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# Research on an internal combustion engine with an injected pre-chamber to operate with low methane number fuels for future gas flaring reduction

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ABSTRACT

Flaring gas is the action of burning waste crude natural gas that is not possible to process or sell during the extraction and processing of oil and gas extraction. However, the surge of the oil and gas (O&G) fields caused an augmentation of flared gas; making society aware of its impact on the planet. To solve this problem, internal combustion engines showed clear advantages. However, there is a lack of information about how to utilize this associated petroleum gas (APG) as fuel for these engines due to its reduced methane number (MN). A methodical investigation about the optimal combustion and design needs for low MN fuels is proposed based on tests conducted in a natural gas engine with an injected pre-chamber ignition technology and a future technology that could replace flaring is proposed. Experiments conducted when using low MN gases showed different misfire limits and knocking margins. A 15% efficiency drop was obtained, however, this could be considered as a good performance as the Brake Mean Effective Pressure (BMEP) or output power reduction was 53.3%. In consequence, different engine design modifications are proposed to improve the former situation.


## 1. Introduction

Flaring gas is the action of burning natural gas crude waste. This gas usually lies in the top part of the reservoir in close contact with oil of which is either burned or expelled into the environment. Corporations typically consider it a by-product of oil, and they dispose of it through a loco flare combustion process [1].

However, it is normal to utilize a portion of the conditioned natural gas as fuel to cover necessities of the site, while the remaining portion of the natural gas is treated, injected to the network or sold. The surplus gas is flared where it is not feasible to process or vent as a result of the absence of infrastructure, nonexistence of a gas market that is near or some other economic reasons [2].

Until a few decades ago, flaring was considered environmentally admissible (Bashir, 2010). The surge of O&G fields caused a rise in flared gas making society aware of its severe impact on the planet [3,4]. During the combustion process various pollutants are released. For instance, nitrogen oxides ($NO_x$), sulphur oxides ($SO_x$), hydrogen sulphide ($H_2S$) and volatile organic compounds (VOCs) are released [5].

The Global Gas Flaring Reduction (GGFR) program led by the World Bank [6] states that the yearly volume of APG flared in the whole planet world ascends to 150 billion $m^3$ (close to the 30% European Union's yearly natural gas consumption) and close to 400 million tons of $CO_2$. That is the reason why the Zero Routine Flaring by 2030 program was released. In fact, this initiative has been supported and signed by more than 55 extraction companies, governments and institutions with the goal of decreasing the impacts caused by using flare practises.

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As a result of the rising consciousness of the environmental concern, and the ratification of the Kyoto protocol and Paris agreement by the majority of the membering countries, the expectation was to considerably reduce the amount of flared gas [7].

A study carried out by Carbon Limits AS (2013) shows evidence to believe that due to corporate standards and policies and regulatory pressures, oil companies will step up their efforts to reduce and **avoid flaring of APG in Russia, Kazakhstan, Turkmenistan and Azerbaijan**. Lately, Russia has increased the awareness of the polluting effects of gas flaring and venting leading to tightening of policies. In 2012, the Russian government introduced penalties and two years later penalties were increased. In this particular case, Russian Federation's aim is to increase the use of APG to 95% and has yet to achieve a significant reduction [8].

Accordingly, there is a big potential in a country like **Algeria**. The Oil&Gas sector showed great potential for gas engine development while 9% of APG was being flared. The potential could rise at least until 300 MW of power generation, reducing Algerian $CO_2$ emissions by $1.43 \times 10^6$ tons. Finally, a remarkable fact was detected in **Romania**, which reflects the potential for these installations in that country. It was recently signed an agreement for 100 MW of engines running on APG for 2020.

Despite all sacrifices, the yearly flaring figures have remained almost constant for the last exercises. This is explained by the increase of worldwide oil production and therefore the growth of wellhead gas which commonly lies in close contact with oil as previously explained. Similarly, the absence of legislation and the current restraints in gas exploitation, market development and infrastructure are the principal contributors to persist flaring (Djumena, 2004).

To sum up, flaring gas is hazardous to the well-being of humans as well as the planet, particularly near the wells where the burning process is located. From an economic perspective, gas flaring is a waste of resources available due to not leveraging the energy from the gas that is being disposed.

Presently there are plenty of alternatives suggested by several scientists to take profit of this by-product. However, many of those are not feasible or efficient from a monetary point of view.

Over the last few years extraction enterprises, driven by environmental guidelines and financial indicators, have evaluated several alternatives to lessen the quantity of gas flared. Mourad et al. [9] presented different ways to avoid flaring:

1. Reinjecting flare gas into wells to improve O&G extraction.
2. Collecting and transporting the APG to treatment plants.
3. Transforming APG into Liquefied Natural Gas (LNG).
4. Transforming APG into liquid: Gas-To-Liquid (GTL) or Liquefied Petroleum Gas (LPG).
5. Using APG for electricity and heat generation (CHP-Combined Heat and Power).
6. Using it as a fuel for onsite necessities.
7. Utilisation of APG as a feedstock for petrochemicals production.

The first two proposals currently only find a payback for amounts superior to 10,000 $Nm^3/h$. However, the electricity generated through CHP seems to be an appropriate alternative for flows around a few thousands of $Nm^3/h$.

Exploring this solution is supported by the findings reached by Bakhteeyar et al. [10]. The conclusions of his study show that CHP becomes the best option when availability is a priority indicator. However, the usage of a pipeline for transporting gas comes first if an environmental indicator is taken as the priority and the reinjection solution secures the best position if a financial indicator takes priority. When all of these factors are considered equally important, CHP obtains results that are comparable with reinjection and pipeline utilisation.

Despite evidence that pipe usage or injection may be the most suitable alternatives, it must be reiterated that these solutions can only be applied in fields where the APG is extracted in incredibly large amounts. Additionally, these methods also need local developed markets and a great infrastructure or distribution network in the area that in most of the times not available. The reasons listed indicate there is a considerable niche for CHP applications, mainly in those fields where there is a little amount of APG available or just when there are scarce possibilities to vend natural gas.

For reduced flows of APG, previous studies showed that power generation seems to be an appropriate option for its application in flare gas recovery (Khalili-Garakani et al. [11] and Mansoor and Tahir [12]. In this sense, power generation uses flare gas to generate electricity to be used on site or for sale. This can be achieved with minimal pre-treatment of the gas in accordance with the needs of the gas turbines or engines. It should be noted that when we use flare gas, the objective is to generate the maximum power while spending minimum capital and operating costs (Khalili-Garakani et al. [11].

An in depth revision of research about the power generation technologies by employing flare gas (Nezhadfard and Khalili-Garakani [13], show analysis under different scenarios with gas turbines, Internal Combustion Engines (ICE) and Solid Oxide Fuel/Cell/Gas turbine Cycles, between others. Its main results showed that ICE has the best economic performance over gas turbines like higher single efficiency, more efficient part-load operation, speedy start-up performance and it requires lower pressure compared to a gas turbine.

At the same time, other research about power generation analysed the economic parameters at different ambience temperatures (Morteza et al. [14], and Jafarian et al., [15]. Its results showed that power generation and its economic implications are a function of the flow and composition of the gas, which implicates again a special interest in ICE.

From the emissions point of view, previous researches showed that power generation from flare gas leads to a significant reduction in $SO_x$ and CO emissions but not $CO_2$. Once again, due to Gas Turbine Cycle having higher emissions the interest in ICE was highlighted [13].

The particular drawback of utilizing internal combustion engines is that the APG is a low methane number (MN). Methane Number (MN) is defined as a gas' resistance to detonation. For APG, methane number could be in a wide range of values between thirty and ninety depending on the extraction well and its composition [16]. That is why APG is a low methane number fuel and the lower the value the higher prone to detonation the fuel will be (knocking risk). This knocking is related to a combustion abnormality created when the front of the flame propagates quicker than the speed of the sound. At that moment, the pressure wave raises the energy of the not burned fuel by compression and this detonation creates an irregular ignition that, as a result, may derive from mechanical failures mainly concentrated in pistons and other areas of the combustion chamber. In consequence, the lower the MN the lower the knocking margin and therefore the engine also suffers a considerable output power reduction which is also linked with a thermal efficiency decrease [1].

