

Future Specification of Automotive LPG Fuels for Modern Turbocharged DI SI Engines with Today's High Pressure Fuel Pumps

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ABSTRACT

Liquefied Petroleum Gas direct injection (LPG DI) is believed to be the key enabler for the adaption of modern downsized gasoline engines to the usage of LPG, since LPG DI avoids the significant low end torque drop, which goes along with the application of conventional LPG port fuel injection systems to downsized gasoline DI engines, and provides higher combustion efficiencies. However, especially the high vapor pressure of C_3 hydrocarbons can result in hot fuel handling issues as evaporation or even in reaching the supercritical state of LPG upstream or inside the high pressure pump (HPP). This is particularly critical under hot soak conditions. As a result of a rapid fuel density drop close to the supercritical point, the HPP is not able to keep the rail pressure constant and the engine stalls. Thus, in order to enable wider use of LPG and to assist the process of standardization, a limitation for maximum content of C_3 fuel components - as propane and propene - is required as a key enabler for LPG DI applications. In order to determine a maximum C_3 limit, hot idle investigations were performed on a modern turbocharged 4-cylinder direct injection spark ignition (DI SI) engine operated in a climate cham. Furthermore, stationary idle tests were carried out on a single cylinder engine with four different LPG fuels, all in accordance with EN 589. The experimental investigations indicate a maximum content of 70 % (m/m) C_3 fuel components as an upper limit for an LPG DI injection concept, when a HPP, which is based on state-of-the-art HPP technology for gasoline engines, is used. For a maximum propane content of 70 % (m/m), pump functionality can be maintained with a fuel pressure of approximately 45 bar upstream the HPP at fuel temperatures of about 110 °C during hot idle. For lower pressures upstream the HPP, cooling measures need to be implemented.

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INTRODUCTION

Due to the worldwide tightening of CO_2 legislation and increasing requirements regarding a reduction of pollutant emissions, internal combustion engines have gone through a tremendous progress during the last decades. Therefore, modern spark ignition engines feature a large number of innovative technologies, such as direct injection, exhaust gas turbocharging, variable valve trains, thermal management, start & stop, intelligent generator management systems and friction reduction, for instance by using demand-controlled oil and water pumps. However, to achieve the drastic CO_2 reduction, which is required to meet future targets, further optimization has to be made in combustion processes. Technologies like exhaust gas recirculation, lean combustion, innovative ignition systems, water injection and variable compression ratio have to be considered.

Moreover, an enormous potential in CO_2 reduction can be realized through the use of alternative fuels. Gaseous fuels like LPG are a sensible short- and mid-term alternative. In Europe, LPG is already the most-utilized alternative fuel for automotive use with an already adequate core infrastructure in place [1, 2, 3]. According to the European fuel standard EN 589 [4], automotive LPG mainly consists of C_3/C_4 paraffins, but also mono-olefins are allowed as fuel components. In changing phase from gaseous to liquid state, these molecules undergo a significant volume reduction, which results in considerable benefits in terms of transportation and storage aspects. From a combustion perspective, LPG is an interesting alternative fuel for modern spark ignition engines, due to its high knock and pre-ignition resistance compared to gasoline. Furthermore, the significantly enhanced auto-ignition characteristics in the low and

