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An Investigation of Compressed Natural Gas Engine for Nitrogen Oxides Reduction

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Abstract: Problem statement: This study describes the use of Reformer Gas (RG) to alter NO_x emission in a CNG-fueled HCCI engine. Comparison with diesel, natural gas has a very high octane number (≈120) and high auto-ignition temperature (≈600°C). Composed mostly of methane, natural gas is the only common fuel to manifest relatively pure, single-stage combustion. Other fuels have stronger low-temperature reaction and the required entropy for main stage combustion can be obtained from the low temperature heat release as a result of compression to moderate pressure and temperature. In deviation, the methane molecule resists destruction by free radicals and produces negligible heat release at low temperature. In consequences, in CNG-fueled HCCI engines the activation energy required for auto-ignition must be obtained by extreme levels of charge heating and compression. This causes inherently to a high rate of heat release. HCCI operation with pure CNG fuel was attained but not really practical due to very high NO_x production. While HCCI operation is usually described as a low NO_x technique, the knocking behavior when running with pure CNG raised the peak combustion temperature to a value well above normal combustion and the critical Zeldovich NO_x production threshold, giving very high indicated NO_x emissions. **Approach:** One approach to improving these properties is to convert part of the base CNG fuel to Reformer Gas (RG). In this study, modified COMET engine was operated in HCCI mode using a mixture of CNG fuel and simulated RG (75% H_2 and 25% CO) can be produced on-board from CNG using low current and non-thermal plasma boosted fuel converter. **Results:** This study shows that despite of having various RG mass fractions, λ was the dominant factor in reducing NO_x production and increasing RG mass fraction had only a small effect on increasing NO_x. This disconnect between the overall equivalence ratio and RG fraction shows that the real benefit of the RG blending was to enable lean (high) operation. Higher λ also effectively reduced maximum pressure and maximum pressure rate. **Conclusion:** Note that due to the low achievable power levels, the NO_x emissions continue to be high and further combustion enhancements and more controlled combustion would be needed to make the CNG-fuelled HCCI engine practical.

Key words: Homogeneous charge compression ignition, compressed natural gas, reformer gas, air/fuel ratio, overall equivalence, RG mass, RG fraction, HCCI mode, COMET engine, NO_x production

INTRODUCTION

The internal combustion engine the vital to the current society. Without the transportation performed by the millions of vehicles on the road and at sea we would not have achieved the living standard of modern life. We have two types of internal combustion engines such as spark ignition engine and compression ignition engine. Both have their merits. The SI engine is a preferably simple product and hence has a lower first cost. This engine type in addition made very clean as the Three-Way Catalyst (TWC) is effective for exhaust after treatment. The problem with the a park ignition engine is the poor part load efficiency due to large losses during gas exchange and low combustion and thermo dynamical efficiency. The compression ignition engine is much more fuel efficient and hence the existing choice in applications where fuel cost is more important than initial cost. The problem with the CI engine is the emissions of nitrogen oxides and particulate matter. The treatment to reduce nitrogen oxides and PM is costly and still not generally available on the market. The natural choice of ideal combination would be to find an engine type with the high efficiency

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of the CI engine and the very low emissions of the spark ignition engine engine with three-way catalyst. One such new concept is named Homogeneous Charge Compression Ignition (HCCI). Homogeneous Charge Compression Ignition engine is a new concept for future power trains which will provide improved fuel efficiency and lower emissions at the same time. It is based on the concept of compression ignition of fuel-air mixtures due to reaching auto ignition temperature. However, there are two critical problems associated with the HCCI engines: control of the autoignition timing and the combustion rate. There are a number of strategies that have been currently investigated to address the above two critical problems, such as a variable compression ratio (Haraldsson *et al*., 2002; Sjoberg and Dec, 2003; Diaz and Prasad, 2010; Al-Khairi *et al*., 2011; Risberg *et al*., 2006) variable valve timing (Kaahaaina *et al*., 2001) variable intake charge temperature or hot exhaust gas recirculation study (Christensen *et al*., 1999). The exhaust gas injection in the intake port changes the intake temperature thereby giving a good control on the combustion phasing inside the HCCI engine. However, EGR consist of many gases such as oxides of carbon monoxide, carbon dioxide, nitrogen unburned hydrocarbon and oxides of nitrogen (NO_2, NO). Recent studies have indicated that nitric oxide can have an important effect on the kinetics of the autoignition of HC inside HCCI engines through NO-promoted production of OH radical, HO_2+NO= OH+NO_2 (Kalateh and Ghazikhani, 2012) The presence of NO in the recirculated exhaust gases in HCCI engines is now perceived as a potentially a promising concept for controlling the combustion phasing inside these engines. To achieve this, a clear understanding of the in-cylinder nitrogen oxide formation inside the HCCI engine is immediately required, which depends on the in-cylinder combustion characteristics. Experimental quantification of the in cylinder nitrogen oxide formation is time consuming and technically challenging.

