

# Experimental Study on the Burning Rate of Methane and PRF95 Dual Fuels

Sotiris Petrakides  
Loughborough University

Dongzhi Gao  
SKLE - Tianjin University

Rui Chen and Daniel Butcher  
Loughborough University

Haiqiao Wei  
SKLE - Tianjin University

## ABSTRACT

Natural gas as an alternative fuel offers the potential of clean combustion and emits relatively low CO<sub>2</sub> emissions. The main constituent of natural gas is methane. Historically, the slow burning speed of methane has been a major concern for automotive applications.

Literature on experimental methane-gasoline Dual Fuel (DF) studies on research engines showed that the DF strategy is improving methane combustion, leading to an enhanced initial establishment of burning speed even compared to that of gasoline. The mechanism of such an effect remains unclear.

In the present study, pure methane (representing natural gas) and PRF95 (representing gasoline) were supplied to a constant volume combustion vessel to produce a DF air mixture. Methane was added to PRF95 in three different energy ratios 25%, 50% and 75%. Experiments have been conducted at equivalence ratios of 0.8, 1, 1.2, initial pressures of 2.5, 5 and 10 bar and a temperature of 373K. At stoichiometric conditions, experiments in an SI engine have been also performed.

It has been found that methane and all DFs have their fastest burning rate at stoichiometric conditions whereas PRF95 at rich conditions ( $\Phi=1.2$ ). At lean conditions ( $\Phi=0.8$ ), all DFs resulted in faster combustion than PRF95, whereas methane is the slowest of all. At rich conditions, DF75 and DF50 are slower than methane. The transition mechanism between the constant volume combustion experiments and those in the engine environment resulted in a larger increase in the burning speed of methane and all DFs in comparison to that of the liquid fuel.

**CITATION:** Petrakides, S., Gao, D., Chen, R., Butcher, D. et al., "Experimental Study on the Burning Rate of Methane and PRF95 Dual Fuels," *SAE Int. J. Engines* 9(2):2016, doi:10.4271/2016-01-0804.

## INTRODUCTION

There is a growing interest for energy efficient transport solutions. Environmental impact of greenhouse gas emissions, as well as future air quality, are forcing governmental bodies to continuously update their legislations, adopting challenging emission standards. Alternative fuels have a central contribution towards compliance with future emission legislations.

Methane has been classified by a number of studies as one of the most promising alternative fuels [1, 2, 3, 4, 5, 6], mainly due to its low carbon content and its knocking resistance. However, the slow

burning speed of methane has been a major concern for its use in SI engines [7, 8, 9]. The slow burning speed degrades the thermal efficiency of an SI engine and it is a contributor for hydrocarbon emissions. The impact of methane hydrocarbon emissions on the greenhouse effect is higher than the impact of CO<sub>2</sub>. As emphasized in literature on experimental studies [10-11], methane-gasoline dual fuelling improves methane combustion, leading to enhanced initial establishment of burning speed even in comparison to gasoline. The mechanism behind the observed phenomena has not been perfectly understood.

In contrast to a methane-gasoline Dual Fuel (DF) blend, substantial efforts have been made to improve the understanding of methane combustion as well as gasoline and its surrogates both in Constant Volume Combustion Vessels (CVCVs) [12, 13, 14, 15, 16, 17, 18, 19] and SI internal combustion engines [20, 21, 22, 23, 24].

Experiments in CVCVs used throughout literature to evaluate the fundamental combustion characteristics of a combustible mixture such as laminar flame speed and Markstein length. The value of Markstein length characterizes the response of the flame speed to the flame stretch. The flame stretch is induced by the curvature of the flame as well as from the nonuniformity and unsteadiness of the flow around the flame. For a positive Markstein length, the flame speed will decrease under stretching and vice versa. The reported laminar flame speed of methane is lower compared to that of gasoline and its surrogates when tested at similar conditions [15, 13, 17, 19]. A part of the study of Gu et al. [17] compared the laminar flame speed and Markstein length of isooctane and methane air mixtures. At 5 bar and 358 K the laminar flame speed of methane found to be 18% lower than that of isooctane. As emphasized [17], these two fuels responded to flame stretch differently with respect to the equivalence ratio. For methane-air mixtures, Markstein length increases with equivalence ratio whereas it decreases for isooctane-air mixtures. At lean mixtures the Markstein length of methane is lower than that of isooctane.

As far as experiments in SI engines are concerned, the findings with respect to the effects of different fuelling modes as well as operating parameters on the overall burning rate usually differ among the studies in literature. Aleiferis et al. [21] reported that at low-load conditions gasoline DI mode increase the rate of heat release as compared to PFI mode. These findings are not similar to those of Zhu et al [22]. Aleiferis et al [23] emphasizing the difficulties in drawing general conclusions from comparisons of experimental data conducted in different internal combustion engines in literature, suggesting that changes in hardware and operating strategies can substantially change the outcomes of a test.

