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The Effect of Humidity on the Knock Behavior in a Medium BMEP Lean-Burn High-Speed Gas Engine

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ABSTRACT

The effects of air humidity on the knock characteristics of fuels are investigated in a lean-burn, high-speed medium BMEP engine fueled with a $\text{CH}_4 + 4.7$ mole% C_3H_8 gas mixture. Experiments are carried out with humidity ratios ranging from 4.3 to 11 g $\text{H}_2\text{O}/\text{kg}$ dry air. The measured pressure profiles at non-knocking conditions are compared with calculated pressure profiles using a model that predicts the time-dependent in-cylinder conditions (P, T) in the test engine ("combustion phasing"). This model was extended to include the effects of humidity. The results show that the extended model accurately computes the in-cylinder pressure history when varying the water fraction in air.

Increasing the water vapor content in air decreases the peak pressure and temperature significantly, which increases the measured Knock Limited Spark Timing (KLST); at 4.3 g $\text{H}_2\text{O}/\text{kg}$ dry air the KLST is 19 °CA BTDC while at 11 g $\text{H}_2\text{O}/\text{kg}$ dry air the KLST is 21 °CA BTDC for the same fuel. Excellent agreement is observed between the calculated knock resistance (using the Propane Knock Index, PKI) and the measured knock resistance (KLST) for the range in water content in air studied in this work. Since the effect of water on autoignition delay time is negligible, the observed increase in knock resistance of the fuel-air mixture is due a decrease in pressure and temperature of the end gas with increasing water content in as a result of changes in the mass burning rate, and thermophysical properties of the fuel-air mixture.

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INTRODUCTION

The occurrence of engine knock is a major limitation for high-efficiency, low-emissions gas engines. Mild engine knock increases pollutant emissions, while severe knock can cause engine failure or physically damage the engine, and should thus be avoided.

Engine knock is caused by autoignition of the unburned fuel gas mixture, the so-called end gas, ahead of the propagating flame in the engine cylinder. Autoignition occurs when the rate of reaction, and thus the rate of heat release, grows exponentially with temperature and the overall reaction will auto-accelerate. The tendency of the gas mixture to autoignite depends on the fuel gas composition, equivalence ratio and the temperature and pressure of the end gas. For example, studies performed in shock tubes and rapid compression machines (RCMs) showed that the autoignition delay time reduces with increasing temperature and pressure [1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13]; as a result the occurrence of knock in the end gas is critically dependent upon the pressure and temperature history of the unburned mixture during the engine cycle. The maximum pressures (and thus temperatures) of the end gas are dependent on the combination of mechanical compression from piston motion and

compression from the progressive combustion of the charge, related to the combustion "phasing". Several studies have shown that changes in fuel gas composition lead to substantial variations in the thermodynamic properties of the fuel gas mixture [14] and in the "phasing" of the combustion process [15, 16, 17, 18] and therefore these properties need to be taken into account to predict the time-dependent pressure and temperature of the end gas accurately.

Besides fuel gas composition, the knock behavior of an engine is also influenced by the ambient conditions such as air humidity. The ambient conditions vary strongly upon the geographical locations and, depending upon the country, large seasonal changes may occur. It is well known that an increase in humidity will reduce the combustion rate and increase the heat capacity. Both effects result a decrease in the end gas temperature and pressure, which influences the knock margin and NO_x emission in a positive way [19, 20]. Understanding the effects of humidity on engine knock will enable OEMs to optimize the new generation of engines and engine management systems to guarantee maximum non-knock performance for their engines under all weather conditions. However, the traditional methods for characterizing the knock

resistance of gaseous fuels, such as the methane number developed by AVL [21] in which the knock propensity of a given fuel is compared with an equivalent methane/hydrogen mixture, do not take into account the effect of humidity.

In recent years we have derived a method to characterize gases for their knock resistance based on the combustion properties of the fuel-air mixtures [22, 23, 24]. In this method, engine knock is predicted by modeling autoignition of the compressed end gas during the combustion cycle, in which the changes in pressure and temperature history of the end gas, caused by changes in fuel composition, is taken into account [22, 23, 24]. To rank the knock resistance of different gases, a propane-based scale (Propane Knock Index, PKI) has been developed. [22, 23, 24]. In this scale, the knock resistance for a given gaseous fuel mixture is expressed as an equivalent fraction of propane in methane under the same engine conditions.