MATERIALS AND METHODS

The Reformer Gas is a mixture of light gases dominated by Hydrogen and Carbon Monoxide and can be produced from Compressed Natural Gas using low current and non-thermal plasma boosted fuel converter. In the COMET engine, HCCI operation on pure Compressed Natural Gas fuel was achieved but not really practical due to very high Nitrogen Oxide production. While HCCI operation is generally characterized as a low Nitrogen Oxides technique, the marginal knocking behavior when running with pure Compressed Natural Gas raised the peak cylinder temperature to a value well above the critical Zeldovich Nitrogen Oxides production mechanism threshold, giving very high indicated Nitrogen Oxides emissions. HCCI operation with leaner mixtures, enabled by Reformer Gas blending, significantly reduced NO_x production. The formation of Nitrogen Monoxide and Nitrogen Dioxide can be divided into thermal route, prompt route, N_2O route and fuel-bound nitrogen route (Warnatz *et al*., 2006). The major NO_x formation route in IC engine combustion is the thermal route (Heywood, 1988). The thermal NO route is the major constituent to the NOx emission and can be described with the following three elementary reactions called as extended Zeldovich mechanism Eq. 1-3:

$$O+N_2 \rightarrow NO+N \quad (1)$$

$$N + O_2 \rightarrow NO+O \quad (2)$$

$$N+OH \rightarrow NO+H \quad (3)$$

Reaction 1 has very high activation energy and is the rate limiting step. The triple bond of the Nitrogen molecule is strong, the consequence of this causes the reaction rate is slow unless the temperature is high. When assuming quasi-steady state for N concentration, the rate of NO production may be described as Eq. 4:

$$\frac{d[NO]}{dt} = 2k_1[O][N_2] \quad (4)$$

The NO_x production may thus be reduced by reducing the concentration of Oxygen or Nitrogen, or reducing the rate coefficient k_1 by reducing the temperature. The rate coefficient k_1 is considered insignificant at temperatures less than 1700 k (Warnatz *et al*., 2006).

A common European standard for emission legislations were introduced in 1992 with the EURO 1 standard. Since then, the EURO 2, 3 and 4 has been put into force of implementation with ever more stringent requirements (Warnatz *et al*., 2006). Table 1 shows the Emission standards for heavy duty diesel and gas engines for the Transient Test Cycle (Heywood, 1988).

In the COMET engine, HCCI operation on pure Compressed Natural Gas fuel was achieved but not really practical due to very high Nitrogen Oxide production.

Experimental setup: All experiments were conducted on a modified COMET engine to operate in HCCI mode using CNG fuel. Table 2 summarizes the engine specifications for the current experiment. The Schematic diagram of Experimental setup is as shown in Fig. 1.

