer from the cylinder surface to the air/fuel mixture at higher engine loads will be diluted by an increasing amount of air/fuel mixture that is fed into the cylinder in order to maintain the high loads. Furthermore, the calculation of the isentropic temperature was also not dependent on the load of the engine used in the present study.

With regard to the impact of the evaporation enthalpy on the compression temperature, isooctane 95 per cent and E85 were tested because of the significantly different evaporation enthalpy of these two fuels. At BDC the established compression temperature for E85 was 27 K lower than for isooctane 95 per cent, which was smaller than expected

because of the partially evaporated fuel. The temperature difference due to the cooling effect was continuously increasing during the compression stroke, and at TDC the compression temperature for E85 was 70 K lower than for iso-octane 95 per cent.

In summary, some impact factors on the compression temperature inside a highly charged SI engine have been investigated. In this investigation it has been demonstrated that the combination of RCARS and the isentropic calculations enables a noninvasive and *in situ* measurement of the compression temperature in harsh combustion environments. The results obtained could be used for a future simulation of the engine operation.

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APPENDIX

Notation

c_p	specific heat capacity
h_{EV}	evaporation enthalpy
L_{ST}	stoichiometric fuel rate
m	mass
P	pressure
Q_{EV}	evaporation heat
T	temperature
V	combustion chamber volume
ΔT_{EV}	temperature cooling
φ	equivalence ratio
κ	isentropic coefficient
λ	ratio of actual air/fuel-ratio to stoichiometry for a given mixture
ω	wave number