A comparative study of combustion of methane in air-fuel, CO_2 diluted partial oxy-fuel, and oxy-fuel modes in an SI engine

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Introduction

By 2050, renewables will represent the bulk of the energy supply [1]. Due to the mismatch between renewable production and demand, the manufacture of "e-fuels" as a form of long-term seasonal energy storage is likely to achieve significant uptake [2, 3, 4]. Furthermore,e-fuels such as synthetic CH_4 and H_2 can be produced where renewable energy is abundant and transported across the globe, resulting in a carbon-netural electrical grid. Currently 30 GW of engine power plant capacity, operating largely with natural gas (NG), contributes to power generation in the EU [5], which is approximately 10% of the current installed wind power across the member states [6]. Due to their high dispatchability and ramping capabilities, the IC engine power plant is an ideal technology for peaking demand and grid stability, which will become increasingly important with electrical grids featuring a high degree of intermittent generation. Currently, these engine power plants emit pollutants, such as oxides of nitrogen [7], which have both global warming potential and a negative impact on human respitory health. The advanced combustion technique known as oxy-fuel combustion is a compelling solution to eliminate these species in IC engines while sustainably reusing existing natural gas engine assets. Oxy-fuel combustion is a process in which a fuel is combusted in the presence of oxygen rather than air, eliminating N_2 in the intake mixture and thereby avoiding the formation of NO_x. In practice, oxy-fuel engines could valorize streams of oxygen produced as a byproduct of water electrolysis for green hydrogen [8]. The CO_2 in the exhaust stream of an oxy-NG engine is substantially less energy intensive to capture than that of conventional engines due to the high CO_2 concentration in oxy-fuel exhaust gas, and an oxy-fuel IC engine operating with synthetic methane could even achieve net-zero CO_2 power generation. Despite these compelling possibilities, realizing robust oxy-fuel combustion in reciprocating engines requires a diluent to replace N_2 to maintain reasonable operating temperatures during the combustion process. Recycled exhaust CO_2 from the oxy-NG combustion process is one possible pathway, but there is limited study regarding the behavior of CO_2-diluted, spark-ignited (SI) oxy-NG combustion in IC engines, particularly with respect to experiments. This work will therefore provide an initial investigation, both experimental and numerical, of the combustion event in an oxy-fuel SI engine using methane as a surrogate for NG. Moreover, this study is conducted over a range of different oxidiser mixtures that feature CO_2 dilution and oxygen enrichment.

Methodology & Results

In typical reciprocating engine studies, the air-fuel mixture strength is characterized by the equivalence ratio or its reciprocal, λ. However, as the oxidiser in this work has a composition that is different than air (i.e. a mixture of O_2, N_2 and CO_2), a more general fuel-to-oxidizer ratio is adopted:

$$\lambda^{'} = \frac{\frac{\dot{m}_{O_2}}{\dot{m}_{CH_4}}}{\left\{\frac{\dot{m}_{O_2}}{\dot{m}_{CH_4}}\right\}_{stoichiometric}} \tag{1}$$

where $\dot{m}_{O_2}$ represents the flow of O_2 and $\dot{m}_{CH_4}$ represents the flow of CH_4 into the engine cylinder. Table 1 shows all operating conditions investigated, where V_{CO_2}, V_{N_2} and V_{O_2} are the volume percentages of CO_2, N_2 and O_2, respectively. The intake pressure of the engine is set to atmospheric pressure to replicate a full load naturally aspirated engine and the engine spark timing is set to the maximum brake torque point at all operating conditions. The single-cylinder engine and test bench layout is modelled using a 1D gas dynamics and 0D combustion solver in the GT-POWER framework, which allows for a detailed characterization of the combustion event based on measured in-cylinder pressure, as well as energy balance analysis over the thermodynamic cycle.

Table 1: Operating conditions for experimentation

Sl. No.	V_{CO_2} $(vol.\%)$	V_{N_2} $(vol.\%)$	V_{O_2} $(vol.\%)$	λ'	IT range $(°CA)$ $(bTDC)$
1	0	79	21	1	21
2	3.5	74.7	21.8	1.03	21 to 23
3	7.7	69.8	22.5	1.07	22 to 24
4	11.3	65.5	23.2	1.10	23 to 25
5	14.8	61.2	24	1.14	24 to 26
6	18.9	56.5	24.6	1.17	25 to 27
7	22.4	52.3	25.3	1.21	26 to 28
8	26	47.9	26.1	1.24	27 to 29
9	29.5	43.8	26.7	1.28	27 to 29
10	32.9	39.6	27.5	1.31	28 to 30
11	36.1	35.7	28.2	1.35	29 to 31
12	39	32	29	1.39	30 to 32
13	42	28.2	29.8	1.42	31 to 33
14	45.5	24.1	30.4	1.45	31 to 33
15	48.1	20.6	31.3	1.49	31 to 33
16	51.4	16.9	31.7	1.53	32 to 34
17	53.7	13.4	32.9	1.57	33 to 35
18	56.9	9.8	33.3	1.60	34 to 36
19	59.4	6.5	34.1	1.63	34 to 36
20	61.7	3.2	35.1	1.67	35 to 37
21	64	0	36	1.70	36 to 38

During experiments, the oxidiser composition was gradually changed from stoichiometric air-CH_4 combustion towards CO_2-diluted oxy-CH_4 combustion with O_2 enrichment. Figure 1 shows the in-cylinder pressure on the left hand side, both for experiments and simulations, and the rate of heat release, $dQ/d\theta$, on the right hand side. Experiments show that the overall in-cylinder pressure and temperature decreased due to the reduction in combustion speed and γ of the in-cylinder mixture. The higher density and higher molar specific heat of CO_2 compared to N_2 results in slower flame propagation with increased CO_2 replacement. Figure 1 shows that the heat release rate profile broadens and the peak heat release rate decreases by 20 $J/°CA$ when considering the extreme cases (i.e. 0% and 64% CO_2 by volume in the oxidiser). This results in an increased combustion duration (CA10-CA90) of 20$°CA$ requiring an advance of ignition timing and an indicated efficiency decrease of 9%.

