

The Effect of Cooled Exhaust Gas Recirculation for a Naturally Aspirated Stationary Gas Engine

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ABSTRACT

Small natural gas cogeneration engines frequently operate with lean mixture and late ignition timing to comply with NO_x emission standards. Late combustion phasing is the consequence, leading to significant losses in engine efficiency. When substituting a part of the excess air with exhaust gas, heat capacity increases, thus reducing NO_x emissions. Combustion phasing can be advanced, resulting in a thermodynamically more favourable heat release without increasing NO_x but improving engine efficiency.

In this work, the effect of replacing a part of excess air with exhaust gas was investigated first in a constant volume combustion chamber. It enabled to analyse the influence of the exhaust gas under motionless initial conditions for several relative air-fuel ratios ($\lambda = 1.3$ to 1.7). Starting from the initial value of λ , the amount of CH_4 was maintained constant as a part of the excess air was replaced by exhaust gas. Temperature of burnt and unburnt mixture, combustion velocity and others were analysed for varying EGR rates.

In the next step, cooled exhaust gas recirculation was employed for a single-cylinder cogeneration engine to evaluate the influence of EGR rates up to 25 % on engine performance and emissions. For the baseline compression ratio of 13.2, a slightly higher IMEP could be achieved coupled with lower ISFC (almost -5 g/kWh) at constant NO_x emissions. Subsequently, the compression ratio was raised to 15.25 and the trials repeated, showing further improvement with the drawback of higher knock tendency.

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INTRODUCTION

The simultaneous use of heat and power allows cogeneration units (or also known as combined heat and power units) to achieve overall efficiencies of over 90 %. Therefore, it represents an attractive approach to reduce greenhouse gas emissions according to the defined target of the European Union (EU). The aim of 80-95 % CO_2 reduction by 2050 relative to 1990 levels was reinforced by publishing the low-carbon roadmap in 2011 [1]. In Denmark, Finland, Russia and Latvia 30-50 % of the generated power was achieved through cogeneration units in 2008. A substantial growth is also to be expected, among others, in USA, Japan, Russia and Brazil [2].

The systems are mostly equipped with natural gas (NG) engines, providing high thermal efficiency and long-term durability. The emission legislations differ worldwide significantly (even within certain countries such as Switzerland or Japan), depending on generated electrical power or thermal energy input. In Germany, cogeneration manufacturers of the small power range orient

themselves to emission standards according to TA-Luft [3]. It obliges large gas engines with a thermal input of 1 MW to meet dry NO_x emissions as low as 500 mg/mN³ at 5 % O_2 (approx. 200 ppm). The significant rise of installed systems over the past years leads to progressively stringent emissions standards. In 2018, a Europe-wide tightening of emissions standards will take place, applying for NG driven engines with a generated electrical power of $P_{el} = 50$ kW and less [4]. The NO_x limit of 240 mg/kWh dry at 0 % O_2 is related to thermal input (higher heating value HHV), amounting to roughly half of the present value according to TA-Luft (i.e. approx. 100 ppm).

For the small power sector of 50 kW_{el} and less, naturally aspirated gas engines conventionally operate unthrottled with external mixture formation with either lean or stoichiometric air-fuel mixture. The latter requires a three-way-catalyst (TWC) in order to reduce tail pipe emissions such as hydrocarbons (HC), carbon monoxide (CO) and nitrogen oxide (NO_x). In lean burn operation, on the other hand, HC and CO emissions are converted through an oxidation catalyst, while

low NO_x emissions result from low combustion temperature. This is not only ensured through a lean mixture but also by retarded ignition timing, leading to a thermodynamically unfavourable centre of combustion (CA50). Since combustion temperature and thus heat wall losses drop with leaning, the engine still reaches relatively high thermal efficiency, in spite of operation with relative air-fuel ratios λ beyond the efficiency optimum.

Figure 1 shows the trade-off between engine efficiency, NO_x emissions and IMEP for a naturally aspirated engine. Further mixture dilution allows the engine to meet future emission standards, however, combustion progressively deviates from ideal Otto cycle and cycle-by-cycle variations increase, leading to lower thermal efficiency. Leaning also reduces fuel fraction for naturally aspirated engines, resulting in lower IMEP and therefore in a higher impact of friction, reducing engine's brake efficiency.