## BURNING RATE DEFINITIONS

The mass burning rate in a homogeneous SI engine can be defined as,

$$\dot{m}_{b,engine} = \rho_u A_f S_L I_0 \quad (1)$$

where  $\rho_u$  is the unburned gas density,  $A_f$  is the actual flame surface area,  $S_L$  is the laminar flame speed, and  $I_0$  is the flame stretch factor.

It can be derived [as shown in Appendix A1] that the mass burning rate is depended on the in-cylinder pressure,  $P$ , unburned gas temperature,  $T_u$ , equivalence ratio,  $\Phi$ , composition,  $c_p$ , and turbulence characteristics,  $u_p$ , as shown in equation 2. The dependence of burning rate on the mentioned parameters is highly nonlinear.

$$\dot{m}_{b,engine} = f(P, T_u, \Phi, c_p, u_t) \quad (2)$$

The parameters that affect the burning rate (eq. 2) are also dependent variables. For example, a change in composition will affect the unburned gas temperature. To help the reader appreciate the mentioned phenomena, a simplified engine compression stroke is modeled with an adiabatic variable volume model, and a compression ratio of 10:1. Three different fuels have been considered at stoichiometric conditions. The fuels consist of  $CH_4$ , a binary blend of primary reference fuels commonly known as PRF95 (95%volliq of iso-octane and 5%volliq n-heptane), and a DF blend (DF50) with half of its energy contributing from  $CH_4$  and half from PRF95.

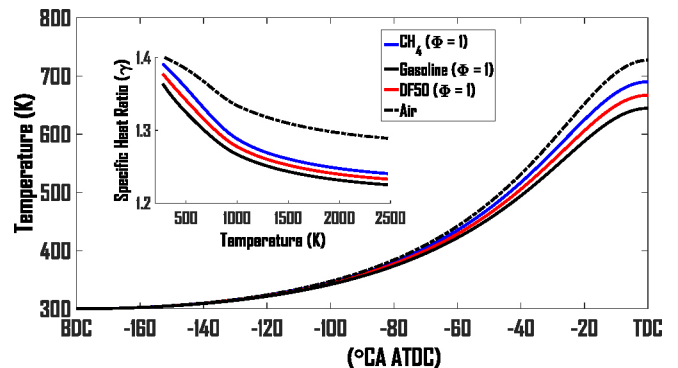


Figure 1. Effect of piston compression on the unburned gas temperature during different fuelling modes

As can be observed from Figure 1 at identical initial conditions and for the same compression, a change in fuelling mode will lead to different compressive temperatures attributed to the different specific heat ratios among fuels. For completeness, the compressive temperature of pure air is also shown in Figure 1. Such analysis illustrates the difficulties in drawing general conclusions from engine experimental data with regards to the burning characteristics of different fuels. That highlights the need for experiments under the same initial conditions in order to draw more confident conclusions when comparing the effects of a variety of fuels.

Experiments in CVCVs allow for the same initial conditions among the different tested fuels due to the nonexistence of piston compression and the minimization of turbulent flow structures. The flow structures inside the vessel can be minimized by giving time for quiescence after the filling process has been completed. Therefore the effect of turbulence on the burning rate can be reasonably neglected. Moreover, the time of quiescence enhances the mixing of air and fuel, limiting any spatial differences of the air to fuel ratio inside the vessel as can be the case in SI engines. Flame instabilities manifested as cellular structures on the flame surface can lead to a self-turbulizing effect inside the CVCV [17]. The phenomenon of self-turbulizing will enhance the burning rate due to an increase of the flame's surface area. Flame instabilities can also occur in an engine environment. However in an engine, the stronger effects of turbulence on the burning rate are masking the ones of instability.

The mass burning rate inside the CVCV can be defined as,

$$\dot{m}_{b,CVCV} = \rho_u A'_f S'_L I'_0$$

It can be derived [Appendix A2] that the mass burning rate is dependent on the pressure, unburned gas temperature, equivalence ratio and composition.

$$\dot{m}_{b,CVCV} = f(P, T_u, \Phi, c_i) \quad (4)$$

At the time of ignition, three out of four parameters that affect the burning rate inside the CVCV as shown in equation 4 can be constrained, and the effects of the fourth parameter on the overall burning rate can be evaluated with confidence.