At the same time, if a determinate temperature is surpassed during the compression stroke, free radicals are produced that provoke the auto-ignition of the charge, liberating its energy sooner than anticipated.

Previous studies [17] investigated the needs of engines manufacturers defining a lower limit of methane number of 80 for its NG fuels. Despite this, the technology aims to employ lower methane

number fuels in internal combustion engines, for instance, to inject hydrogen in the existing natural gas network or for upgrading APG gases to be utilized in power plants [18]. This would reduce significant investment costs associated with the development of new transmission and distribution infrastructures of NG and the payback investment, respectively.

It is time to define the causes and main strategies to prevent the knocking and auto-ignition phenomenon. Most of the standard gas engines available on the market were designed to burn fuels with a composition comparable to natural gas in terms of methane number which is above sixty-five. Thus, low methane number (such as APG) is not compatible with this type of internal combustion engine. To prevent knocking issues, internal component modifications will be necessary to successfully run the engine with fuels with MN values in the range of 35–60.

As a consequence of this, the drawback expressed can be tackled utilizing two main strategies. The first way would consist of raising the methane number of the APG through a cleaning system allowing a standard gas engine to run without detonation risks. From now on, this method will be named as Non-Derated Internal Combustion Engines (ND-ICEs).

These days, various gas treatment technologies that increase the MN extracting heavy hydrocarbons are offered by the market. In fact, the most relevant technologies are listed below [19].

1. Low-temperature separation.
2. Separation of gas fractioning.
3. Adsorption.
4. Membrane.

By utilizing these treatments, the methane number could be raised in a range of ten and twenty points [20].

As outlined by Zyryanova [18] during her financial comparison between different engine OEMs (Perkins and Waukesha) and electrical output powers in the range of 1000 kWe, sites equipped with a catalytic reformer that converts well gas into methane–hydrogen mixture seem to provide a quicker capital return, in comparison to the sites that are directly running on APG. Better financial performance is reached due to longer maintenance intervals, increased life of the components and higher output power. Thus, looking at the results the former technology that is able to upgrade the MN of the extracted gas seems to be a valuable option for CHP applications.

Conversely, the second methodology resides in directly feeding the engine with APG without any treatment. This causes a reduction of the output power which consequently reduces thermal efficiency. From this point on, this method will be named Derated ICEs (D-ICEs).

It is also known that feeding the engine directly with APG creates unfavourable conditions with the operation of the rotating equipment. Also this reduces the life cycle of the equipment (even more than twice) as a consequence of several motives. In this sense, the study conducted by Iora et al. [1] that examines and weighs D-ICEs and ND-ICEs from an environmental and financial standpoint deduces that D-ICEs present considerable strength from an environmental viewpoint when compared to ND-ICEs, having both solutions a comparable return for the needed investment. Through the aforementioned study, it was concluded that using D-ICE to burn APG is a feasible solution to reduce the amount of flared gas and indeed one of the most robust ways to tackle this environmental problem.

The design improvement of engines to support these low methane fuels were investigated in the last decades and showed the need of a new normalized knock indicator to enable the comparison of different engines designs. In particular, laminar flame speed was assumed to be more sensible to MN variation parameters to be employed to compare different engine designs [21].

To control this indicator, other works aim to improve this engine design with different modifications like engines with prechamber [22] or an E–Pilot approach and a large–squish piston [23] showing an improvement of the Break Thermal Efficiency and $CH_4$ emissions. At the same time, other research works showed mathematical methods to optimise the performance of internal combustion engines with new fuel tests [24].

All these previous works are general studies about internal combustion engines and no detailed information about how to improve its design and how to optimise its working indices are shown based on real sampled data. In field, it is in fact quite complicated to distinguish between knocking and auto-ignition. They indeed create identical consequences and the only way to differentiate them would be by adding a pressure sensor inside the combustion chamber and plotting the pressure graph. This instrumentation is normally only installed in test beds under a controlled atmosphere such as an R&D laboratory as it needs distilled water to cool down. In consequence, a detailed real case study to understand the main problems into employing APG in ICE is described in this research work.

In particular, after releasing a 2 MW natural gas fired engine named SGE-86EM, and based on the research needs in power generation with APG, Siemens Energy in collaboration with the University of A Coruña, started the investigation about the design and operation requirements of a particular engine that could run with low methane number fuel and yield such high power output. All in all, the goal of the article is to investigate based on real sampled data of the technical parameters to convert an ICE designed to operate with natural gas to burn APG in agreement with previous research works indications and the international needs.

## 2. Material and methods

### 2.1. Test bench

The present project was developed in Araba's Technological Park (Basque Country) and more specifically in Siemen's R&D centre. There, the initial analytical tests defined the physicochemical characteristics and combustion properties of the gas used by means of a gas chromatograph micro GC490. The first screening served to certify the fuel properties, such as methane number, lower heating value, carbon, hydrogen and nitrogen contents.

After defining the gas properties, engine tests were carried out on the SCE test bench. Fig. 1 shows the layout of the test bench and its monitoring is mainly completed by the instrumentation listed in Table 2. The engine used was a single cylinder engine (SCE) specifically designed for developing the two engine versions mentioned before. In this sense, the SCE is prepared to simulate the atmosphere of the recently launched MCE version with the consequent benefits explained by S. Brewster et al.

Like in similar studies [25–27], during the combustion process all the variables measured were recorded for further computational analysis. Amongst all the measured variables some of them can be highlighted: ignition timing, pre-combustion period, end of main combustion, main combustion period, post-burning period, maximum rate of heat release, knocking, exhaust temperature, thermal efficiency and cylinder pressure, between others.

The AVL PUMA system was installed as a central element for controlling the testing process. This equipment is responsible for collecting all the data originated in the gauges installed in the engine (the most relevant outputs can be observed in Table 1). In the same way, it centralizes the control of various subsystems of the

**Fig. 1.** SCE test bench.

SCE test bed. Each of them, has a small Programmable Logic Controller (PLC) which is communicated with the PUMA.

The combustion characteristics were evaluated by means of the AVL Indicom module. This is a combustion measurement software which is mainly used for the analysis and measurement of the intake, exhaust and combustion chamber pressure curves, as well as engine specific variables such as valve lift curve, flow coefficients, fuel parameter. The results also include the heat release rate, energy and thermal balance, heat losses, mass flow, residual combustion gases in the main chamber and many other variables. The data acquisition system used is the universal high speed Indimaster Advanced GigabitTm module with high channel flexibility, having up to 8 channels available. At the same time, a new branded HORIBA MEXA 7100D exhaust gas analyser was used for the analysis of the exhaust gases of the SCE such as THC, CO, $CO_2$ or $NO_x$.

### 2.2. Test design

Due to the possibility to reduce gas emissions by employing flare gas in natural gas combustion engines, it is needed to identify the modifications of the technical parameters of internal combustion engines to get adequate combustion when utilizing flare gas (propane in our research) with respect to a well-known gas such as natural gas as a consequence of its different methane numbers of 75 and 35, respectively. This way, it is possible to evaluate how much the current engine had to be derated in order to safely burn a low methane number fuel and therefore consider if a combustion chamber re-design is needed for the application proposed.