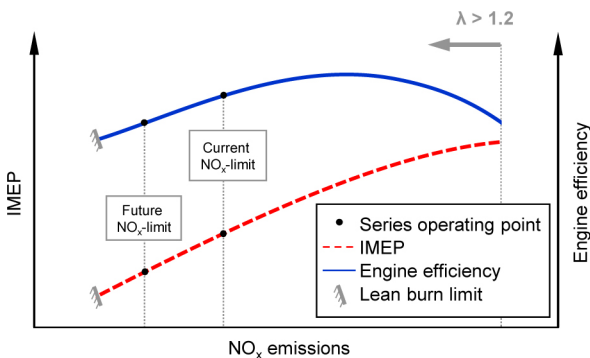


Figure 1. Trade-off between engine efficiency, NO_x emissions and IMEP for a naturally aspirated cogeneration engine schematically

This paper provides an analysis of an alternative mixture configuration for small-scaled cogeneration engines with the aim of improving the previously described trade-off. Exhaust gas is recirculated into the intake section to replace merely a part of the excess air, increasing mixture's heat capacity. As a result, engine efficiency can be improved by advancing combustion phasing at equal NO_x emissions.

The paper begins with the fundamentals of exhaust gas recirculation, addressing the definition of lean burn and revealing the influence of EGR on the relative air-fuel ratio. The results of 0D engine calculations suggest the theoretical potential of EGR for lean burn operation. Subsequently, experiments of different mixtures compositions using a constant volume combustion chamber are presented in order to show the effect of mixture's composition on combustion, in particular on its burning velocity, under motionless conditions. The next chapter covers the experimental setup used for engine trials, followed by experimental results for the baseline but also for an increased compression ratio. Eventually, a brief summary is given.

EXHAUST GAS RECIRCULATION

EGR represents an established means to dilute air-fuel mixture. It is widely used whether for Otto engines to lower pumping losses in part load operation or for Diesel engines to reduce combustion temperature and thus to counter NO_x emissions. The formation of

NO_x is inhibited as a result of cooling the combustion through so-called mass and caloric effect [5, 6]. The first describes the increasing stoichiometric mass of the non-premixed mixture that combusts at approximately $\lambda = 1$, while the caloric effect arises from the higher specific heat capacity due to the high H_2O and CO_2 amount of the exhaust gas [5, 6] (see Figure 2).

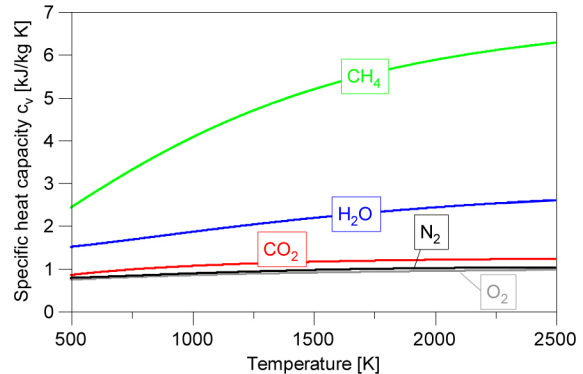


Figure 2. Specific heat capacity of methane (CH_4), nitrogen (N_2), oxygen (O_2), carbon dioxide (CO_2) and water (H_2O)

In contrary to Diesel, natural gas engines emit practically no soot [7] that could contaminate components such as EGR valve or cooler and appears therefore to be suitable for long-term operation.