mid temperature regime, especially for propane [5]. This enables higher compression ratios and therefore higher combustion efficiencies. The thermodynamic efficiency of the LPG fueled engine can be improved by direct injection of LPG. This concept allows high torques at low engine speeds in comparison to external LPG mixture formation systems [6]. Furthermore, LPG DI reveals reduced pollutant emission characteristics. Gaseous emissions, such as HC, can be reduced significantly [7, 6] mainly due to severe flash-boiling and a better mixture formation of LPG compared to gasoline [8]. To meet future emissions legislation for DI SI engines, it is noteworthy that direct injection of LPG offers tremendous particulate emissions reduction [4, 7, 9, 10, 11, 12, 13]. Besides all these numerous positive aspects, the biggest challenge for the direct injection of LPG is the hot fuel handling. Considering state-of-the-art gasoline direct injection engines, high fuel pressure is built up using mechanically driven high pressure fuel pumps upstream of the fuel rail. When using such a high pressure fuel pump to pressurize LPG fuels, the high vapor pressure, especially of propane and propene, can result in evaporation of LPG upstream or inside the HPP, since the thermal conditions of these pumps are considerably influenced by heat transfer from the engine. As a result, the engine stalls, due to the fact that the HPP cannot compress a gas or a vapor. Furthermore, the supercritical state can be reached. This physical state is defined by the critical point, in which the saturated liquid and saturated vapor curves meet. It is heavily dependent on the components of the fuel mixture. Pure propane/propene transits to the supercritical state at a pressure of 42.5/46.7 bar, which is lower than usual fuel pressures in the high pressure part of direct injection systems. The corresponding critical temperatures are 96.7 °C/92.4 °C. Therefore, this physical state is particularly relevant under hot soak and hot idle conditions, in which such high fuel temperatures within the fuel system can be reached. As a result of a rapid fuel density drop close to supercritical point, the high pressure pump is not able to keep the rail pressure constant and the engine stalls. Countermeasures to avoid LPG boiling and the supercritical state in the fuel system at engine operation and hot soaks are required for an increased usage of LPG direct injection applications in the future. On the one hand, cooling measures can be considered, which keep LPG in the liquid state. On the other hand, the fuel pressure at inlet of the HPP can be increased, in order to counteract the high compressibility at high fuel temperatures. Moreover, an optimized HPP design, such as an increased compression ratio of the pump, can be reasonable. However, there is still little knowledge about the effectiveness and suitability of one of these methods or even a combination of them, depending on the gas mixture components.

This paper is believed to be the first study on LPG hot soak and hot idle investigations performed on a modern turbocharged 4-cylinder DI SI in a climate cell. In order to enable wider use of LPG fuels and to assist the process of standardization, a limitation for maximum content of C₃ fuel components will be recommended. For these purposes, four different LPG fuel formulations, which all are in accordance to the current LPG fuel standard EN 589, have been tested. Besides engine warm-up and hot soak tests, variations of the

fuel pressure at inlet of the HPP have been investigated. Comparisons of time durations to build up fuel rail pressure are made as a function of the C₃ content. Furthermore, the injection and combustion behavior is studied in stationary tests on a single cylinder engine for different physical states.

FUELS

Table 1 summarizes properties of the fuels investigated in this study.

Table 1. Fuel properties

	RON 95 E5	LPG 1	LPG 2	LPG 3	LPG 4
Motor octane number	86.3	89.4	94.6	93.3	91.5
Methane number	—	26	28	31	15
Density (15 °C) / kg/m ³	749	515	526	520	560
Critical temp. / °C	—	93.6	113.7	102.8	133.6
Critical press. / bar	—	44.7	42.6	43.8	41.7
Ethane / % (m/m)	—	0.4	0.3	0.8	0.1
Propane / % (m/m)	—	48.3	67.4	82.1	16.1
Propene / % (m/m)	—	50.2	0.2	0.2	8.2
n-Butane / % (m/m)	—	0.4	21.2	2.3	50.7
Isobutane / % (m/m)	—	0.7	10.8	1.6	24.6
C ₄ olefins / % (m/m)	—	—	0.03	13.0	0.2
n-Butene / % (m/m)	—	—	—	7.9	—
Isobutene / % (m/m)	—	—	0.03	0.3	—
(Z)-2-Butene / % (m/m)	—	—	—	1.3	—
(E)-2-Butene / % (m/m)	—	—	—	3.4	—
C ₅ paraffins / % (m/m)	—	—	0.1	0.1	0.1
Spec. enthalpy of vaporiz. / kJ/kg _{air}	29.5	28.2	25.2	26.1	23.9
Stoichiometric air requirement / 1	14.17	15.19	15.56	15.52	15.4
Lower heating value / MJ/kg	41.99	46.08	46.17	46.20	45.82