To reach this objective, an original testing procedure was

**Table 1**
Engine parameters.

| | Version 1 | Version 2 |
|---|---|---|
| Manufacturer | Siemens | Siemens |
| Type | SGE-86EM | SGE-100EM |
| Number of cylinders | 12 in V | 12 in V |
| Output power | 2000 kW | 2000 kW |
| Rated engine speed | 1500 rpm | 1200 rpm |
| Bore | 195 mm | 195 mm |
| Stroke | 240 mm | 280 mm |
| Connecting rod length | 530 mm | 510 mm |
| Total swept volume | 86 l | 100 l |
| Number of valves | 2 inlet and 2 exhaust | 2 inlet and 2 exhaust |
| Swirl ratio | 0 | 0 |
| Volumetric compression ratio | 13.5 | 13.5 |
| Turbocharger | High efficiency | High efficiency |
| Fuel and air mixer | Venturi mixers | Venturi mixers |
| Ignition system | Fuel injection prechamber | Fuel injection prechamber |
| Spark plugs | Guascor spark plug | Guascor spark plug |

**Table 2**
Test bench sensors.

| Engine parameter | Range | Measuring equipment | Nomenclature |
|---|---|---|---|
| Prechamber/intake pressure | 0–40 bar/4–20 mA | WIKAS -10 | PA4_101 |
| Prechamber intake temperature | 0-100 °C/4–20 mA | WIKA TR 30-W | TA4_101 |
| Prechamber inlet flow | 0–2 kg/h | VÓGT LIN GSC C9TA | QA4_101 |
| Chamber intake pressure | 0–40 bar/4–20 mA | WIKAS -10 | PA4_001 |
| Chamber intake temperature | 0-100 °C/4–20 mA | WIKA TR 30-W | TA4_101 |
| Chamber inlet flow | 0–6500 kg/h | ENDRESSHAUSER-PROMASS | QA4_101 |
| Exhaust gas temperature | 0-750 °C | TC K | TA5_001 |
| Exhaust gas pressure | 0-150 °C/4–20 mA | AVL PT100 | PA5_001 |
| Water inlet pressure | 0–6 bar | TECSIS P3249b074001 | PWP_002 |
| Water inlet temperature | 0-150 °C/4–20 mA | AVL PT100 | TWP_002 |
| Water outlet pressure | 0–6 bar | TECSIS P3249b074001 | PWP_301 |
| Water outlet temperature | 0-150 °C/4–20 mA | AVL PT100 | TWP_301 |
| Oil inlet pressure | 0–10 bar | DANFOSS | P05_001 |
| Oil inlet temperature | 0-150 °C/4–20 mA | AVL PT100 | T05_001 |
| Oil outlet pressure | 0–10 bar | DANFOSS | P09_001 |
| Oil outlet temperature | 0-150 °C/4–20 mA | AVL PT100 | T09_001 |
| Crankcase pressure | (−) 300-300 mbar | | P08_001 |
| Main chamber air inlet flow | 0–70000 kg/h | PROMASS ENDERHAUSSER | QAR_001 |

proposed in two phases in order to identify the optimal pre-chamber gas volume and timing to be employed in a subsequent analysis of the effect of each fuel on the engine performance.

*2.2.1. Phase I: optimal pre-chamber gas volume definition*

The percentage of gas in the pre-chamber is the division between the quantity of gas in the pre-chamber and the quantity of gas in the main chamber. To identify the optimal pre-chamber gas volume needed for each fuel a constant value was selected for each different test followed by increasing the BMEP gradually until the timing that offers a desired knocking margin was found. After this, in order to analyse the effect of the air-fuel ratio (AFR), this parameter was increased slowly starting from the lean limit (engine misfire) and finishing on the rich limit or smooth detonation area.

*2.2.2. Phase II: Ignition timing definition*

Due to the ignition timing being variable for each combustible, this had to be selected prior to starting the tests and giving the priority to output power as opposed to engine efficiency. The exhaust temperature and the pressure curve were the limiting parameters for the engine timing reduction, therefore, the ignition timing was reduced as much as possible. If the ignition timing was too retarded, the exhaust temperature would raise and may go over the maximum temperature allowed by the exhaust manifold material. At the same time, if the pressure curve becomes unstructured (too slow), this phenomenon would affect the engine's stability causing a trip by misfire.

In consequence, employing this methodology with both fuels (Natural gas and propane), two base cases of Natural gas as fuel and another two experiments with this new fuel (propane) were carried out:

- **Base case 1:** Natural gas (75MN and 38,000 kJ/Nm$^3$) as fuel to determine the optimal pre-chamber flow.
- **Base case 2:** Natural gas (75MN and 38,000 kJ/Nm$^3$) as a fuel with the standard configuration of the SGE-86EM to get the $NO_x$ curves.
- **Experiment 1:** Pure propane. A low methane number gas (35MN and 93,350 kJ/Nm$^3$) as fuel to determine the optimal pre-chamber flow.
- **Experiment 2:** Pure propane. A low methane number gas (35MN and 93,350 kJ/Nm$^3$) as a fuel and the standard configuration of the SGE-86EM is used to get the $NO_x$ curves with several timings.

Finally, it is interesting to highlight that all the experiments were done with the standard combustion chamber configuration for natural gas which can be summarised as the following:

a. Fuel injected pre-chamber
b. High compression ratio flat pistons
c. Cylinder heads with limited swirl
d. Miller cycle

At the same time, the boundary conditions for all the experiments are the following ones:

- Main Outlet water temperature: 90 °C.
- Oil temperature: 83 °C approx.
- Oil pressure: 4.5 bar.
- Inlet air temperature: 48 °C approx.
- Water flow: 116 l/min.
- Water pressure: 3 bar.
- Backpressure control: constant efficiency; 0.8 turbine and 0.79 compressor.
- Speed: 1500 rpm.

Based on the previous tests, the effect of fuel on engine performance will be analysed with the following main objectives of:

- Characterise the performance of the **fuel injected pre-chamber** with low methane application (Experiment 1) and compare it with the natural gas one (Base case 1).
- To know the **maximum power** for an acceptable knocking margin of the low methane number fuel (Experiment 2).
- To know the maximum **efficiency** that can be obtained for that maximum power with the low methane number fuel (Experiment 2.)
- To know the **performance range** of the low methane number fuel at the obtained maximum load (Experiment 2).
- Compare the low **methane number** fuel performance with the NG performance (comparison between Base case 2 and experiment 2).

Finally, results are presented as a function of NOx emissions which enables the direct evaluation of the effect of the configurations in the knocking margin. Even if NOx values vary, rest of the engine parameters are fixed except for lambda values, which means that graphics represent NOx variations as a consequence of lambda modification. The decision to represent these curves under NOx values instead of lambda, is justified by the fact that all current regulations make reference NOx rather than lambda. Further to this, NOx is a value that can be easily measured on site which provides the opportunity to compare values whereas lambda figures are complicated to measure.

## 3. Results

In this section the main sampled data obtained is shown. Each point figures the mean of the variable during the last 60 s. In particular, base case 1 employs 14 samples, base case 2 employs 27 samples, experiment 1 employs 12 and, finally, experiment 2 employs 13 samples. Finally, base case 1 and 2 results will be briefly explained as they are considered to be a baseline of natural gas engines to be compared with propane (experiments 1 and 2).

*3.1. Base case 1 and base case 2: natural gas as fuel*

The most representative results of base case 1 and base case 2 are presented through the curves of sampled data showed in Figs. 2 and 3. In accordance with Fig. 2, it can be observed that the optimal percentage of gas in the pre-chamber for the natural gas engine is set at 1%, which is justified as the maximum efficiency achieved with the lowest flow possible. At the same time, Fig. 3 shows that for natural gas the BMEP at 500 mg$NO_x$/Nm$^3$ is 100% while the knocking margin would be 1600 mg$NO_x$/Nm$^3$.

*3.2. Experiment 1: propane (MN35) as fuel*

Once the baseline tests with natural gas were finished, the running point for a low methane number fuel (propane) was determined. Based on the previous test, it was concluded that a better efficiency-knocking margin compromise is obtained by reducing timing than by reducing gas flow. In particular, by reducing the gas flow the efficiency varies considerably while the knocking margin is barely reduced.