Work of Others

Numerous comparisons between both operation modes pure lean burn and stoichiometric with EGR can be found in literature [8, 9, 10]. In contrast to the mentioned work, the subject of this paper is to combine lean burn operation and EGR as (only) a part of the excess air is substituted with exhaust gas. To date, this approach was primarily investigated in the course of research projects [5, 6, 7, 10, 11, 12, 13, 14, 15, 16]. Numerical work points out potential in reducing NO_x emissions, while efficiency behaviour depends on engine size and the type of EGR strategy. Lee et al. [11] investigated a dedicated EGR system that reduces fuel consumption of small 4-6 cylinder engines with 338 $\text{cm}^3/\text{cylinder}$, however, the dedicated EGR cylinder was operated with rich air-fuel mixture while the other cylinders operated with lean ($\lambda < 1.3$) or stoichiometric mixture, respectively. Wimmer [5] and Pizzirani et al. [12] studied the effect of EGR for large lean-burn gas engines, concluding that NO_x emissions significantly fall at the slight expense of engine efficiency. Since knocking tendency also decreases, Pizzirani et al. see potential in increasing efficiency by augmenting compression ratio.

Wimmer also studied the effect of EGR experimentally for a single-cylinder research engine (6.2 litres) in stoichiometric and lean burn operation [6]. Lean burn operation coupled with moderate EGR rates (15 %) reduces NO_x emissions notably while efficiency suffers only slightly due to imperfect combustion and altered charge composition. Tschalamoff [13, 13, 14] employed EGR (EGR rates up to 30 %) for a single-cylinder 4.83 litre engine with scavenged and unscavenged prechamber spark plug. NO_x emissions and mean effective pressure (MEP) were maintained constant by adjusting fuel

quantity and ignition timing, respectively. For an unscavenged spark plug, engine efficiency remains unchanged, whereas it increases in the case of scavenged prechamber spark plug by 1.5 %-points. The benefit was found in thermodynamically more favourable CA50 at same NO_x emissions affected by higher specific heat capacity of the mixture. Since temperature is also one of the driving parameters for knocking combustion, EGR extends the knock margin for both lean [5, 13, 14, 17] and stoichiometric operation [6, 7, 9, 17]. This allows the engine to operate at higher compression ratio and/or higher IMEP.

Definition of Mixture Dilution and EGR Rate

For the later course of this paper, the parameters to describe mixture composition are of vital importance. The amount of exhaust gas is expressed by the EGR rate x_{EGR} , determined by dividing the volume flow of the recirculated exhaust gas $\dot{V}_{\text{EGR}}$ with the aspirated mixture, consisting of the volume flow of exhaust gas $\dot{V}_{\text{EGR}}$, air $\dot{V}_{\text{air}}$ and fuel $\dot{V}_{\text{fuel}}$:

$$x_{\text{EGR}} = \frac{\dot{V}_{\text{EGR}}}{\dot{V}_{\text{EGR}} + \dot{V}_{\text{fuel}} + \dot{V}_{\text{air}}} \quad (1)$$

The relative air-fuel ratio λ or its inverse, the fuel-air equivalence ratio ϕ , are commonly used parameters for defining mixture composition. These are formulated by the ratio of moles of oxygen (O_2) of the in-cylinder charge n_{O_2} and the required moles of O_2 for a stoichiometric combustion $n_{\text{O}_2 \text{ stoich}}$ or vice versa:

$$\lambda = \phi^{-1} = \frac{n_{\text{O}_2}}{n_{\text{O}_2 \text{ stoich}}} \quad (2)$$

Stoichiometric combustion takes place at $\lambda = 1.0$, while $\lambda > 1.0$ is defined as lean combustion. When substituting a part of excess air of a lean mixture with exhaust gas, the amount of fuel remains unchanged, leading to a lower relative air-fuel ratio thanks to the smaller O_2 fraction of the exhaust gas compared with air. For better understanding of the dependency between EGR rate and relative air-fuel ratio, the dilution ratio D is introduced

$$D = \frac{n_{\text{air}} + n_{\text{EGR}}}{n_{\text{air stoich}}} \quad (3)$$

where n_{air} and n_{EGR} are the number of moles of air and EGR of the in-cylinder charge and $n_{\text{air stoich}}$ is the number of moles of air that results in a stoichiometric combustion.