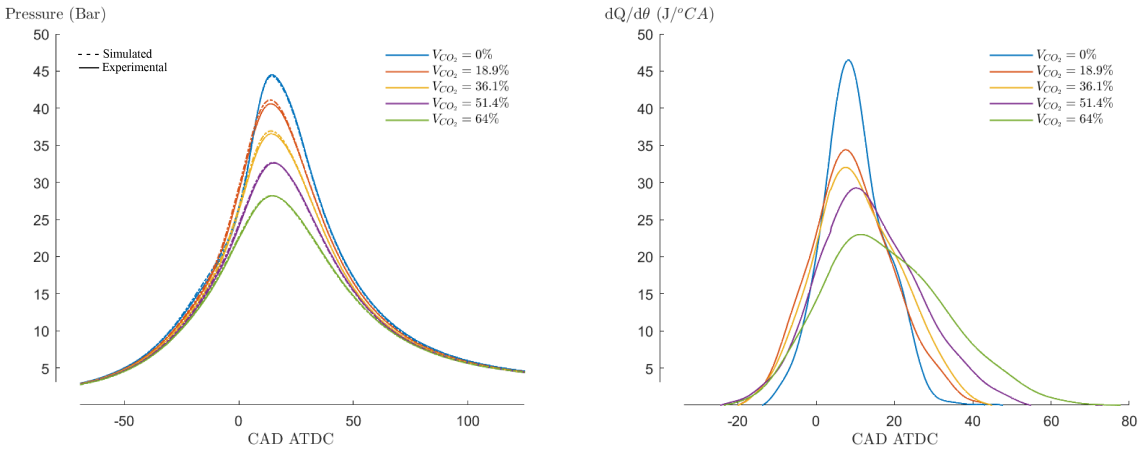

Figure 1: Experimental and simulated in-cylinder pressure and $dQ/d\theta$.

The decrease in indicated efficiency can be explained from the energy balance over the cycle. Figure 2 displays internal energy, work and heat losses at exhaust valve opening (EVO). As CO_2 replaces N_2 in the oxidizer mixture, internal energy rises gradually, while work and heat losses decrease. More energy goes into heating the gasses in the combustion chamber instead of producing work, and although the heat losses are decreasing, it is not sufficient to overcome the loss in work.

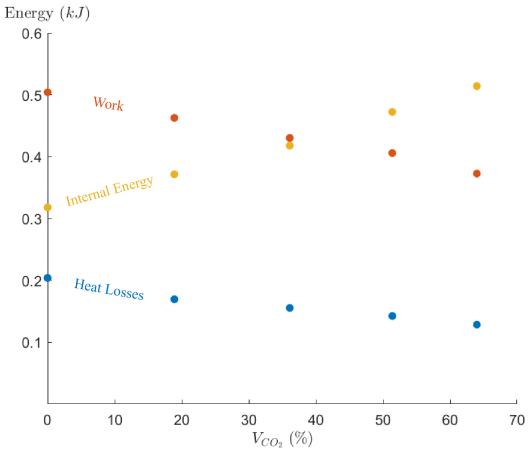

Figure 2: Internal energy, work and heat losses at EVO for changing oxidiser compositions.

References

[1] Gauthier Limpens, Xavier Rixhon, Francesco Contino, and Hervé Jeanmart. Energyscope pathway: An open-source model to optimise the energy transition pathways of a regional whole-energy system. *Applied Energy*, 358:122501, 2024.

[2] Joseph Ferrari. *Renewable fuels for long-term energy storage*, pages 109–138. Elsevier, 2021.

[3] Véronique Dias, Maxime Pochet, Francesco Contino, and Hervé Jeanmart. Energy and economic costs of chemical storage. *Frontiers in mechanical engineering*, 6:21, 2020.

[4] Xavier Rixhon, Gauthier Limpens, Hervé Jeanmart, and Francesco Contino. Taxonomy of the fuels in a whole-energy system. *Frontiers in Energy Research*, 9:660073, 2021.

[5] European Engine Power Plants Association. Engine power plants: A dynamic eu-based manufacturing sector.

[6] WindEurope. Wind energy in europe: 2022 statistics and the outlook for 2023-2027.

[7] Kelly Senecal and Felix Leach. *Racing toward zero: the untold story of driving green.* SAE International, 2021.

[8] International Energy Agency. promising signs in electrolyser manufacturing add to growing momentum for low emissions hydrogen. https://www.iea.org/news/promising-signs-in-electrolyser-manufacturing-add-to-growing-momentum-for-low-emissions-hydrogen, 2022.