For a homogeneous mixture of the same equivalence ratio and composition, the transition mechanism characterizing the difference in burning rate from the constant volume combustion experiments to those in the engine environment can be revealed by considering the equations 2 and 4. For a particular fuelling mode, the contributors of the transition mechanism as indicated in equation 5 are the effects of turbulence on the burning rate as well as the difference in temperature and pressure due to piston compression.

$$\dot{m}_{b,CVCV} - \dot{m}_{b,engine} = f[(P_{CVCV} - P_{engine}), (T_{CVCV} - T_{engine}), u_t] \quad (5)$$

Where  $P_{CVCV}$ ,  $T_{CVCV}$  is the pressure and temperature inside the CVCV and  $P_{engine}$ ,  $T_{engine}$  is the pressure and temperature in the engine environment.

Following the previous discussion, it is evident that for a confident comparison of the burning characteristics of different fuels, there is a clear need for combustion experiments under the same initial conditions. In contrast to methane as well as gasoline and its surrogates, the burning rate of methane-gasoline dual fuel blends has not been adequately investigated. It is therefore the aim of this study, to experimentally investigate how a change in DF ratio is affecting the burning rate under the same initial conditions. In the current experimental work PRF95 has been used as the liquid fuel. Methane has been used as the gaseous fuel. The duration of 10% and 10-90% Mass Fraction Burned (MFB) are measured in a CVCV at initial pressures of 2.5, 5, 10 bar (abs), equivalence ratios of 0.8, 1, 1.2, and a temperature of 373 K. The particular pressure conditions have been selected to reflect idling, part and full load conditions in a heavily turbocharged engine at the point of spark. The particular temperature has been selected to allow reference with the available combustion characteristics of methane and PRF95 that have been already published in literature. The same MFB durations have been measured in an SI engine at stoichiometric conditions. The transition mechanism characterizing the difference in the response of MFB with DF ratio between constant volume combustion experiments and those in the engine environment is also investigated.

## METHODOLOGY

### Constant Volume Combustion Vessel

A 100 mm inner diameter cylindrical combustion vessel with a volume of 2.2 L was employed for the experimental CVCV study. The experimental setup is illustrated in Figure 2. The entire vessel was uniformly preheated by a set of electrical heating elements totaling 3.2 kW. One of the heaters was fully inserted inside the vessel to induce a transient temperature difference only during the filling process. The temperature difference evoked natural convection to stir the mixture enhancing the mixing of fuel and air. Similar technique has been used in [13]. The interior air temperature was controlled within 3 K using a closed-loop feedback controller set to 373K. The temperature could also be observed manually from a second temperature sensor mounted on the top of the vessel. The pressure rise during the combustion process was recorded using a Kistler 6113B pressure transducer at 6 kHz. The mixture was ignited using a slightly modified standard ignition plug with extended electrodes. The ignition system generated a spark with duration of 0.7 ms and the timing was synchronized with the interior pressure rise recording. For safety reasons, a 6 MPa pressure release valve was installed on the combustion vessel.

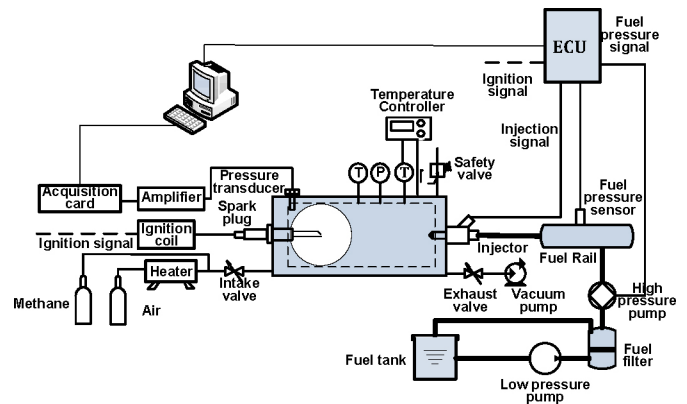


Figure 2. Schematic diagram of the CVCV setup

### Dual Fuel Mixture Preparation

The dual fuel blends consist of Methane and PRF95 in three different energy ratios (25, 50, 75), that have been defined based on equation 6. A blend with 25% of its energy contributing from methane was labelled as DF25, with 50% DF50, and for 75% DF75.

$$DF_{Ratio} = \frac{M_{CH_4} \times LHV_{CH_4}}{M_{PRF95} \times LHV_{PRF95} + M_{CH_4} \times LHV_{CH_4}} \quad (6)$$

Three different equivalence ratios have been tested (0.8, 1, 1.2). The different equivalence ratios have been calculated using the method of chemical balance and assuming products of complete combustion. High purity technical air was used with an oxidizer concentration  $[O_2/(O_2+N_2)]$  of  $0.2 \pm 0.01$ .