Figure 3 (left) shows the influence of D and x_{EGR} on the relative air-fuel ratio quantitatively, while Figure 3 (right) illustrates a mixture with a dilution ratio of $D = 1.40$ over an EGR rate from 0 to 20 % schematically. In the case of pure mixture dilution through air, the relative air-fuel ratio matches with the dilution ratio $\lambda = D = 1.40$. With rising EGR fraction but constant dilution ratio $D = 1.4$, the amount of O_2 decreases, leading to lower air-fuel ratios of $\lambda = 1.28$ and $\lambda = 1.12$ for an EGR rate of 10 and 20 %, respectively.

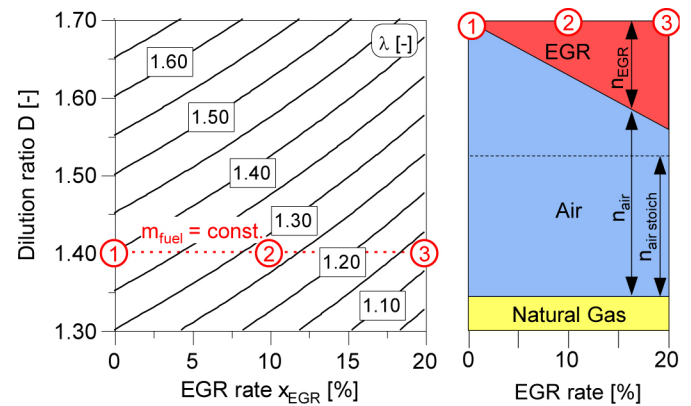


Figure 3. Relative air-fuel ratio of the combustion λ as function of the dilution ratio and EGR rate (left) and influence of the EGR rate on mixture's mole fraction at constant dilution ratio schematically (right)

0D ENGINE CALCULATION

For the ideal engine cycle, compression ratio and mixture's composition determine engine efficiency. Therefore, the effect of dilution ratio and EGR rate on the trade-off between engine efficiency, IMEP and NO_x emissions was studied using a 0D engine calculation model (created with AVL BOOST). It considers the cylinder as adiabatic system with perfect in-cylinder filling (i.e. no residual gas) and assumes isochoric combustion and gas exchange. The computation considers the temperature-driven change of thermochemical properties, while the initial temperature and pressure of the air-methane-exhaust gas mixture are defined to $T_0 = 293.15 \text{ K}$ and $p_0 = 1.0 \text{ bar}$. The water of the exhaust gas is assumed to have a gaseous phase at initial conditions, which is only a minor error due to the rapid temperature rise as compression takes place.

Figure 4 shows thermal efficiency, peak combustion temperature and normalised indicated work over compression ratio ($\text{CR} = 10$ to 18) for several mixture settings. Due to the high temperature dependency of NO_x emissions, peak combustion temperature represents their indicator. As expected, peak combustion temperature, indicated work and thermal efficiency increase with compression ratio. An engine setup characterised by a dilution ratio of $D = \lambda = 1.50$ without EGR and a compression ratio of $\text{CR} = 13.0$ is a commonly used setting for a series operating point of small gas engines, thus serving as a reference for comparison. When replacing a part of the excess air of the reference configuration with exhaust gas ($D = 1.50/x_{\text{EGR}} = 20 \%$), peak combustion temperature decreases in spite of same quantity of fuel added to the cycle. This arises from the higher specific heat capacity of the in-cylinder charge, absorbing a larger amount of heat as combustion takes place. As a result, peak combustion pressure drops, reducing cycle work and efficiency. A rise of the compression ratio ($\text{CR} = 13.50$) compensates losses in efficiency, while indicated work is still below the reference operating point. This is contributable to the smaller amount of fuel added to the cycle as compression ratio increases and therefore cylinder volume at BDC decreases. The cycle outputs the same amount of work when further enhancing compression ratio up to $\text{CR} = 13.95$. Engine efficiency increases by 0.4 %-points while peak combustion temperature remains significantly lower.