In this paper, the methodology is applied to characterize the knock resistance of gaseous fuels with varying humidity in the combustion air. We demonstrate the veracity of the extended knock model by comparing the predictions of autoignition of the end gas, expressed as the computed PKI, with the Knock-Limited Spark Timing (KLST) measured in the same lean-burn engine used previously for a range of humidity levels in combustion air (RH values ranging from 22.4% to 76.6 %) and using $\text{CH}_4 + 4.7 \text{ mole\% C}_3\text{H}_8$ gas mixtures as fuel. The predicted variation in knock resistance with varying humidity is shown to be in excellent agreement with the KLST measurements.

EXPERIMENTAL METHODS

The experiments are performed using the 208 kW 1500 rpm lean-burn ($\lambda=1.5$) turbocharged, intercooled 6-cylinder gas engine designed for combined-heat-and-power (CHP) duty, used previously [22, 23, 24]. Table 1 lists the key specifications of this engine. We remark here that this CHP system maintains constant equivalence ratio, engine speed and power output. As described earlier [22], the engine management systems and additional instrumentation provided precise adjustment, monitoring and registration of all relevant engine settings and process conditions. The in-cylinder pressures in all cylinders were measured with Kistler type 6052 piezoelectric pressure sensors and a Kistler type 5011 charge amplifiers connected to a Smetec Combi-Pro indication system. A crankshaft-mounted pulse generator provided 0.1 °CA resolution for the cylinder pressure data acquisition. The Smetec Combi-Pro system was also used for knock detection through monitoring of the maximum amplitude of pressure oscillations in the high-pass-filtered cylinder pressure data in a window of consecutive cycles. Threshold settings used were 2 bar for the amplitude and 30 cycles for the window. For knock-limited spark timing experiments ([14], see below), the knock limit was chosen to be one knock event or cluster of knock events within a 15 minute steady-state test run.

Table 1. Key specifications of the test engine.

Engine make & type	MAN E2876LE302
Engine configuration	- 6 cylinder in-line - turbocharged - intercooled
Combustion system & combustion stoichiometry	- mono-gas - open chamber - lean-burn
Rated power & speed (corresponding BMEP)	208 kW at 1500 rpm (13.0 bar)
Bore x Stroke	128 x 166 mm
Compression ratio	11.0:1

During the experiments, fuel for the test engine was delivered by a gas mixing unit. This unit allows on-stream variation of the fuel gas composition by independent adjustment of the flow rates of up to six source gas streams, up to a capacity of ~4 MW.

The fuel gas composition used to assess the effects of water fraction on the combustion phasing and knock behavior is a methane/propane (4.7% propane) mixture. The propane was added to reduce the knock resistance of the fuel as a countermeasure to the expected knock-inhibiting effect of increasing the water fraction in the combustion air [19, 20]. This measure assured that the advance in spark timing required for inducing knock in the KLST experiments would remain within an acceptable range as compared to the standard ignition timing. The water fraction in the combustion air is controlled by using an industrial-type air dryer for the experiments requiring a lower water vapor fraction than that present in the ambient air. For the experiments requiring an elevated water fraction in the combustion air, a water injection system was developed in-house comprising a hot-water nozzle-spray system. A 20 m length of tubing, connected to the inlet of the air filter housing of the engine, was used to allow the injected water to evaporate fully and form a homogeneous water vapor/air mixture. Sensors were used to measure the resulting relative humidity, pressure and temperature conditions of the combustion air entering the air filter housing. The fuel gas composition, engine intake manifold conditions and combustion air water vapor fractions are presented in Table 2.

The fuel composition delivered to the engine, as measured by a gas chromatograph, was recorded at roughly 4 minute intervals. Once the composition of the fuel/air mixtures at equivalence ratio $\lambda=1.5$ and the other relevant cylinder parameters are set, the pressure traces of the gases under non-knocking condition, needed to test the combustion phasing model discussed below, are measured at a constant spark timing of 14°CA before top dead center (BTDC). Next, the spark timing is gradually advanced up to the point of the onset of borderline knock, taken as Knock-Limited Spark Timing or KLST. For the sake of maintaining a near-constant fuel and air composition, each non-knocking (“phasing”) experiment was followed by the corresponding KLST experiment. The typical reproducibility of the measured KLST is $\pm 0.75^\circ \text{CA}$.