In consequence, experiment 1 was carried out at a BMEP at which the engine did not knock and maintaining constant

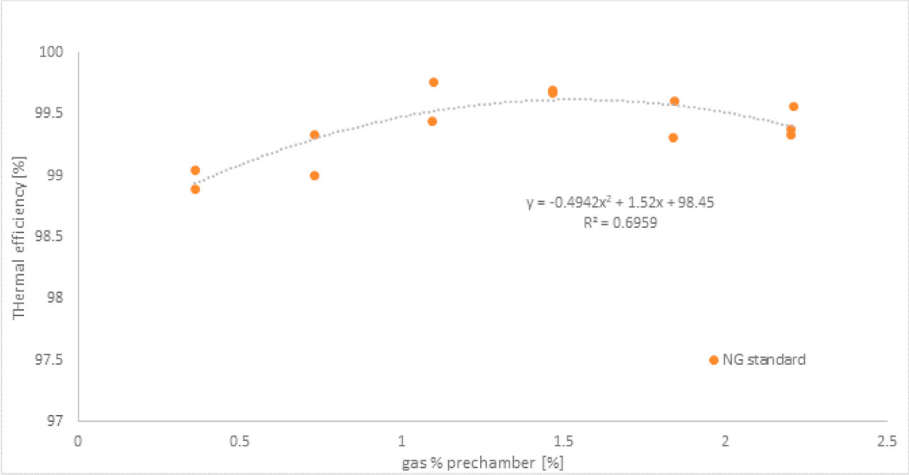


**Fig. 2.** Thermal efficiency vs gas % pre-chamber (14° BTDC@500mg$NO_x$/$Nm^3$, Natural gas).

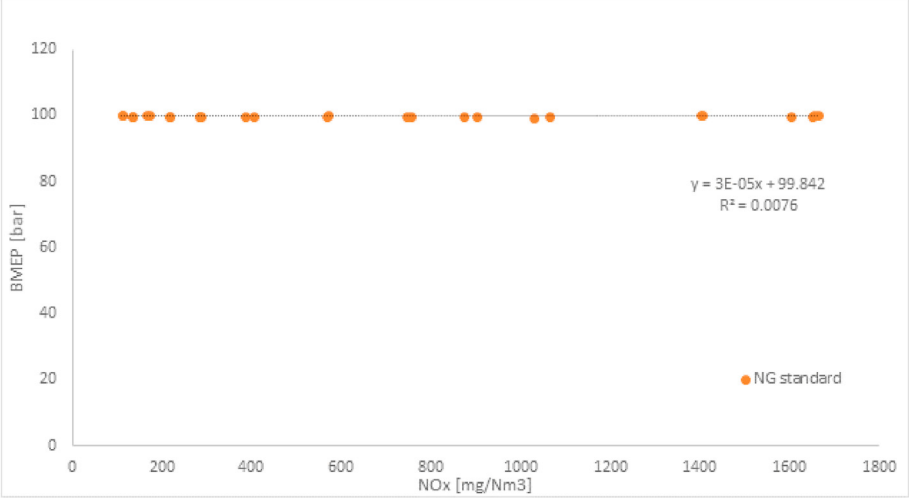


**Fig. 3.** BMEP vs $NO_x$ (Natural gas).

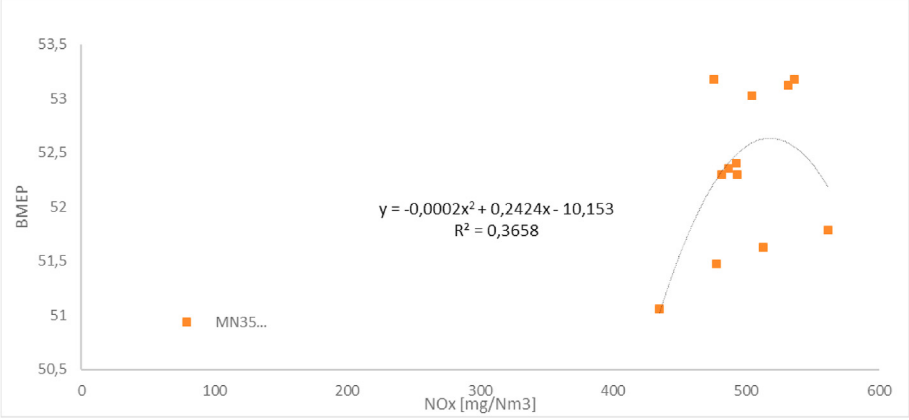


**Fig. 4.** BMEP vs $NO_x$ (Timing 14°BTDC).

emissions at 500 mg$NO_x$/$Nm^3$. Results showed that the power output was adjusted to avoid knocking which resulted in a 48% decrease compared to the natural gas nominal power, as shown in Fig. 4.

It is important to highlight that these results are preliminary, applying an expected optimal timing, for which the desired knocking margin is set and must be re-evaluated. In particular, this first test was conducted with a set value of 14° BTDC and

subsequently the optimal value will be defined in Experiment 2.

The criterion for selecting the optimal pre-chamber flow lies in selecting the lowest flow from among those that provide the highest thermal efficiency. As it can be observed in Fig. 5 the efficiency does not increase after the 0.6% pre-chamber gas point. This point is a very good compromise between three important factors that are shown in the aforementioned figures.

- Engine's thermal efficiency is optimal (Fig. 5)
- The temperature inside the pre-chamber is not the highest. As a general rule, the higher gas flow the higher the temperature inside the pre-chamber will be. The lower this temperature, the better it is for the spark plug life avoiding problems such as beading.
- The misfire limit for excess of gas is close to 0.8% and therefore it was not possible to record more points (Fig. 4). In fact, the Coefficient of Variation (COV) with 0.8% in Fig. 6 shows that combustion is already unstable. Therefore, it was concluded that it is best to try to move away from that point of 0.8%.

### 3.3. Experiment 2

To select the appropriate timing and knowing that the optimal flow in the pre-chamber was 0.6%, $NO_x$ curves were obtained under two different timings of 18° BTDC and 14° BTDC, as it is shown in Fig. 7. Lower timings were not tested as the combustion became too slow and the engine got too unstable to continue. From Fig. 7 it can be concluded that the greater possible ignition timing the better, as long a correct margin to knocking is reached. Therefore, as the goal was to get a 2000 $mgNO_x/Nm^3$, only the 14° BTDC can be selected, see Fig. 7.

In order to achieve the optimal operating point, several timings and output powers have been tested until reaching the combination that allows to have the desired knocking margin. In this sense, 2000 $mgNO_x/Nm^3$ in the case of MN35 was selected, but 1500 $mgNO_x/Nm^3$ for natural gas due to its greater stability in the composition. Power was increased until the engine started to knock. At that point, timing was reduced and continued to increase power until knocking appeared again. This iteration was repeated until the combustion was unstable or the exhaust temperature was too high. During the test, it was determined that the exhaust temperature with 14° BTDC is low enough so that the exhaust manifold can stand the values and misfire does not appear neither.

Once the optimal pre-chamber flow (0.6%) and timing (14° BTDC) were empirically selected, the combustion performance analysis will be done in the discussion section.