A higher dilution ratio of $D = \lambda = 1.60$ (i.e. $x_{\text{EGR}} = 0\%$) necessitates a compression ratio of 17.80 to achieve equivalent indicated work as the reference due to lower amount of fuel. Peak combustion temperature lies slightly above the one of $D = 1.50/x_{\text{EGR}} = 20\%$ at $\text{CR} = 13.95$, whereas the benefit in efficiency is significantly higher. A lower dilution ratio ($D = 1.40$) without EGR results in peak combustion temperature higher than the value of the reference operating point. It decreases significantly by adding EGR ($x_{\text{EGR}} = 20\%$), however, the reference temperature is only achieved by also reducing the compression ratio and thus deteriorating thermal efficiency notably.

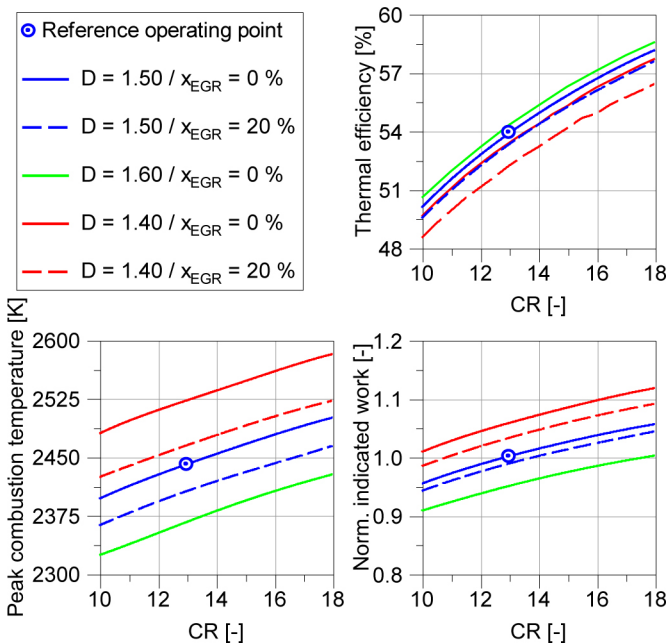


Figure 4. Thermal efficiency, peak combustion temperature and normalised indicated work per cycle over compression ratio for several mixture compositions (0D-Simulation)

In both cases $D = 1.60/x_{\text{EGR}} = 0\%$ and $D = 1.50/x_{\text{EGR}} = 20\%$ high compression ratios ($\text{CR} = 17.8$ and 13.95 respectively) are calculated to maintain indicated work of the reference operating point. Due to the high knock resistance of natural gas as well as the knock-reducing effect of EGR known from literature [5, 13, 14, 17], a compression ratio of $\text{CR} = 13.95$ appears to be a feasible value with respect to real engine operation (without knocking). A configuration of leaner mixture and even higher compression ratio ($D = 1.60/x_{\text{EGR}} = 0\%$ at $\text{CR} = 17.80$), on the other hand, results in similar combustion temperatures but would increase the chance of knocking in real engine operation significantly. This arises from its dependence on pressure and temperature history as well as on combustion speed, which however were neglected in this study.

EXPERIMENTS WITH A CONSTANT VOLUME COMBUSTION CHAMBER

The laminar burning velocity of a premixed air-fuel-exhaust gas mixture highly influences the heat release during the cycle and therefore engine efficiency and its emissions. It only depends on composition, temperature and pressure of the mixture. It is defined as

the velocity, with which unburnt gas of the premixed mixture propagates through the flame front and is transformed to products under laminar flow conditions [18, 19]. To determine the laminar burning velocity of various mixture compositions, experiments using a constant volume combustion chamber (CVCC) were carried out at the University of Valladolid, Spain.

Experimental Setup and Numerical Analysis

The CVCC is made of stainless steel and has a spherical shape with an inner diameter of 200 mm, corresponding to a 4.19 litre volume. It is connected to several gas bottles as well as an air compressor on the inlet and a vacuum pump on the outlet side (see Figure 5). To measure the initial conditions of pressure and temperature, four piezoresistive sensors are used and a thermocouple is located in the CVCC. The CVCC is thoroughly insulated and equipped with an electrical heater on the bottom, enabling initial mixture temperatures up to 650 K. Two prolonged electrodes form the spark gap in the centre of the CVCC, leading to a uniform flame propagation over the vessels radius. The pressure signal over time is measured by the means of a Kistler 7063A pressure transducer that is connected to an oscilloscope for recording the data.