Table 2. Fuel gas compositions (mole fractions), combustion air water vapor fractions and experimental engine settings

	condition 1	condition 2	condition 3	condition 4
Methane	0.9521	0.9538	0.9542	0.9545
Ethane	0.00003	0.0000	0.0002	0.0000
Propane	0.04708	0.0454	0.0449	0.0447
n-Butane	0.00016	0.0002	0.0002	0.0002
i-Butane	0.00051	0.0006	0.0006	0.0005
P _{H2O} , Sat. kPa	3.72	2.70	2.35	2.27
P _{H2O} , kPa	0.69	1.08	1.4	1.74
Humidity, %	18.4	40.1	59.4	76.6
g H ₂ O/kg dry air	4.30	6.80	8.80	11.00
Air factor, λ	1.54	1.54	1.54	1.54
P _{intake} , bar	2.10	2.12	2.12	2.14
T _{intake} , K	337.3	337.9	336.8	337.3
KLST, °CA BTDC	19.0	20.0	20.5	21.0

KNOCK MODEL

The knock model discussed above [22, 23, 24] consists of two separate modules: a combustion phasing model and an autoignition model. The combustion phasing model calculates the time-dependent changes in the pressure and temperature of the unburned end gas caused by both mechanical compression and thermal compression for a given fuel/air composition. The computed pressure history from the combustion phasing model serves as input to the autoignition model, which uses this data to compute the autoignition of the fuel/air mixture. Below we discuss how these models have been adapted to account for humidity.

COMBUSTION PHASING MODEL

We first adapt our two-zone thermodynamic model [22, 23, 24] to account for humidity. A detailed description of this model can be found elsewhere [22]. The input data needed for the model are the engine geometry, in-cylinder conditions (P, T) at closure of the inlet valve, operating conditions such as spark timing, air factor, humidity and the fuel composition.

The simulation solves the time-dependent states of the two zones in the combustion chamber, using the conservation equations for mass and energy, and the ideal gas equation of state [22]. Until the time of ignition and after the mixture is completely burned the mixture is simulated as a single zone. Immediately after spark ignition the burned zone is created. The combustion process is simulated using the approach proposed by Chmela [25] and applied by others [26, 27] where the mass burning rate (dm_b/dt) is related to the laminar burning velocity (S_L) and a single fit parameter independent upon the gas composition (C_{model}), according to [22]:

$$\frac{dm_b}{dt} = \frac{C_{model}}{Le_{eff}} (1 - x_B) \rho_u S_L t^2 \quad , \quad (1)$$

where Le_{eff} is the effective Lewis number, x_B is the burned mass fraction, ρ_b is the burned gas density and t is the time.

An essential aspect of inclusion of the burning velocity of the end gas into the combustion phasing sub model is parameterizing the burning velocity, which changes during compression and combustion. To do so, we use a well-known scaling method [28]:

$$S_L = S_{L0} \left(\frac{T}{T_0} \right)^\alpha \left(\frac{P}{P_0} \right)^\beta \quad (2)$$

where S_{L0} is the laminar burning velocity of the gas-air mixture for the pressure (P_0) and temperature (T_0) at the spark timing, calculated using PREMIX code [29] in the CHEMKIN II library [30], using USC Mech Version II reaction mechanism described in [31]. As described previously [22], the parameters α and β which take into account the temperature and pressure dependence of burning velocity, are derived from our own database generated from numerical laminar burning velocities for a broad range of gas compositions, and operating conditions (pressure, temperature and air/fuel ratio, $\lambda=1.5$) relevant for the engine used. To include the effect of water vapor we extended our database by including numerical laminar burning velocities for a wide range of water fractions in combustion air. In this way, only one laminar burning velocity needs to be computed in detail for each phasing simulation (at the spark timing); this considerably shortens the computation time needed for the model.

AUTOIGNITION MODEL

To account for the changes of pressure and temperature of the unburned end-gas during the engine cycle, we derive the specific volume of the assumed adiabatic core [32] of the unburned end gas from the pressure history computed by the combustion phasing model as discussed further below. Using the specific volume, the composition of the fuel/air mixture and the initial intake pressure and temperature as input in the simulations the pressure history is computed using the SENKIN code [33] in the CHEMKIN II library [30] and the chemical mechanism from [6, 7, 8, 9, 10], which includes the effect of H₂O on autoignition chemistry. Obviously, as is illustrated in Figure 1, when using non-knocking conditions the simulation should (and does) not predict autoignition of the end gas. To create engine knock in the simulations we artificially increase the intake manifold temperature of the mixture until knock occurs in the simulations [32]. This temperature offset is chosen such that the model predicts autoignition of the end gas near the end of the burn period for an arbitrarily chosen reference fuel. The rapid increase in the simulated pressure trace in Fig. 1 shows "knock" at 90% heat release for condition1 (see Table 2), which required an increase in the simulated intake temperature by 36 K; the sudden spike observed in the simulated pressure history is used here to indicate knock in the end gas.