## 4. Discussion

### 4.1. Comparison of combustion performance

The different performances for the pre-chamber gas flow with natural gas and propane can be drawn out from Fig. 8. On the one

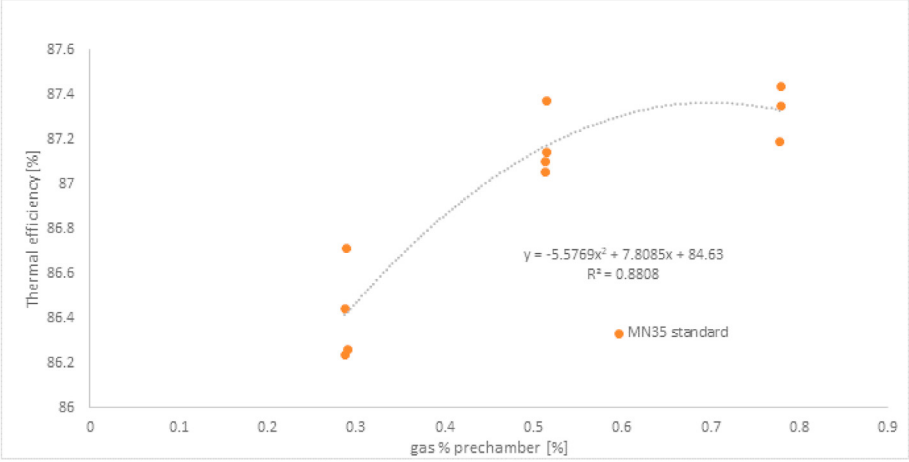


**Fig. 5.** Thermal efficiency vs gas % pre-chamber (MN35).

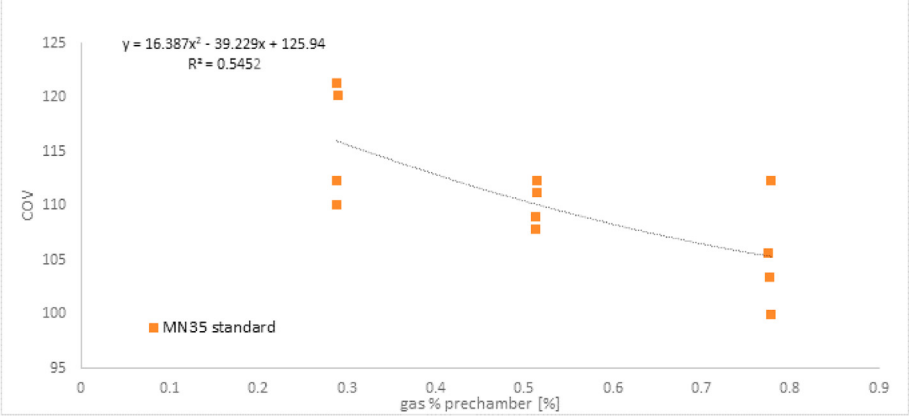


**Fig. 6.** Coefficient of Variation (COV) vs gas % pre-chamber (MN35).

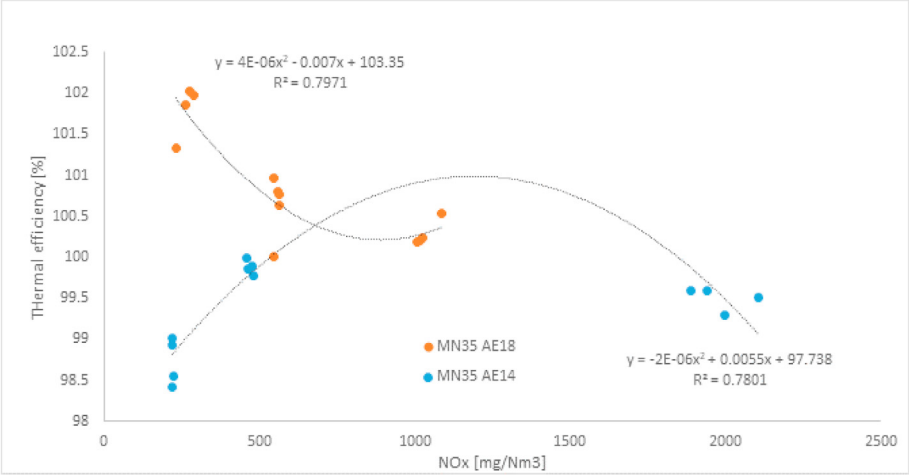


**Fig. 7.** Thermal efficiency vs NOx.

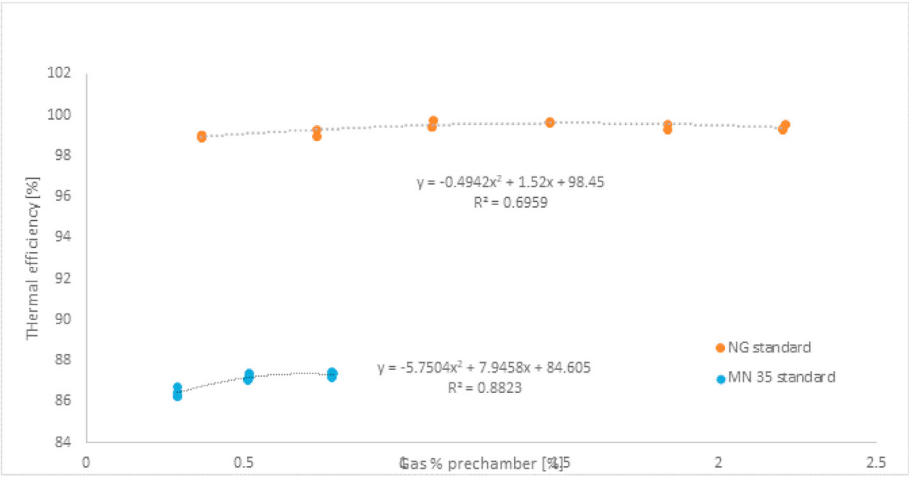


**Fig. 8.** Thermal efficiency vs gas % pre-chamber (NG and MN35).

hand, the misfire limits are completely different in both gases. Natural gas can work from 0.4% values to 2.0% whereas the margin in MN35 are very much shorter (from 0.3% to 0.8%). To understand this result, the lower and upper flammability limits (LFL and UFL) must be considered and fixed in 5%–15% for NG and 2.4%–9.6% for propane. This can be explained with the energy needed in the torches of the pre-chamber for getting an efficient combustion will depend on the energy of the gas. The more energy inherent to the gas the less quantity will be needed. For instance, as the lower heating value LHV of the natural gas (MN75) is 38,000 kJ/Nm$^3$ and the one for the propane (MN35) gas is 93,350 kJ/Nm$^3$, it seems reasonable to have a lower % value in the low methane number application.

The modification in the behaviour of the BMEP for each combustible is showed in Fig. 9 showing 2 different trends with the desired knocking margin: 1500 mg$NO_x$/Nm$^3$ for natural gas and 2000 mg$NO_x$/Nm$^3$ for the low methane number application. In this sense, this margin needs to be more conservative for several reasons. From one end, the APG gas quality fluctuation is higher than NG and from the other it must take into consideration that at the moment that this configuration is applied to a multi-cylinder engine, the risk of detonation usually increases due to variability in cylinders.

Once knocking margins for both fuels were analysed, special attention must be paid to the BMEP correlated with those margins as it is shown in Fig. 9. In particular, a reduction of the BMEP is closely related to a reduction of the output power. Given the above, it was concluded that Experiment 2 suffers a 53% of reduction in the BMEP. Base case 2, makes sense since the gas composition is much more prone to knocking and it starts sooner. It is important to note that this fuel is under laboratory conditions, and despite having a low methane number, it obviously does not have the same composition of a field gas which may contain impurities that worsen the performance and the life of the engine.

If the BMEP would remain untouched, the engine would not work smoothly, and sooner or later it would have a mechanical breakdown. The reason is simple, once the knocking margin is exceeded, the knocking phenomenon would appear. As a result, the engine's internal parts have to deal with pressures and temperatures higher than the ones they are prepared to handle. A simple way to overcome this problem is to lower the mean effective pressure as we have done in the present test.