All LPG fuels are market relevant fuels according EN 589 [4]. A conventional RON 95 E5 according to EN 228 [14] serves as reference fuel for all investigations. The selection of the LPG fuels was made in order to cover a wide range of possible fuel components. LPG 1 has the highest C₃ content according to EN 589, LPG 2 represents a typical winter grade LPG, whereas LPG 3 has a high propane content and LPG 4 a high butane content. The last two fuel formulations represent homologation fuels according to [15].

The motor octane number of the LPG fuels is calculated based on the mass fractions according to EN 589, while the methane number is calculated in accordance to the so called AVL algorithm partially standardized in DIN 51624 [16] and established in [17, 18, 19]. Comparing the critical points, the strong dependence on the fuel mixture components is evident. Isobutane/n-Butane has a critical temperature of 134.7 °C/152 °C and a critical pressure of 37/38 bar. Therefore, the higher the content of C₄ paraffins, the higher is the critical temperature and the critical pressure. Furthermore, compared to gasoline, the lower density of all LPG fuels can be noted. While the heating value is higher for the LPG fuels, the lower densities lead to a lower volumetric heating value in comparison to conventional gasoline. Due to high vapor pressures, high propane and especially high propene contents are reflected in high enthalpies of vaporization approximated via Clausius-Clayperon equation.

EXPERIMENTAL SET-UP

Series Production Engine

Hardware

As a modern downsized engine with direct injection, the 1.6 l inline turbocharged gasoline engine of the Ford Motor Company was chosen, which is presented in detail in previous publications [5]. A short summary of the technical data of the engine is given in Table 2.

Table 2. Series production engine hardware specification

Bore / mm	79
Stroke / mm	81.4
Number of cylinders	4
Cylinder displacement / cm ³	399
Valves per cylinder	4
Compression ratio / 1	10
Injection system	Bosch Multihole
Maximum fuel pressure / bar	150

The engine is quantity controlled with a stoichiometric combustion system and a conventional valve train with variable cam phasing on intake and exhaust side. The four valve combustion chamber roof with central arranged injector ($Q_{\text{stat}} = 12.5 \text{ cm}^3/\text{s}$ at 10 MPa with n-Heptane) and spark plug aligned with the crankshaft axis has a compression ratio of CR = 10 [20]. For the control of the engine, an

RCP hardware system, consisting of a dSPACE MicroAutoBox II and a RapidPro was chosen. Measurements were performed with the ControlDesk NG experiment software. A base engine model was used employing a calibration based on the series ECU calibration, due to the fact that the main focus of the investigations was set on the behavior of the injection system and the fundamental functionality of the HPP under high fuel temperatures.

For the direct injection of LPG, the conventional components of the gasoline high pressure injection system of the engine were used (high pressure pump, fuel rail, fuel injectors). No modifications of the fuel-carrying components have been made, except for the fact that a thermocouple has been installed in the fuel line between the HPP and the fuel rail, in order to be able to measure the fuel temperature at rail inlet. Further thermocouples were positioned at the surface of the HPP, the fuel rail and the solenoid housing of the injector of cylinder 3 (counted from flywheel side). An additional middle pressure pump has been applied upstream the HPP in order to be able to vary the HPP inlet pressure. To keep the fuel temperature at the inlet of the HPP constant, an electrical heater was mounted to the fuel line upstream the HPP with a distance of about 20 cm to the pump to avoid direct heat transfer. For fuel changes, the system was completely drained and flushed with the new fuel to minimize possible contamination by other fuels.

