Int.J. Applied Thermodynamics,
Vol.5 (No.4), pp.169-175, December-2002
ISSN 1301-9724

Lean-Burn Cogeneration Biogas Engine with Unscavenged Combustion Prechamber: Comparison with Natural Gas

Anne ROUBAUD*, Roger RÖTHLISBERGER and Daniel FAVRAT
Laboratory of Industrial Energy Systems (LENI)
Swiss Federal Institute of Technology (EPFL)
CH-1015 Lausanne-Switzerland
Tel: +41 (0) 21 693 59 68, Fax: +41 (0) 21 693 35 02
E-mail: anne.roubaud@epfl.ch

Abstract

Gaseous fuels produced, for example, by waste or agricultural by-products fermentation (biogas) can be burned in-situ by cogeneration systems like spark-ignition internal combustion engines. However, the more and more stringent legislation for exhaust gas emissions requires improvement of the combustion process particularly when catalytic after treatment is not reliable as in the case of sewage or landfill biogas. The system proposed in this paper is the use of an unscavenged combustion prechamber instead of direct ignition on a turbocharged 6 cylinder 150 kW gas engine. This prechamber is used for operation with a simulated biogas (40% CO_2 in natural gas). The results show that, compared to natural gas operation for the same rated power output of 150 kW and the same NOx emissions, the CO emissions are reduced by 15% and the HC emissions at least by 8%. Efficiencies higher than 36% are achieved which is very promising and the lower CO emissions give a margin to consider an increase of compression ratio.

Key words: *combustion prechamber, cogeneration engine, biogas, natural gas, efficiency, emissions*

1. Introduction

As humankind becomes more and more concerned with sustainable development, biomass represents a serious alternative to substitute part of fossil fuels in decentralized power plants. Among the available technologies, cogeneration gas engines can burn most biogas fuels such as those produced in sewage plants. The benefit in this case is : the CO_2 balance is usually considered as neutral and this system valorizes urban waste or agricultural by-products. Gas engines have proved able to achieve an efficient conversion of the primary energy in numerous applications with natural gas. However, emissions can be a limitation. With a limit of 250 mg/Nm^3 (400 in the case of biogas operation) for NO_X and 650 mg/Nm^3 for CO (concentrations referred to combustion gases with 5% residual O_2 (Le Conseil Fédéral Suisse 1999), Switzerland has the most stringent exhaust gas emissions legislation in Europe for stationary combustion engines. Several engine operating modes enable the attaining of these requirements, but they all in practice rely on a catalytic exhaust gas after treatment for natural gas operation. The first one is based on the use of a stoichiometric mixture and a three-way catalyst, and another one is based on the lean burn mode with oxidation catalyst. Both modes were investigated previously (Röthlisberger et al. 1998 and 2000). A third solution is to use exhaust gas recirculation (EGR) and a three-way catalyst (Nellen and Boulouchos 2000).

However, solutions relying on a catalytic after-treatment are not adapted to biogas since it contains significant quantities of sulphur and heavy metals. Those compounds can form a deposit on the combustion chamber walls or in the catalyst and then will rapidly deactivate it. A catalytic after-treatment is therefore not reliable in the case of sewage or landfill biogas. Upstream filtration technologies are being developed but at present they are cumbersome, unreliable and costly. The literature is rather limited when it comes to the analysis of the influence of the biogas composition on the

* Author to whom all correspondence should be addressed.

behavior of spark ignition engines considering both efficiency and

TABLE I. MAIN ENGINE SPECIFICATIONS

Manufacturer	Liebherr	Nozzle orifices of the prechamber	4 orifices of 2,12 mm
Type	G 926 TI	Prechamber volume	4540 mm^3
Number of Cylinders	6	Volumetric compression ratio	12,0
Bore	122 mm	Turbocharger	KKK K27 3371 OLAKB
Stroke	142 mm	Ignition system	Fairbanks Morse IQ 250
Conrod length	228 mm	Spark plugs	Bosch W6DC
Total swept volume	9.96 L	Number of valves	2

emissions (Henham 1998, Bucksch et al. 1999, Muller 1995, Stone et al. 1993 and Huang et al. 1998). The main way so far to reduce emissions is to decrease the compression ratio, which results in a lower efficiency than normally achievable. Therefore, new solutions are to be found at the level of the combustion process to increase efficiency. Prechamber ignition is the alternative pursued in this work. This combustion mode has been proved to reduce the CO and unburned hydrocarbons emissions respectively by 40, 55% and NOx emissions below 250 mg/Nm^3 with only a slight reduction in fuel conversion efficiency for natural gas operation of a turbocharged 6 cylinder 150 kW spark-ignited gas engine (Röthlisberger 2003a, 2003b, 2002a, and 2002b).