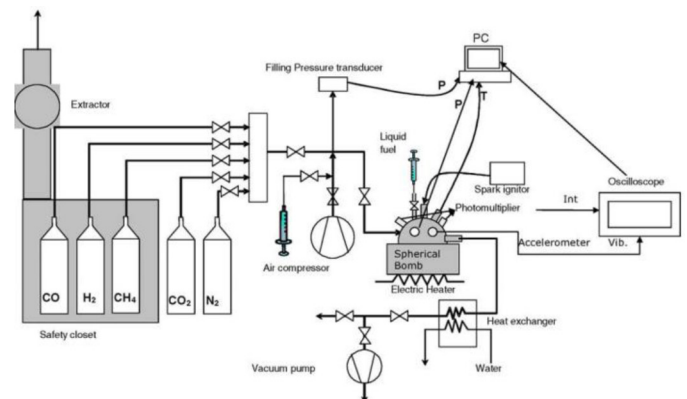


Figure 5. Schematic layout of constant volume combustion chamber [20]

The data is evaluated using a proprietary two-zone combustion analysis model, dividing the gas within the CVCC into burnt and unburnt fraction. The model considers exhaust gas of the initial mixture as products of perfect combustion CO_2 , N_2 , O_2 and H_2O , whereas for combustion itself it solves 12 equilibrium equations of the major species. The thermochemical properties taken from NIST-JANAF tables consider temperature dependency of the specific heat capacity. The calculation model assumes following behaviour:

- Combustion commences in the centre of the vessel, leading to a flame front that propagates spherically with negligible thickness.
- The pressure during combustion is uniform throughout the whole vessel.
- The CVCC is an adiabatic system, meaning that no heat is lost through the walls.
- Ideal gas behaviour is assumed.
- Unburnt gas is isentropically compressed as combustion proceeds.

The first law of thermodynamics for a non-stationary open system is applied to burnt and unburnt zone, describing their change of internal energy. After adding mass conservation and volume equation, the laminar burning velocity C_c yields to Eq.(4)

$$C_c = \frac{dm_b/dt}{A_f \cdot \rho_u} \quad (4)$$

where the mass burning rate dm_b/dt is achieved from pressure data, the flame area A_f is determined over the burnt volume by assuming spherical flame shape and ρ_u is the density of the unburnt zone. For the derivation of Eq.(4) as well as a more detailed description of the pressure analysis model and the experimental setup, the authors refer to former work [18, 20, 21, 22]

Testing Procedure

All experiments were conducted at same initial temperature of $T_0 = 423.15$ K and initial pressure of $p_0 = 3.73$ bar, resulting in a constant amount of moles. The initial conditions were determined by isentropic change of state starting from 293.15 K and 1.0 bar. The dilution ratio was varied from $D = 1.3$ to $D = 1.8$ in increments of 0.1. For each dilution ratio, a part of the excess air was substituted with exhaust gas, starting from an EGR rate of $x_{EGR} = 5\%$ up to $x_{EGR} = 20\%$.

After every trial, the CCVC was scavenged for a duration of 5 minutes using compressed air to avoid any remaining combustion products. The vessel was then evacuated and refilled first with CH_4 and after with compressed air. The corresponding partial pressures of each mixture were calculated by assuming ideal gas behaviour. After a waiting time of 5 minutes, the mixture was ignited. Prior investigations proved this to be a sufficient duration to achieve well-mixed and motionless mixture [18, 19].

Experiments including EGR necessitated an additional working step. In order to obtain exhaust gas components such as H_2O and CO_2 within the vessel, first a stoichiometric combustion was conducted. After a waiting time of 3 minutes the desired partial pressure to obtain the required CO_2 and H_2O fraction from the former combustion was adjusted using the vacuum pump. Subsequently, CH_4 and air were added, including the missing excess part of N_2 and O_2 of the exhaust gas after the stoichiometric combustion.