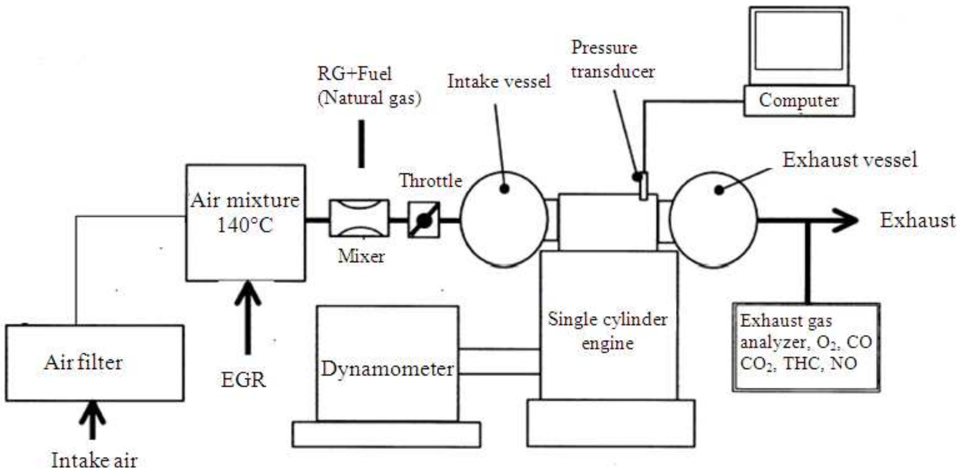

Fig. 1: Schematic diagram of Experimental setup (Satoshi S, SAE, 2001-01-1034)

Table 1: Emission standards for heavy duty diesel and gas engines (motor vehicles over 3500 kg as maximum laden mass) for the transient test cycle (Heywood, 1988)

Emission standards for diesel and gas engines, ETC test, g/kWh							
Tier	Date	Test	CO	NMHC	CH_4^a	NO_x	PM
Euro III	1999.10 EEVs only	ETC	3.00	0.40	0.65	2.0	0.02
	2000.10	ETC	5.45	0.78	1.60	5.0	0.16
							0.21
Euro IV	2005.10		4.00	0.55	1.10	3.5	0.03
Euro V	2008.10		4.00	0.55	1.10	2.0	0.03

A: For natural gas engines only, B: Not applicable for gas fueled engines at the year 2000 and 2005 stages, C: For engines of less than 0.75 dm^3 swept volume per cylinder and a rated power speed of more than 3000 min^{-1} (Heywood, 1988)

Table 2: Experimental apparatus and fuels

Engine make	Comet
Engine type	Four stroke single cylinder engine
Rated power output	3.5 kW at a speed of 1500 rpm
Bore diameter	30 mm
Stroke length	110 mm
Throttle	Fully open
Main fuel	CNG
Additive fuel	RG
CR	17

RESULTS

Table 3-7 shows the results obtained from experiments for different operating conditions.

DISCUSSION

Effect of mixture strength: λ: As mentioned already, λ represents the total air/fuel ratio considering both the CNG and RG as a combined fuel.

In HCCI combustion, the total chemical energy inside the cylinder plays the major role.

Figure 2 indicates that despite of having various RG mass fractions, λ was the dominant factor in reducing NO_x production and increasing RG mass fraction had only a small effect on increasing NO_x.

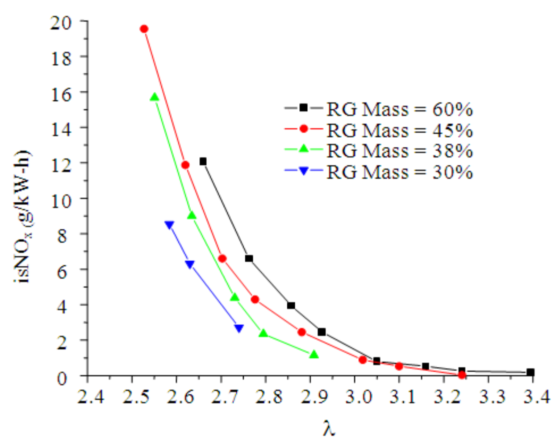

Fig. 2: Effect of relative air fuel ratio on is NO_x at constant RG mass fraction lines.

Figure 2 shows that the governing factor to decrease NO_x is λ, especially when looking at λ greater than 2.9. This disconnect between the overall equivalence ratio and RG fraction shows that the real benefit of the RG blending was to enable lean (high λ) operation, (which is similar to the benefits seen for hydrogen and RG blending for non-HCCI engines).

