

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.