Results of Varying Mixture Composition

Figure 6 shows pressure versus time for all investigated dilution ratios without exhaust gas (left) ($D = \lambda$) and for a dilution ratio of $D = 1.4$ of varying EGR rates (right) at constant ignition timing $t_{ignition} = 0.00$ s. Under the premise of constant initial conditions, the amount of fuel falls for higher dilution ratios. Peak and gradient of the combustion pressure decrease, leading to retarded peak combustion pressure. A similar behaviour is observed when exhaust gas is added for constant dilution ratio. Specific heat capacity increases for higher EGR rates, resulting in lower pressure rise with a lower pressure peak and retarded combustion.

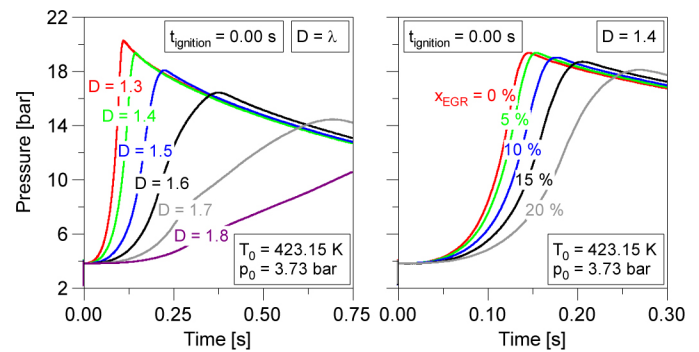


Figure 6. Pressure over time for different dilution ratios D without EGR (left) and for dilution ratio of $D = 1.4$ of varying EGR rates (right)

Figure 7 shows the calculated burning velocity C_c against the temperature of the unburnt zone T_u of selected mixture compositions. Note that same T_u corresponds to very similar pressure thanks to the isentropic calculation of T_u . The first part of the plotted traces between the initial temperature and approximately 475 K is futile with respect to evaluate mixture's burning velocity due to instability of the values. Burning velocity decreases with higher dilution ratio and higher EGR rate significantly. The lower peak in combustion pressure as a result of leaning or adding EGR, respectively, correlates with the maximum temperature of the unburnt zone, thus narrowing the evaluation window of burning velocity. Therefore, only dilution ratios of $D = 1.3$ to 1.5 with and $D = 1.6$ without EGR are plotted in Figure 7. After burning velocity reaches its maximum, it declines rapidly until combustion is completed. The gradient of this section decreases with both higher dilution ratio and EGR rate, enhancing the time that the flame front dwells in the vicinity of the wall and thus promoting the chance of flame quenching.

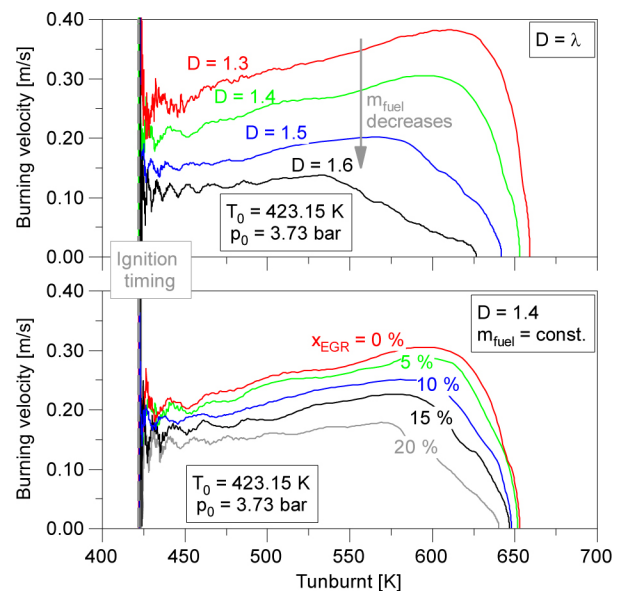


Figure 7. Burning velocity over temperature of unburnt zone for different dilution ratios (top) and for a constant dilution ratio of $D = 1.4$ of varying EGR rates (bottom)

Figure 8 (left) depicts mixture's initial specific heat capacity c_v (before ignition) against the EGR rate for various dilution ratios D . For a constant D , i.e. constant m_{fuel} , the specific heat capacity