Climate Cell Instrumentation

For the hot soak investigations, the engine was mounted into an idle rig. The cylinder pressure was measured with Kistler 6041 pressure transducers mounted flush in the combustion chamber roof of each cylinder. Sampling was performed via Kistler 5011 charge amplifiers and a FEVIS 3.2.4. Dynamic intake and exhaust gas pressures were measured with Kistler 4045 A5 pressure transducers and sampled in 1° CA resolution. Measurements of static pressures and temperatures were performed with piezoresistive pressure transmitters and thermocouples. Engine oil and coolant was not conditioned by any external conditioning system. The coolant circuit contains two controllable valves and a thermostat, in order to allow different cooling strategies. The required air mass flow to cool down the coolant was delivered by an electric fan. The fan was located in front of the cooler in a manner that a coolant temperature of up to 120 °C could be reached. The temperature in the climate cell was set-up to ~30-35 °C to create warm ambient conditions, representing a hot summer day in northern Europe. Therefore, the temperature of the intake air, which is sucked into the intake snorkel, equals the temperature of the air inside the climate cell.

Experimental Procedure

All tests were performed in homogenous operation mode with single injection into the intake stroke. Valve timing was set to maximum negative valve overlap of 38° CA. Pre-investigations had shown that a constant idle phase at high engine speed of 4000 1/min, followed by

a hard shutdown, results in fuel temperatures of only $\sim 60^\circ\text{C}$, ~ 25 minutes after shutdown. This is also known as hot soak because heat is “soaking” from the engine to the fuel. Higher fuel temperatures were found to be a result of long constant idle phases. Therefore, the engine was warmed up at a constant idle speed of 800 1/min until the high temperature level stabilized. The results of this warm-up phase are presented in the result chapter exemplarily for LPG 1. This warm-up was applied right before each HPP inlet pressure variation or hot start.

Single Cylinder Research Engine

Hardware

The steady state idle tests were conducted on a homogeneously operated, DI SI single cylinder research engine, presented in previous publications [5, 21]. Injector and spark plug are placed in central cross position in the combustion chamber roof, which means that the spark plug is installed between the exhaust valves, while the injector is installed between the intake valves. The tumble ratio of the symmetrical filling ports was increased by an additional tumble flap up to 4.1 [22]. Table 3 summarizes the most important engine parameters.

Table 3. Research engine hardware specification

Bore / mm	75
Stroke / mm	82.5
Displacement / cm^3	364
Number of valves	4
Compression ratio / 1	10.93
Injection system	Bosch HDEV 5.2
Maximum fuel pressure / bar	200

The direct-injection gasoline injector ($Q_{\text{stat}} = 13.5 \text{ cm}^3/\text{s}$ at 10 MPa with n-Heptane) is a six-hole solenoid actuated injector and was also used for LPG direct injection. Concerning the fuel system, the HPP used was volume flow controlled. To reach the supercritical state, the fuel was heated up by an electrical heater mounted to the complete fuel line downstream the high pressure pump including the fuel rail and pipes upstream the injector.

Test Bench Instrumentation

The single cylinder engine instrumentation was similar to that used for the series production engine and includes only minor differences, such as the cylinder pressure transducer Kistler A6043 A100. In total, 200 consecutive cycles were measured and averaged. The engine was coupled to an eddy-current brake and an electric motor to maintain the desired engine speed with an accuracy of ± 1 1/min. The intake air mass flow was measured with an ultrasonic air mass meter. The gaseous exhaust gas composition was determined from a partial mass flow of exhaust gases. The exhaust gas sample was fed to the following emission analyzers via a transfer line heated to 193°C :

- HC: Flame ionization detector (Rosemount NGA 2000, calibrated with C_3H_8)
- O_2 : Paramagnetic oxygen analyzer (Rosemount NGA 2000)
- CO and CO_2 : Infrared gas analyzer (Rosemount NGA 2000)
- NO_x : Chemiluminescence analyzer (Eco Physics 700 EL ht)

Experimental Procedure

The investigation on the single cylinder engine focuses on the injection, combustion as well as the emission behavior of LPG direct injection during idle. Therefore, the engine speed and the load were set to 800 1/min and 0.7 bar IMEP as for the hot idle investigations. The selected settings are summarized in Table 4.