Table 3: Effect of Relative air fuel ratio on is NO_x at constant RG mass fraction

Relative air fuel ratio, λ	Is NO_x Emission at RG = 60%, in (kg/kW-h)	is NO_x Emission at RG = 45%, in (kg/kW-h)	is NO_x Emission at RG = 38%, in (kg/kW-h)	is NO_x Emission at RG = 30%, in (kg/kW-h)
2.76221	6.60716	11.871220	9.003120	6.316180
2.85822	3.93113	6.601210	4.382150	2.716711
2.92627	2.45150	4.291410	2.351400	2.301140
3.0418	0.80109	2.415200	1.158120	1.101820
3.1519	0.53132	0.891060	0.800160	0.790160
3.24018	0.21630	0.521780	0.501780	0.427180
3.39514	0.18148	0.035516	0.030516	0.025516

Table 4: Effect of relative air-fuel ratio of maximum pressure and maximum pressure rate at constant RG mass fractions

Relative air fuel ratio, λ	P_{max} at RG = 45% In bar	$\left(\frac{dp}{d\theta}\right)_{max}$ at RG = 45% In bar	P_{max} at RG = 30% In bar	$\left(\frac{dp}{d\theta}\right)_{max}$ at RG = 30% In bar
2.76211	77.72815	77.952230	76.91253	75.42233
2.85182	77.52110	77.298180	73.97526	73.01128
2.92267	77.53713	75.531730	67.40224	69.32167
3.04298	73.11115	73.101990		
3.1529	70.69422	71.501820		
3.24108	68.56329	70.042254		
3.39514	65.85129	68.662740		

Table 5: is NO_x increase with an increasing RG mass fraction of constant relative air fuel ratio

RG% λ=2.5	Is NO_x Emission at λ=2. 5 in (kg/kW-h)	RG% at at λ=2.6	is NO_x Emission λ=2. 6 in (kg/kW-h)	RG% at λ=2.7	is NO_x Emission at λ=2. 7 in (kg/kW-h)	RG% at λ=2.8	isNO_x Emission at λ=2. 8 in (kg/kW-h)
16	7.6887	30	8.58019	30	2.9425	37	2.63320
23	11.8026	38	9.11722	38	4.57148	42	2.62280
30	16.15721	41	10.38528	46	4.44144	46	2.62180
39	15.6622	45	11.95628	53	6.00837	53	3.28114
46	19.53233	52	13.40229	59	12.06137	60	4000000

Table 6: Effect of RG on maximum pressure rate at constant relative air fuel ratio

RG% at λ=2.5	$\left(\frac{dp}{d\theta}\right)_{max}$ Timing (CAD, ATDC)	RG% at λ=2.6	$\left(\frac{dp}{d\theta}\right)_{max}$ Timing (CAD, ATDC)	RG% at λ=2.7	$\left(\frac{dp}{d\theta}\right)_{max}$ Timing (CAD, ATDC)	RG% at λ=2.8	$\left(\frac{dp}{d\theta}\right)_{max}$ Timing (CAD, ATDC)
15	13.3475	34	15.4059	30	12.1584	34	9.6236
22	15.0894	38	15.8812	38	14.1386	37	11.1287
30	17.3862	42	16.9109	45	14.8515	45	12.0787
38	18.0939	45	17.703	52	15.6436	52	13.7426
46	19.76224	52	18.4158	59	18.495	60	15.4059

Table 7: Effect of RG on maximum pressure at constant relative air fuel ratio

RG% at λ = 2.5	P_{max} In bar	RG% at λ = 2.6	P_{max} In bar	RG% at λ = 2.7	P_{max} In bar	RG% at λ = 2.8	P_{max} In bar
15	72.8985	34	74.888	30	69.8297	30	63.238
22	75.5058	38	75.1576	38	72.5689	38	67.3999
30	78.3729	42	75.675	45	72.4535	45	68.1964
38	78.1336	45	75.9556	53	73.2535	53	69.4976
46	78.5433	52	76.2255	59	74.5638	59	71.3389

Higher λ effectively reduced the maximum pressure and maximum pressure rate. The input energy in a lean mixture is lower, so a lower combustion temperature, lower maximum pressure (P_{max}) and the lower maximum pressure rate $\left(\left(\frac{dp}{d\theta}\right)_{max}\right)$ were expected as shown in Fig. 3. P_{max} was mostly dominated by λ rather than RG mass fractions. At very lean conditions, the influence of RG mass fraction on P_{max} increased, presumably by ensuring combustion of the most dilute zones in the combustion chamber. $\left(\left(\frac{dp}{d\theta}\right)_{max}\right)$ is a strong function of both λ and RG mass fraction as indicated in Fig. 3.