Apart from the aforementioned, it is noteworthy that with the Natural Gas combustion technology 200 mg$NO_x$/Nm$^3$ have been achieved with a stable operation. This test was not focused to specify the minimum achievable $NO_x$, nonetheless, this is definitely

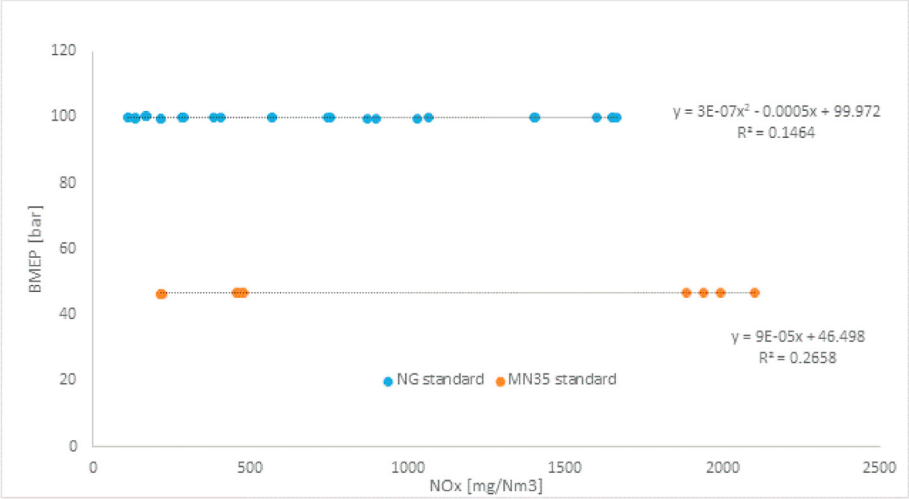


**Fig. 9.** BMEP vs $NO_x$.

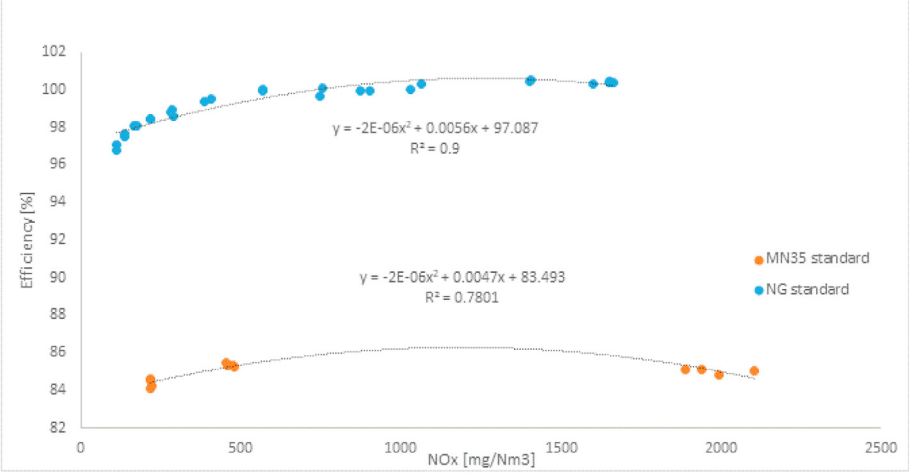


**Fig. 10.** Thermal efficiency vs $NO_x$.

a remarkable fact as it behaves in a similar way to its counterpart in natural gas (I [26].

By means of picture 10, it is confirmed that a considerable change of efficiency exists between Base case 2 and experiment 2. The main reason for this to happen is based on the BMEP operation point but the timing effect also needs to be acknowledged. The respective thermal efficiency decrease at 500 $mgNO_x/Nm^3$, between Base case 2 and experiment 2, was 15%.

In consequence, Fig. 10 indicates that there is a considerable loss of efficiency between both configurations. In order to analyse the reason of this reduction, further parameters of the combustion will have to be considered and evaluated in detail. In this sense, it is well known that a faster combustion implies a higher combustion efficiency. Considering this principle, Fig. 11 shows that Base case 2 (NG) and experiment 2 (MN35) have very similar values, it can be concluded that as the combustion technology remains the same there is not a direct effect from it on the combustion and therefore the thermal efficiency.

At the same time, Fig. 12 presents exhaust temperature as another parameter that could indicate a poorer performance of the new configuration. A high exhaust temperature is an indicator of a slower or retarded combustion and a more energy is wasted and not transformed into power according to Carnot Cycle's definition of efficiency. Despite this, high temperatures have a positive aspect that reside in the selection of the turbocharger. Due to the high temperature, exhaust gases possess higher energy to move the turbocharger, which will increase the boost pressure of the turbocharger (if needed). However, in this particular scenario, the exhaust temperatures are pretty similar which means that this parameter does not reflect the cause of the efficiency decrease either. At the same time, it is important to mention that the values keep increasing as the NOx becomes higher. Therefore, those higher values in Experiment 2 are normal and would be equivalent in Base case 2 if we would have shown the curve with a higher knocking margin.

Finally, to develop a complete analysis of the loss of efficiency, the peak pressure must be considered. This new variable is showed in Fig. 13 for both experiments indicating lower peak pressure in the Experiment 2, which is coherent and linked to the reduction of the BMEP in order to decrease the risk of possible detonations.

In conclusion, amongst the three variables analysed for explaining the reduction of the thermal efficiency, it was obtained that peak pressure is the one with the biggest impact in the efficiency reduction.