Table 4. Single cylinder engine calibration settings

Engine speed / (1/min)	800
IMEP / bar	0.7
Start of injection / $^\circ\text{CA BTDC}$	280
Fuel rail pressure / bar	40/50
IVO (1mm) / $^\circ\text{CA ATDC}$	13
EVC (1mm) / $^\circ\text{CA BTDC}$	32
Relative air/fuel-ratio / 1	1
Air temp. in intake manifold / $^\circ\text{C}$	25
Coolant and oil temperature / $^\circ\text{C}$	90

A rail pressure of only 40/50 bar was chosen in order to investigate the effect of decreasing density close to the critical point more clearly. Valve timing was set to maximum negative valve overlap of 45°CA . With these settings spark timing sweeps for different fuel temperatures were performed for all LPG fuels and compared to the reference fuel.

RESULTS

Warm-Up

Preliminary investigations had shown, that long, constant idle operation represents the most severe operation mode regarding changes of physical states of LPG fuel components. During such a period of constant idle, the engine got fully warmed up and a rail pressure drop was observed at certain fuel temperatures inside the HPP dependent on the LPG fuel. Due to the fact, that the critical temperatures of the tested LPG fuels differ significantly, the rail pressure drop is expected to occur at different fuel temperatures during the investigated warm-up phases. Figure 1 exemplarily shows the warm-up phase with LPG 1.