Effect of RG mass fraction: RG addition was found to be an effective means of expanding the lean boundary of the HCCI operating window. As mentioned earlier, the operating region of CNG for this engine is not a practical operating window. For the knock boundary, a limit of $\left(\left(\frac{dp}{d\theta}\right)_{max}\right) = 20\frac{bar}{CAD}$ was chosen to represent a medium knocking condition, while in other experimental HCCI studies, $10\frac{bar}{CAD}$ is usually considered as a boundary (Iida *et al*., 2001).

Also, the lean operating limit measured in this study is not a misfiring boundary and was defined as the maximum usable λ without a drop in engine speed. Increasing the fuel's RG mass fraction expanded the operating window significantly on the lean side, while the pure CNG-fueled HCCI engine could operate at λ ≈ 2, blending a fuel with 60% RG increased the lean operation range to λ ≈ 3.5. Hence, the mechanism of all the positive λ effect on operating parameters such as decreasing maximum pressure, maximum pressure rate and is NO_x (in Fig. 2) RG blending to achieve HCCI combustion without knock substantially decreased NO_x emissions as indicated in Fig. 4. Note that due to the low achievable power levels, the specific NO_x levels continue to be high and further combustion enhancements would be needed to make the CNG-fueled HCCI engine practical. RG addition has a secondary impact on NO_x emissions. Looking at any of the constant λ lines in Fig. 4 indicates that displacing CNG with RG at a constant air/fuel ratio actually increases NO_x.

The observed trends of engine and combustion parameters resulting from RG additions in a CNG fueled HCCI engine can be further investigated using the cylinder pressure traces collected in this study. Figure 5 and 6 shows that RG increased the maximum cylinder pressure and maximum cylinder pressure rate substantially. With NO_x, λ has the dominant effect. Hence, expanding the operating region towards leaner mixture can reduce P_{max} and $\left(\left(\frac{dp}{d\theta}\right)_{max}\right)$, while, for a given λ increasing RG mass fraction increased the P_{max} and $\left(\left(\frac{dp}{d\theta}\right)_{max}\right)$. If we considered a reasonably acceptable noise level of $\left(\left(\frac{dp}{d\theta}\right)_{max}\right) = 10\frac{bar}{CAD}$, the only operating points would be at λ = 2.8 with RG mass fraction less than 35%. Also, increasing the RG mass fraction advanced P_{max} and $\left(\left(\frac{dp}{d\theta}\right)_{max}\right)$ timings earlier in the combustion cycle.

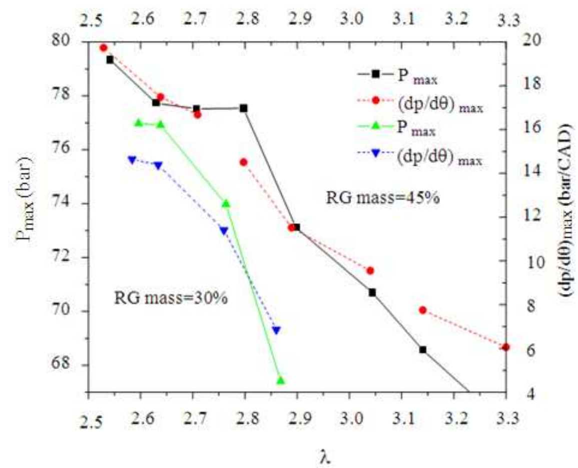

Fig. 3: Effect of relative air fuel ratio on is O_x at constant RG mass fraction lines. Effect of relative air fuel ratio of maximum pressure and maximum pressure rate at constant RG mass fraction lines

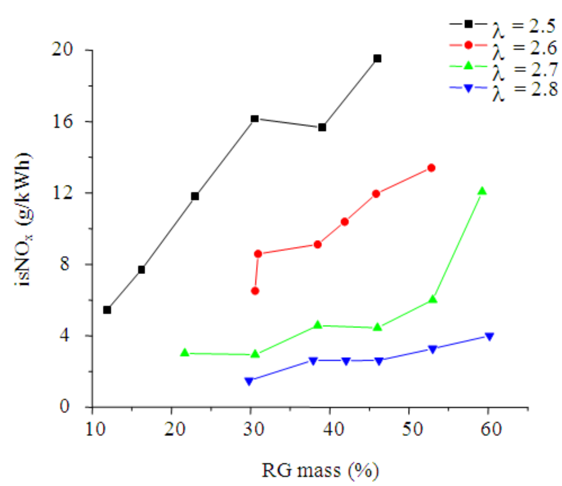

Fig. 4: is NO_x increase with increasing RG mass fraction at constant Effect of relative air fuel ratio on is NO_x at constant RG mass fraction lines