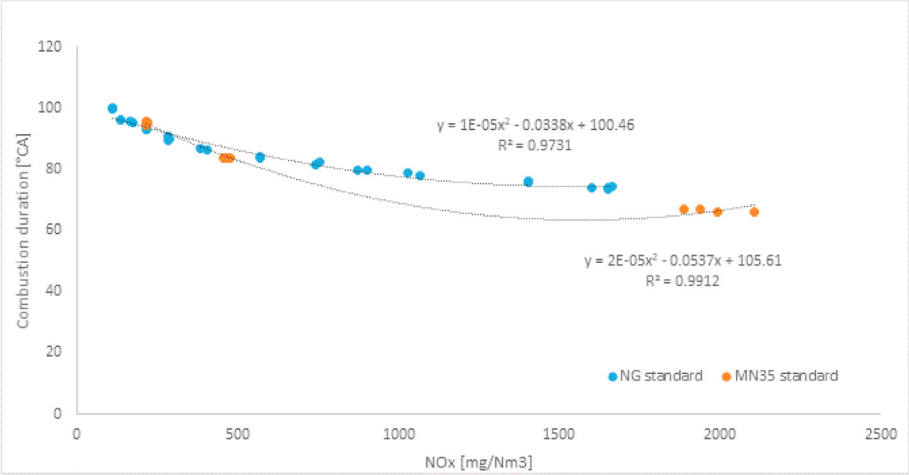


**Fig. 11.** Combustion duration vs $NO_x$.

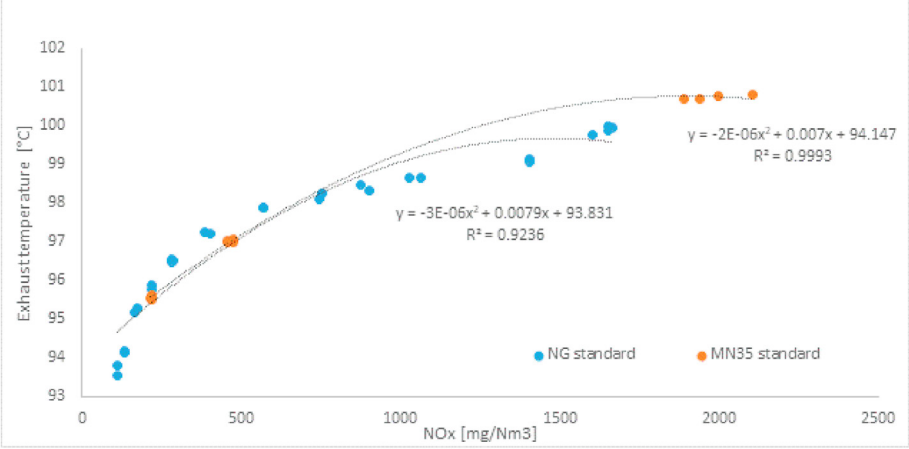


**Fig. 12.** Exhaust temperature vs $NO_x$.

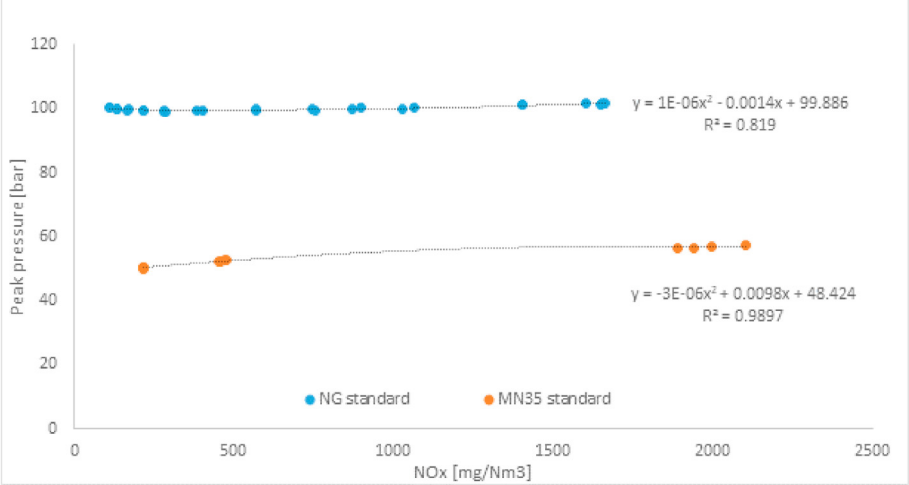


**Fig. 13.** Peak pressure vs $NO_x$.

**Table 3**
Summary of test results.

| Engine | Fuel | BMEP | Timing | Pre-chamber gas flow | Knocking margin | Efficiency at 500mg$NO_x$/$Nm^3$ |
|---|---|---|---|---|---|---|
| SGE-86EM | Natural Gas | 100% | 18 | 1% | 1500 | 100% |
| SGE-86EM | Propane | 46.7% | 14 | 0.60% | 2000 | 85% |

In a nutshell, the summary Table 3 that comprises both experiments results would be as follows:

### 4.2. Techno-economic analysis

Based on the real sampled data obtained from the test bench, its techno-economic implications in internal combustion engine design and operation will be described in this section. In particular, as it was shown in the previous sections, there is a limited combustion performance of engines with MN35 as fuel, especially due to the low output power reached. Despite this, the efficiency results are similar to the natural gas engines values when its power is reduced by 53%. Nonetheless, it is not only a matter of performance related to combustion efficiency or power but the fact this application requires high robustness and reliability. These are parameters that must be analysed at the time to select ICE for APG flaring and will be analysed in this section.

Generally, in order to classify an ICE in terms of a design analysis three main parameters are needed to be clearly defined before the product development starts: Cost, efficiency and robustness/reliability.

1. The first term is directly related to the **profitability** whereas the last two are connected to it in a transversal way. It is evident that the cost of the product has a direct influence on the profitability since the price that is paid for acquiring and installing an engine will vary the initial investment. This would also be applicable to the operational expenses such as the preventive maintenance or the well-known OPEX.
2. On the other hand, **efficiency** also has an influence on the project's payback or profitability. However, it is not as direct as the cost of the engine; in this case it is related to the fuel's price. Obviously, depending on the efficiency of the engine more or less fuel will have to be used for providing the same amount of kW. Annual or daily fuel cost will have a direct influence on the payback.
3. Finally, engine's **reliability** is also closely related to the profitability. Current gas engines are really sensible and have very high protection systems, so it is common to suffer load decreases or emergency stoppages. This is related to corrective maintenance and every time the engine is not generating power, a monetary loss must be considered.

The importance of these three parameters varies according to the demands of the application. The objective of the original 86 EM was to operate with natural gas, so these three variables must be reconsidered for a wellhead gas application.

According to Breaux et al. [28], a user in an urban environment, operating on more expensive, refined fuel and subject to strict site-specific emissions restrictions will have a different set of requirements than a remotely located generator set that is servicing the oil field and operating on relatively inexpensive, un-refined wellhead gas. The former will prefer a high power density, high efficiency solution with little fuel quality tolerance while the latter will sacrifice power density and efficiency for added operating range. This application's specific set of requirements leads to significant engine-to-engine variation in terms of hardware combinations and combustion recipes employed by engine designers.

As explained in the previous paragraph, the efficiency for this application is not determinant since the amount of associated gas (100 $m^3$ per oil ton, on average) is typically produced in amounts too large for this gas to be entirely spent for satisfying the internal needs of the oil field and its surrounding areas. However, it is always important to have the highest efficiency as possible if the reliability and the cost are maintained in desired levels.

APG is a highly demanding fuel due to its high content of heavy hydrocarbons, impurities and fluctuating nature. Furthermore, many oil extraction sites are totally isolated from any interconnected grid and require a very secure system of electricity supply in order to guarantee oil production at the same time. Security of having a stable power supply is no longer an economical variable but becomes the primary objective [29]. If stable power supply is sought and taking into account flare gas characteristics, a simpler, more reliable and durable technology must be implemented in the engine. Hopefully designing a more robust engine is closely linked with a cheaper product.

In conclusion, this application requires robustness, simplicity and reliability while the efficiency shifts to the background unlike the engine developed for gas natural generation. However, it is more than obvious that reducing the price of the product by appealing the need of simplicity of the elements will be a priority.

The natural gas configuration presents a high risk in terms of **reliability and cost** for burning APG that must be reconsidered in the final product configuration. However, per the results it is quite evident that it is not viable to release a product that is designed to stand, for instance, 2 MW with an output power of approximately 1 MW. This is mainly justified by the price of the 2 MW unit which is way more expensive than the 1 MW unit (approximately double). Thus, the €/kW rate would be too high for the market.

As for the **efficiency**, even if it has been stated that this factor is not a driver for this application, it is concluded that the figures shown in the results are **promising as the efficiency with MN35 for the same power is equivalent to the natural gas engine.** This NG engine is currently the combustion engine at the range of 2 MW with the highest figures in efficiency in every manufacturer's portfolio. Thus, at the moment the power is increased we would expect to improve current values.

In contrast, the **robustness** is definitely a challenge and the first danger consists on the utilisation of the injected **pre-chamber**. When the engine is running on APG, which is a gas that contains considerable impurities and has a significant amount heavy hydrocarbons, there are high chances to obstruct the non-return valve that is inside the pre-chamber housing. The blocking of this check valve could lead into a combustion or ignition failure which could end up tripping the unit. Contaminants are a critical preoccupation with APG and if they are not appropriately withdrawn, they can seriously harm the engine.

In the event that it is decided to install an injected pre-chamber, an exhaustive and frequent cleaning schedule must be put in place. It is estimated that every 500 h a preventative maintenance would need to be planned for the right behaviour and functioning of the injected pre-chamber gas supply system. Cleaning check valves takes a while and it is executed using an ultrasound machine and

solvent. This fact would lower the reliability while it would also raise the OPEX.

In addition, if the pre-chamber ignition technology is kept, a process to clean the gas which does not increase the MN would be required to be added on site.

Moreover, this technology needs a pressurised line at 6 bar and an additional gas train. Gas at the aforementioned pressure could be available in the gas supply line, but if this is not the case, the installation of a gas compressor would be a mandatory request converting the full solution more expensive and making the system more intricate and costly.

The decision of not operating with an injected pre-chamber decreases the efficiency for an equivalent detonation margin. Nevertheless, robustness will be a priority against performance or efficiency.