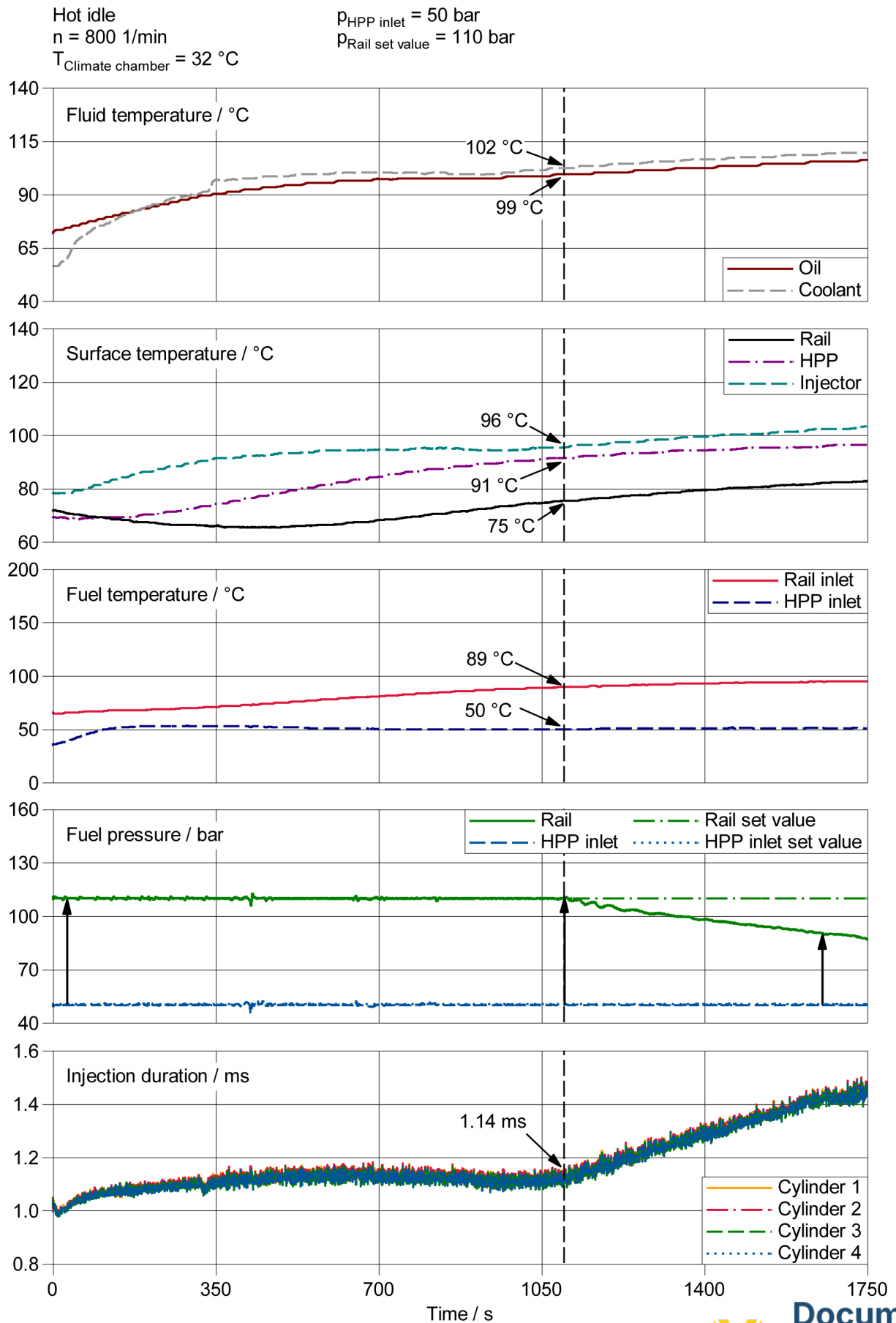


Figure 1. Warm-up phase with LPG 1

This fuel can be seen as the “worst case” fuel with regard to hot idle conditions due to the high C_3 content. It is evident, that high oil and coolant temperatures in the range of 110 °C are reached during constant idling at 800 1/min. At 337 seconds the thermostat opens, cooled water out of the radiator enters the engine, while hot water flows out of it. This results in a slight increase of the coolant temperature at engine outlet. The highest surface temperatures during the warm-up phase were observed at the injector solenoid housing (cylinder 3). At the inlet of the HPP, the fuel temperature was conditioned to 50° C, whereas the fuel pressure was set to 50 bar. Therefore, the physical state is either liquid or supercritical depending on the fuel temperature and outgassing of the fuel is prevented, neglecting occurrence of local cavitation inside the high pressure fuel system. While the fuel flows through the HPP and the pipe between the HPP and the rail, it is heated through the heat transfer from the engine and the fuel temperature increases steadily with continuous engine warm-up. This can be seen with regard to the fuel temperature at the inlet of the fuel rail, which reached 94 °C at the end of the measurement. The required fuel rail pressure was set to 110 bar. In [Figure 1](#), the fuel pressure increase from the low pressure level of 50 bar at the inlet of the HPP to the rail pressure is illustrated exemplarily by three arrows at different times in the diagram of the fuel pressure. Up to 1125 seconds, the HPP is able to deliver the required rail pressure. However, a clear rail pressure drop can be seen, which starts at this time and continues till the end of the measurement. This can be explained using [Figure 2](#), in which the isobaric density diagram of propene is shown.

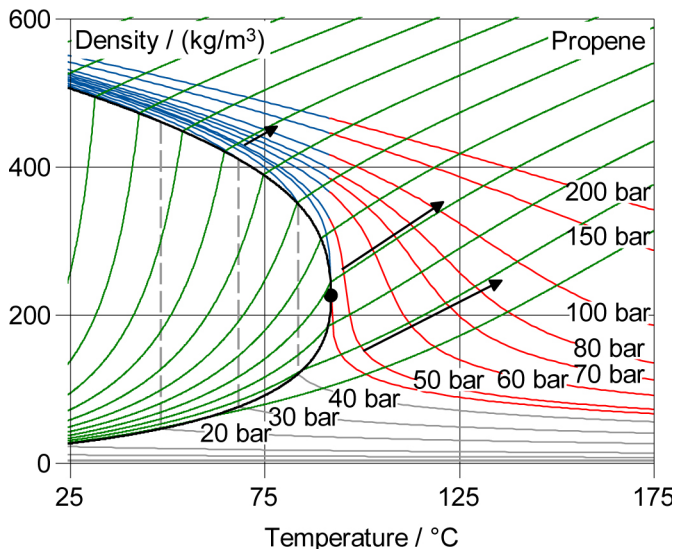


Figure 2. Isobaric densities [23] and isentropes of propene [24] with three estimated isentropic changes of state of LPG 1 during the warm-up phase corresponding to [Figure 1](#)