Figure 1. Measured pressure history and calculated autoignition for condition 1 subjected to the complete measured pressure and temperature history, at both the actual intake temperature T_0 and at T_0+36 K

RESULTS AND DISCUSSION

Effect of changing the water fraction in air on engine knock

Figure 2 shows the measured KLST for the conditions from Table 2. Clearly, an increase in the water content in air substantially increases the Knock Limited Spark Timing (KLST), as is known in practice. From this we conclude that the presence of water vapor increases the resistance to knocking of the gaseous fuel/air mixture substantially.

To investigate the physical/chemical origin of how the water fraction in air affects the engine knock, we analyze the chemical changes in autoignition behavior of the fuel/air mixture itself with increasing humidity, and separately we consider to which extent the changes in the in-cylinder conditions (P/T) caused by changes in air humidity affects the autoignition behavior using the experimental data and the knock model.

Figure 2. Measured knock-limited spark timing (KLST) for conditions 1-4 versus water fraction in air (see Table 2). The error bars indicate the uncertainty of the KLST measurements (± 0.75 °CA)

Analysis of Changes in Autoignition Behavior upon Varying Water Vapor Content in Air

To analyze the effect of water vapor on the autoignition behavior of gaseous fuels we performed a number of simulations at constant pressure and temperature conditions relevant to the in-cylinder conditions of the used gas engine; $T=1020$ K, $P=75$ Bar and $\lambda=1.5$. The calculations have been performed for different fractions of water in methane and methane/propane (4.5% propane). As can be seen in Figure 3 the presence of 4.5% propane reduces the autoignition delay time compared to the methane mixture, while the addition of water results in a marginal increase of the autoignition delay time for both mixtures. From this we conclude that the substantial increase in the knock resistance with increasing humidity as observed in Figure 2 is not related to the autoignition chemistry itself but to changes in the pressure and temperature history of the unburned end gas caused by variations in the water fraction in air, as will be analyzed below.

Figure 3. Computed effect of water on autoignition delay time of methane and methane/propane (4.5% propane) at DNV GL test engine conditions (1020 K, 75 bar, $\lambda=1.5$) at knocking conditions.

Analysis of Changes in Pressure and Temperature of the End Gas with Varying Water Vapor Content in Air

Figure 4 presents a portion of the measured cylinder pressure profiles under non-knocking conditions for methane with 4.7 mole% propane gas mixture for varying water fraction in air (conditions 1-4, see also Table 2). From Fig. 4 it can be seen that the water content in air clearly affects the combustion phasing as demonstrated by the decrease in the peak combustion pressure with increasing water fraction. The observed reduction in (peak) pressure and, consequently, the temperature will affect the autoignition behavior of the end gas as will be shown below in the simulations.

Measured and simulated pressure traces for conditions 1-4 (see also Tables 2 and 3) are shown in Fig. 5. Apart from a small residual shift in the peak pressures along the crank angle axis [22], it can be seen in Fig. 5 that the predicted and measured pressure traces are in good agreement with the measured ones¹.

1. It should be noted that the autoignition computations using the experimental pressure trace show negligible differences with those using the predicted pressure trace for the gases studied [22, 23, 24].

Figure 4. Measured cylinder pressure profiles for methane/propane mixtures for varying water vapor content in air (see also Table 3).

Figure 5. Computed and measured cylinder pressure profiles and peak temperatures for conditions 1-4, see Table 2.

As can be seen from the Figure above, increasing the water fraction in air reduces the peak temperature from 923.5 K (4.3 g H₂O/dry air) to 913.9 K (11 g H₂O/dry air). This reduction in temperature is the main cause for the observed increase in knock resistance as shown in Figure 2.