As explained in the previous passage this difficulty could be tackled by implementing a distinct combustion technology. For instance, an unfuelled/unscavenged pre-chamber or even an open chamber could be used. This determination is critical since depending on the chosen ignition technology the rest of the components of the combustion chamber will need to be adapted to it.

It must be considered that the usage of an unscavenged pre-chamber involves a hurdle in renovating the charge of pre-chamber cavity volume. In the injected version, only a little part of the combustion gases in the pre-chamber are voided during the exhaust stroke as the biggest portion is voided during the intake stroke. At the moment the gas is injected in the pre-chamber. It is well-known that the fuel injection no longer exists with the unscavenged pre-chamber ignition system, thus, it converts the charge renovation a challenge that needs to be addressed. The explained setback is faced optimising the pre-chamber nozzles and volume and properly selecting the spark plug insertion into the pre-chamber.

If these obstacles were not enough, the unfuelled pre-chamber design has even more complications from a design perspective. As it was demonstrated, in the NG application on this engine it was not necessary to generate a turbulence inside the combustion chamber since the high velocity torches coming from the fuelled pre-chamber were able to burn all the charge efficiently and achieved around a 100 mm fire radius. Nonetheless, if the previous turbulence levels are maintained with the new pre-chamber, apart from the already mentioned renovation issues, the combustion will probably be incomplete and inefficient (it is estimated that the new fire radius would be 20 mm). That is why the swirl and squish must be increased inside the combustion chamber by modifying the design of the cylinder head and piston.

Being more specific, cylinder head swirl would need to be increased through intake port shape modification and the geometry of the piston would also need to be modified moving from a flat piston to a bowl shaped one to increase the squish.

With the proposed modifications it is expected to have a more reliable and robust engine. Nevertheless, coming back to the high €/kW it may be needed to propose other set of modifications to make this engine for low methane number gases attractive and commercially feasible.

During this study, it is out of interest to define ways to reduce costs in the engine. That is why it was focused on increasing the power output for improving the €/kW ratio. This could be achieved by manipulating engine's performance by introducing **geometric changes.**

Once the piston design is modified to enhance the squish, it is also proposed to be modified to reduce the risks concerning detonations while increasing the power output. This is mainly achieved by reducing the compression ratio (CR) of the piston, but obviously, changing this parameter will have some effects in the engine apart from a considerable loss of efficiency.

Amongst other consequences, further modifications will have to be made in the exhaust manifold and camshaft due to the CR reduction.

There are several reasons why it is suggested to **replace the dry exhaust manifold** to a refrigerated exhaust manifold. The most influential argument is that the exhaust temperatures considerably rise due to various factors. First, as the CR decreases, the exhaust temperature rises to levels that the material cannot tolerate. The main reasons for this event to happen are as follows:

a. When the compression ratio is reduced, the expansion ratio is also reduced. The gas has a smaller expansion and therefore it has less time to cool down during the expansion.
b. Lambda tends to be richer when the CR is reduced. Then, the richer the lambda the higher the temperature in the cylinder and the exhaust.

Secondly, it is important to state that it is always more reliable and safe to **operate with lower temperatures**. Furthermore, the materials used for the dry exhaust manifold are much more expensive than those used in the refrigerated one, of which makes this modification even more interesting. However, water cooled exhaust manifold reduces the temperature at the turbine inlet and therefore the maximum boost achievable on the intake manifold is lower. Because of this fact, the valve timing profile (Otto cycle, Miller cycle …) needs to be carefully defined. Lower boost requires high volumetric efficiency on the intake part and therefore the use of advanced Miller cycles might be limited.

**The Miller cycle used could be replaced by an Otto cycle**. The main supporting reasons for choosing it are based on the price and robustness achieved in the peripherals such as the turbocharger and the exhaust manifold after this decision. Once more, the elimination of complex and sophisticated elements favours the robustness needed for this application.

In a nutshell, the main benefits of designing a new combustion chamber for burning APG mainly resides in the achievement of much higher output power together with a lower cost and an increased reliability. In contrast, the thermal efficiency will decrease while it is not considered to be a critical parameter.

Given the above, Table 4 shows an initial guide proposed combustion configuration for low methane number fuels.

**Table 4**
Modifications for running on APG.

| MN75 | MN35 |
|---|---|
| Fuelled pre-chamber | Unfuelled pre-chamber |
| Cylinder head without swirl | Cylinder head with swirl |
| High compression ratio flat piston | Low compression ratio bowl shaped piston |
| Dry exhaust manifold | Refrigerated exhaust manifold |
| Hard Miller cycle | Otto cycle |
| High efficiency and CR turbocharger | Medium efficiency and CR Turbocharger |

## 5. Conclusions

The most important conclusions drawn after this experimental study show the more adequate modifications in the configuration of a SGE-86EM engine (pre-chamber injected engine) as a consequence of its new performance running on low MN gas. To employ this flare gas as fuel will be a proposal for the reduction of $NO_x$ emissions by replacing flaring. The main conclusions can be summarised and related to the output power and efficiency losses due to the usage of low methane number fuels:

1. The misfire limits are completely different in both gases. Natural gas can work from 0.4% values to 2.0% whereas the margin with APG (MN35) is very much shorter (from 0.3% to 0.8%).
2. Two different trends were obtained with the desired knocking margin: 1500 $mgNO_x/Nm^3$ for natural gas and 2000 $mgNO_x/Nm^3$ for the low methane number application.
3. Thermal efficiency decreases at 500 $mgNO_x/Nm^3$ with APG. Considerable losses were detected with a 35 methane number fuel, where the BMEP reduction was 53.3%. The efficiency loss is justified due to the need of the reduction of the mean effective pressure and the peak pressure. The exhaust temperatures are pretty similar and is not the cause of the efficiency decrease either.
4. An ignition technology modification would be recommendable to address the low methane number market. This would enhance the reliability and shorten CAPEX and OPEX.

Finally, based on the obtained results, and despite the fact that this proposal is clearly adequate to replace flaring gases, it is concluded that the traditional natural gas combustion chamber configuration seems to be inadequate in order to face this market. Therefore, subsequent tests will have to be carried out in order to improve the design a final combustion chamber configuration to meet the demand of the current needs.

## Credit author statement

Ander Ruiz, Iñaki Loroño, Iñigo Oregui, José A. Orosa: Conceptualization, Methodology, Software. Ander Ruiz, José A. Orosa Data curation, Writing – original draft preparation. Ander Ruiz, Iñaki Loroño, Iñigo Oregui: Visualization, Investigation. Iñaki Loroño, Iñigo Oregui, José A. Orosa: Supervision. Ander Ruiz, José A. Orosa: Writing- Reviewing and Editing,

## Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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## Definitions/Abbreviations

*AFR:* Air Fuel Ratio
*APG:* Associated Petroleum Gas
*BMEP:* Brake Mean Effective pressure
*BTDC:* Before Top Dead Centre
*CAPEX:* Capital Expenditures
*CHP:* Combined Heat and Power
*COV:* Covariance
*$CO_2$:* Carbon dioxide
*CR:* Compression Ratio
*GGFR:* Global Gas Flaring Reduction
*GTL::* Gas-To-Liquid
*$H_2S$:* Hydrogen Sulphide
*ICE:* Internal Combustion Engine
*LFL::* Lower Flammability Limit
*LHV:* Lower Heating Value
*LNG:* Liquefied Natural Gas
*LPG:* Liquefied Petroleum Gas
*MCE:* Multi Cylinder Engine
*MN:* Methane Number
*ND:* Non-Derated
*NG:* Natural Gas
*NOx:* Nitrogen Oxides
*O□G:* Oil and Gas
*OPEX:* Operating Expenses
*PLC:* Programmable Logic Controller
*R□D:* Research and Development
*SCE:* Single Cylinder Engine
*SOx:* Sulphur Oxides
*UDC:* University of Coruña
*UFL::* Upper flammability limit
*VOC:* Volatile Organic Compounds