Propene is suitable for an estimation of a change of state of LPG 1 due to the fact, that 50.2 % (m/m) propene is contained in this fuel and the critical temperature of propene is lower than that of propane. Besides the isobars and the two-phase region, which ends in the critical point at 92.4 °C, the isentropes are depicted. It is obvious, that the isobars of pressures higher than 46.6 bar show a rather constant density gradient in a temperature range of 0 °C to ~80 °C. However, as temperature increases the isobars of 46.6 bar to 80 bar show a heavily nonlinear behavior between 80 °C and 140 °C. For instance,

the density changes about -5 %/°C at 50 bar and 100 °C. The three highlighted arrows in this Figure correspond to the arrows of the warm-up in [Figure 1](#). They represent three estimated changes of state of the fuel due to compression by the HPP. The thermodynamic changes of state are estimated under the following assumptions. The compression is considered to be isentropic. Furthermore, at the beginning of the compression the state of the fuel is described by the estimated fuel temperature inside the HPP and the pressure at the inlet of it. The temperature inside the HPP was estimated to be an average from oil and HPP surface temperature, since the HPP is exposed to the temperature of the climate cell and its piston is driven by fuel pump cams of the exhaust camshaft, which are in turn lubricated by the warm oil of the engine. Thus, the fuel is heated up severely inside the pumping chamber, because the upper part of the piston is in contact with the fuel, while the lower part is in contact with the warm engine oil. Moreover, the fuel flows through the inner drillings of the HPP, which wall temperatures are influenced by the heat transfer from the valve cover. Due to this fuel heating process the estimated temperature of LPG 1 inside the HPP is 70.5 °C at the beginning of the warm-up process. The changes of state at this temperature correspond to the left arrow in [Figure 1](#) and [Figure 2](#). Thereby, the HPP is able to pressurize the fuel from 50 bar to the required rail pressure of 110 bar. The middle arrow illustrates the change of state at the beginning of the rail pressure drop at an estimated fuel temperature inside the HPP of 95 °C. Further temperature increase results in progressing rail pressure drop. Finally, the right arrow shows the change of state at an estimated fuel temperature inside the HPP of 100.5 °C. At this time, the HPP is only able to compress the fuel from 50 bar to 89 bar. The rail pressure drop with increasing temperatures can be explained by two reasons. Firstly, the strong decrease of density results in a reduced filling of the cylinder of the HPP. The reduced filling in turn limits the maximum deliverable mass flow to the fuel rail. Secondly, the isentropic compressibility is increasing as it can be seen by the increasing distance between each isobar at temperatures higher than the critical one. The isentropic compressibility K_S limits the pressure at the end of compression. It is defined as [25]:

$$\kappa_s = -\frac{1}{V} \cdot \left(\frac{\partial V}{\partial p} \right)_s \quad (1)$$

In order to examine the effect of an increasing isentropic compressibility with increasing temperatures according to [Figure 2](#), an exemplary calculation will be presented. For this purpose, the maximum achievable pressure at the end of compression is determined for two different temperatures, 80 °C and 100 °C, in case of propene. The temperatures refer to the beginning of compression.

The initial pressure is 50 bar. Assumptions are an isentropic compression, a constant pump compression ratio ϵ_{HPP} of 1.15, a constant pump displacement and a constant mass of substance. At a temperature of 80 °C and a pressure of 50 bar, the density of propene is 395.14 kg/m³. In [Figure 2](#) this state can be found at the end of the linear density range of this isobar. At the end of compression the density ρ can be calculated to 454.3 kg/m³ with the following equation.