Table 3. Measured- and computed peak pressures and temperatures at non knocking conditions for conditions 1-4 (see also Table 2).

	g H ₂ O / kg dry air	P _{meas} (bar)	P _{sim} (bar)	ΔP, (bar)	T _{sim @ peak(K)}
Condition 1	4.3	75.93	76.25	-0.32	923.50
Condition 2	6.8	75.13	75.83	-0.70	921.04
Condition 3	8.8	74.96	75.20	-0.24	916.54
Condition 4	11.0	74.63	74.73	-0.10	913.92

The two main contributions to the reduction in the in-cylinder conditions (P,T) upon increasing the water fraction in air are changes in mass burning rate (primarily the laminar burning velocity, see eq. 1 and 2, [34]) and thermophysical properties (primarily the heat capacity) of the fuel-air mixture. To examine the contribution of these effects separately we computed the pressure profiles for condition 4

for four different cases, presented in Table 4. In case 1 the pressure profile under actual experimental conditions is calculated (11 g H₂O/kg dry air, see Figures 5 and 6). Next, in case 2 the pressure profile for condition 4 is calculated by using dry air (0 g H₂O/kg dry air, see case 2 in Table 4). The peak pressure and temperature are substantially lower for case 1 due to the presence of water. This observed decrease in temperature and pressure are caused by a decrease in laminar burning velocity (29.3 cm/s to 26.7 cm/s at the ignition point) resulting in a decrease in mass burning rate (see eq. 1) and a slight increase in thermophysical properties (predominantly an increase in heat capacity (~1%). To illustrate the contribution of the changes in mass burning rate to the observed decrease in pressure and temperature, we performed simulations by using the thermophysical properties of dry air and mass burning rate for 11 g H₂O/dry air (case 3 in Table 4). The effect of thermophysical properties was illustrated in a similar way; see case 4 in Table 4.

Figure 6. Computed cylinder pressure profiles for the four cases in Table 4 (see text for details). In the legend the computed peak temperatures (T_{peak}) are presented

Table 4. Overview of the four cases studied (see text for details) and computed peak temperature, pressure and autoignition delay times.

	Case 1 (condition 4)	Case 2	Case 3	Case 4
Mass burning rate	11 g H ₂ O/dry air	0 g H ₂ O/dry air	11 g H ₂ O/dry air	0 g H ₂ O/dry air
Thermophysical properties (heat capacity is dominant factor)	11 g H ₂ O/dry air	0 g H ₂ O/dry air	0 g H ₂ O/dry air	11 g H ₂ O/dry air
Peak pressure (bar)	74.73	79.16	77.46	76.37
Peak temperature (K)	913.92	927.86	923.05	918.36
Autoignition delay time (ms)	5.2	4.4	4.7	4.9

Both the reduction in mass burning rate and the increase in heat capacity with increasing water fraction in air reduce the peak pressure and temperature as shown in Figure 6. The observed changes in peak pressure and temperature have a significant effect on the autoignition behavior of the end gas. To illustrate this, we rank the knock resistance for 4 cases in Table 4 based on the autoignition delay times calculated using the complete specific volume history derived from

the pressure histories given in Fig. 6 as input in the simulations, see Figure 7. As illustrated above to create knock we increase the intake temperature by 42 K for all four cases.

Figure 7. Calculated autoignition for cases 1-4 using the compute pressure histories as shown in Figure 6 (see Table 4 and text for details)

Autoignition calculations for the end gases presented in Figure 7 show the impact of water for cases 1-4. Case 2 shows the shortest autoignition delay time as a result of the highest peak temperature and pressure (see also Fig. 6), and thus most prone to knock. As discussed above, the addition of water results in a decrease of peak pressure and temperature, which increases the autoignition delay time as observed for case 1, 3 and 4. The mass burning rate and thermophysical properties both substantially contribute on the increase in autoignition delay time as observed in Fig. 7.

Since the effect of water on the autoignition delay time is negligible, as discussed above, we conclude that the observed increase in knock resistance upon increasing the water fraction in air is a direct result of changes combustion phasing, which means in the mass burning rate and thermophysical properties of the fuel-air mixture.

INCORPORATING HUMIDITY IN THE PROPANE BASED SCALE

To compare the knock resistance of the mixtures studied with different air humidity with each other, following a suggestion in Ref. [14], we use the PKI scale [22, 23, 24]. Here, the equivalent methane/propane mixtures are computed using dry air. In this procedure, which is illustrated in Figure 8, the occurrence of engine knock at the point at which 90% of the fuel is burned is calculated for a given fuel composition, in this example condition 2.