In every experimental condition, the equivalence ratio was prepared inside the vessel using the partial pressure method. Initially the vessel was heated up to the desired temperature (373 K). The necessary

amount of liquid and gaseous fuel was determined depending on the targeted dual fuel ratio, equivalence ratio, and the associate pressure. The liquid fuel was injected into the combustion vessel using a multi-hole gasoline direct injector with an injection pressure of 12 MPa. The injected fuel mass was controlled by individual injections using pre-calibrated data. The pre-calibration process involves the determination of the mass of liquid per single injection. After the injections were completed, two minutes was given to allow for complete evaporation of the liquid fuel. Considering the correct increase of pressure change caused by the evaporation of the liquid fuel compared to the thermodynamic ideal-gas law calculations, methane and then air was fed in slowly using a fine needle valve and a pressure transducer to control the filling process. The technical air was heated by an external heater before flowing into the combustion vessel to better approximate an isothermal filling process. After the filling process was completed, the heater mounted inside the vessel was turned off, and 3 minutes of quiescence has been given to minimize any flow structures and/or temperature stratifications inside the vessel. The quiescence time also enhance the homogeneous mixing of fuel an air.

For each tested condition, the described experimental procedure that allowed the evaluation of the Heat Release Rate (HRR) and therefore the duration of 10% and 10-90% MFB was carried out at a minimum of three times. The average values of MFBs are reported as well as error bars evaluated based on standard error.

### SI Internal Combustion Engine

A 0.5L displacement, single cylinder SI research engine shown at [Figure 3](#) with a bore of 88 mm, stroke of 82.1mm and compression ratio of 10:1 was used for the SI engine experiments. The engine speed and load were held constant at 2000 RPM and 2.7 bar IMEP respectively. As suggested by Stanfield [25] and supported by the authors, this has been a common industry test condition and typical of the NEDC test. In the current engine, Stanfield [25] has quantified the bulk incylinder air motion reporting values of the tumble and swirl ratio at the same test condition.

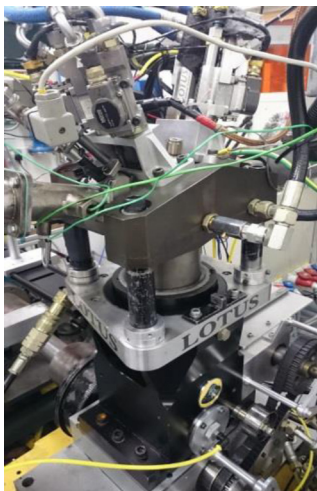


Figure 3. Single cylinder research engine

To allow for the gaseous experiments, the engine was fitted with a natural gas port fuel injector, Bosch NGI-2. The injector was supplied with compressed natural gas (92-96% CH<sub>4</sub> mass content) via a two-stage regulator which allowed regulation up to 13bar; this was set to 4bar for all tested conditions. Fuel injection pressure was chosen to be within the linear operating range of the injector specification where existing characterization data was available. The engine specifications and operating conditions are presented in [Table 1](#).

Table 1. Engine specifications and operating conditions

|                                    |                              |
|------------------------------------|------------------------------|
| Displaced Volume                   | 0.5 L                        |
| Bore                               | 88 mm                        |
| Stroke                             | 82.1 mm                      |
| Compression Ratio                  | 10:1                         |
| Intake Valves Open / Close / Lift  | 15° BTDC / 45° ABDC / 9.35mm |
| Exhaust Valves Open / Close / Lift | 45° BTDC / 15° ABDC / 9.35mm |
| Engine Speed                       | 2000 RPM                     |
| Engine Load                        | 2.7 bar IMEP                 |
| Spark Timing                       | 35° BTDC                     |

For all experiments, gas was injected into the manifold immediately after the intake valves are closed to maximize the available time for mixing. Gasoline fuel was introduced via an early direct injection (30° ATDC) using a multi-hole GDI injector and an injection pressure of 12MPa. This timing was chosen to maximize the time for mixing whilst not impinging the piston. For all experiments, spark timing was set constant at 35° BTDC and the equivalence ratio to stoichiometric. Control of the fuelling as well as the ignition was provided by an AVL 2410 instrument controller. This allowed full flexibility of the start and duration of each injector independently.

To calculate the required fuelling durations, the natural gas run was carried out first. Throttle position and injection duration was adjusted in order to achieve the target engine load of 2.7 bar IMEP. The mass of gas required at this condition is calculated from a previously obtained injector calibration relationship. For subsequent runs with dual fuel, this mass is divided according to the required dual fuel ratio, to give the desired natural gas mass. The gasoline injection duration is then adjusted in parallel with a fine tune of the throttle position, until the same engine load of 2.7 bar IMEP is achieved at stoichiometric operation. For example, at DF50, half of this amount of natural gas is injected ensuring that approximately 50% of the required energy to achieve the engine load comes from the natural gas, with the other 50% from the adjusted gasoline injection.

The process for each test condition required the engine head to first be heated using the lubricating oil to a temperature of 318-323 K. The engine was then motored at 2000 RPM before activating the ignition