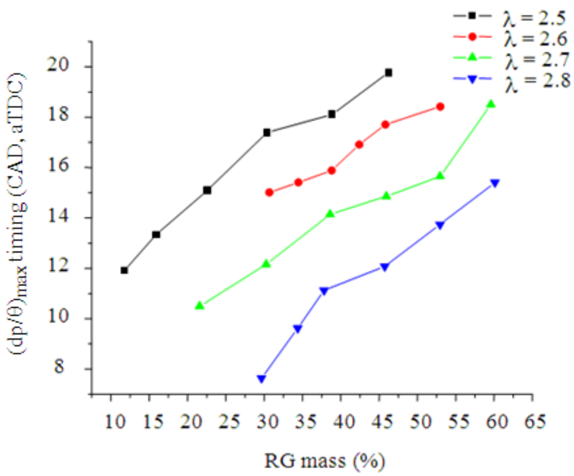

Fig. 5: Effect of RG on the maximum pressure rate at constant Effect of relative air fuel ratio on is Knox at constant RG mass fraction lines

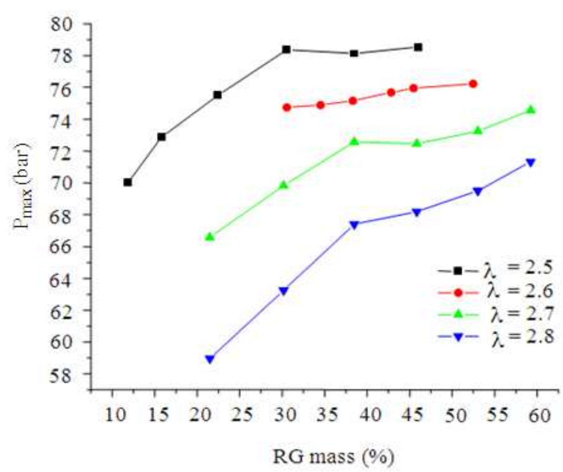

Fig. 6: Effect of RG on maximum pressure at constant Effect of relative air fuel ratio on is NO_x at constant RG mass fraction lines

The effect of RG addition on combustion onset is a complicated thermal/chemical phenomenon that cannot be explained just by looking at engine operating parameters. The base fuel characteristics play an important role. For example, in another study on an HCCI engine fueled with normal heptanes (Machrafi *et al*., 2008) showed that increasing RG fraction actually retarded combustion timing. In that case, adding RG, which is a higher octane component than the base fuel, retarded ignition. In contrast, RG added to a CNG-fueled engine is a lower octane component than the base fuel and it advanced ignition timing. In this case, the combustion timing change could be a result of added H_2O_2 production before the main stage of methane auto-ignition.

The tendency of RG to shift the allowable operating range towards leaner mixtures advances the peak pressure timing. Also, despite the elimination of audible knock, the advance in peak cylinder pressure timing generally resulted in higher peak pressures in RG addition, (Fig. 6) Overall, both the peak pressure timing and peak pressure were highly correlated with RG fraction and λ.

CONCLUSION

A CNG-fueled COMET engine was modified to operate at high compression ratios and high intake temperature enabling to attain HCCI combustion. With CNG fuel the operating range was very limited between both boundaries marked by heavy knock and misfire. The attainable engine speed range was low. Overall, the COMET engine appeared to be poorly suited for HCCI combustion with natural gas fuel. As a result, HCCI operation on pure CNG was considered unsuccessful because of high indicated specific NO_x, high cyclic variation and low efficiency. A Partial Reformar Gas replacement was found to be beneficial for expanding the operating range of fuel rich side, reducing knock severity and reducing indicated specific NO_x emission, maximum peak cylinder pressure and rate of pressure rise with respect to crank angle which could not be achieved on pure CNG fueling. However, considering the situation of constant relative air fuel ratio the peak cylinder pressure, rate of pressure rise with respect to crank angle and NO_x levels were increased substantially while replacing Compressed Natural Gas with Reformar Gas. This implies that the best quantity of RG is the minimum necessary to enable and enhance operation at the desirable operating point.

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