To find the “propane equivalent”, propane is added to methane and engine knock is calculated by the integrated knock model (simulation of both combustion phasing and autoignition) while keeping all other parameters constant. The model automatically changes the propane fraction in methane iteratively up to the moment that knock occurs at the same position in the cycle as the gas under comparison. The calculations shown as an illustration in Figure 8 indicate that the gas composition of condition 2 (RH 40.1%) is equivalent to 3.2%

propane. Thus, 40.1% humidity in air for a fuel with 4.7% propane is equivalent to reducing the propane fraction by 1.5%, a substantial increase in knock resistance.

Figure 8. Knock simulations for condition 2 (see Table 3) using the knock model.

COMPARISON WITH EXPERIMENTS

To assess the veracity of the model predictions, we calculated the Propane Knock Index (PKI) for the methane/propane mixtures with different water vapor content in air (see also conditions 1-4 in Tables 2-3) and correlate the results with the measured knock resistance (KLST), as plotted in Figure 9. For the purposes of comparison, the PKI computed for the methane/alkane compositions from previous work [22, 23, 24] are also shown. As can be seen in Figure 9, the deviations of the computed PKI from the fit of the methane-propane data are well within the uncertainty of the KLST measurements for all gases studied (± 0.75 °CA), demonstrating the excellent predictive power of the method. Here we observe that the measured effect of water vapor on the knock resistance is captured by the model exceptionally well. According to calculations, an increase in humidity from 4.3 to 11 g H₂O/kg dry air results in a drop in PKI from 3.8 to 2.4. Here we note that this is equivalent to an increase in methane number from 78 to 83 points when converting the propane based scale into a methane-hydrogen scale [22], with 100 having the knock equivalent of pure methane, and 0 being nominally assigned to hydrogen.

Figure 9. Computed knock resistance (PKI) versus measured knock resistance (KLST).

CONCLUSIONS

The impact of humidity on the knock resistance is experimentally investigated in a lean burn ($\lambda=1.5$), fixed-speed, medium BMEP gas test engine. The results show that increasing the water content in air results in a substantial increase in the knock resistance of the fuel/air mixture: at 4.3 g H₂O/kg dry air the KLST is 19 °CA BTDC while at 11 g H₂O/kg dry air the KLST is 21 °CA BTDC for the same fuel.

To analyze the origins of the effect of water content in air on the knock resistance the effects of humidity on phasing and autoignition are incorporated in the knock model. Analyses of computations using the knock model show that the effect of increasing the water content in air on the autoignition delay time at constant pressure and temperature conditions is negligible. Furthermore, the computations and measurements at non-knocking conditions indicate that the observed changes in thermophysical properties of the fuel-air mixture (predominantly heat capacity) and mass burning rate (predominantly the laminar burning velocity) with increasing water content reduces the end-gas temperature and pressure substantially. These effects in turn reduce the autoignition delay time substantially, which is the origin of the experimentally observed increase in the knock resistance upon increasing the water content in air.

Comparison between the calculated knock resistance (PKI) and the measured knock resistance (KLST) shows excellent agreement for the experimental variations in water content in air studied; moreover, the effect of varying water content in air on the pressure profile is very well captured by the updated knock model. The results show that the PKI drops from 3.8 to 2.4 when the water content in air increase from 4.3 to 11 g H₂O/kg dry air, which is equivalent to an increase in the methane number of 5 points.

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DEFINITIONS/ABBREVIATIONS

α - Parameter describing temperature dependence of laminar burning velocity

β - Parameter describing pressure dependence of laminar burning velocity.

λ - Excess air ratio (actual air/fuel ratio divided by the stoichiometric air/fuel ratio)

ρ_b - Burned gas density

AVL - Anstalt für Verbrennungskraftmaschinen List

BMEP - Brake mean effective pressure

BTDC - Before top dead centre

CHP - Combined heat and power

C_{model} - Fit parameter

dm_b/dt - Mass burning rate

KLST - Knock-limited spark timing

Le_{eff} - Effective Lewis number

LNG - Liquefied natural gas

OEM - Original equipment manufacturer

P - Pressure

PKI - Propane Knock Index

RCM - Rapid compression machine

S_L - Laminar burning velocity

$S_{L,0}$ - Laminar burning velocity of the fuel/air mixture at ignition conditions

t - Time

T - Temperature

x_b - Burned gas mass fraction