2. Experimental Setup

The engine used is a 6-cylinder in-line Liebherr heavy-duty diesel engine type D 926 TI converted for gas fuel operation with spark ignition. The engine is equipped with specially modified cylinder heads and piston geometry to reach a volumetric compression ratio of 12.0. The piston geometry has specially been designed to enhance turbulence in the combustion chamber (Nellen at al. 2000). The engine is turbocharged and intercooled. Cylinder liners allow reduction of the crevice volume at the level of the cylinder head gasket and hence CO and HC exhaust gas emissions.

The cylinder head is fitted with small water-cooled combustion prechambers having a volume corresponding to 3% of the cylinder compression volume (*Figure 1*). Those prechambers have been optimized during the PhD thesis of R. Röthlisberger et al. (2001).

The prechambers used are unscavenged. At exhaust valve closure the prechamber is filled with burnt gases. During compression, a fresh fuel air blend enters the prechamber and mixes with the residual gases. At spark timing the prechamber mixture is ignited and the hot gases from the combustion create strong gas jets in the main chamber. Those jets ignite the main cylinder charge at multiple locations. The main engine specifications are given in TABLE I.

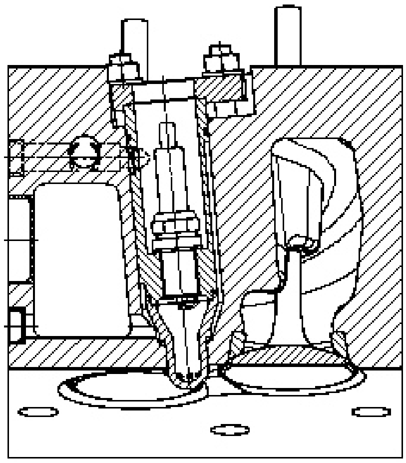

Figure 1. Combustion prechamber integrated in the cylinder head

The test bed is equipped with analyzers of O_2 (paramagnetism), CO_2 (IR), CO (IR), NOx (CLD) and HC (FID). The engine is fully instrumented to measure all needed flows, temperatures, pressures, combustion air humidity, torque and rotation speed through the dynamometer to control precisely the experimental conditions. The engine is also equipped for pressure indication in the prechamber and the main chamber of cylinder 1.

TABLE II. NATURAL GAS COMPOSITION

	Volumetric %
Nitrogen	2.036
CO2	0.739
Methane	91.799
Ethane	3.983
Propane	1.009
i-butane	0.159
n-butane	0.175
i-pentane	0.038
n-pentane	0.032

HC with more than 6 carbons	0.031

This testing configuration has been fully described in Röthlisberger et al. (2001). The test bed fuel and air alimentation was modified to use a natural gas/CO_2 blend as fuel. As CO_2 is stored in liquefied form, its pressure is reduced after preheating only and the flow is controlled by a mass flow controller before mixing with the combustion air. For the tests reported here, the CO_2 volumetric flow is adjusted to 40% of the total CO_2+natural gas flow. The composition of the natural gas used in all the experiments reported here is given in TABLE II. A compressed natural gas storage, which was large enough to keep the same natural gas composition for all tests presented here, is used,.

3. Test Conditions

The main experimental conditions are summarized in TABLE III. The engine speed is set to 1500 rpm, which corresponds to cogeneration engine specifications, and the engine rated brake power output is 150 kW.

Exhaust gas emissions are expressed in mg/Nm^3 at normal conditions and corrected for 0% humidity and 5% residual oxygen (Swiss standard). The relative air to fuel ratio is calculated on the basis of the measured fuel and air mass flows.

TABLE III. GENERAL EXPERIMENTAL CONDITIONS

Crankshaft rotation speed	1500 ± 5 rpm
Rated brake mean effective pressure	12 ± 0.1 bar
Rated brake power output	150 ± 1.3 kW
Intake air pressure	960 ± 5 mbar
Intake air temperature	23 ± 2°C
Intake air relative humidity	50 ± 0.5 %
Exhaust gas pressure after turbocharger	1050 ± 5 mbar
CO_2 percentage in fuel	40 ± 0.5 %

4. Results

The objectives of this work are to compare the benefits, in terms of engine efficiency, stability and exhaust gas emissions, of using a prechamber with either synthetic biogas or natural gas.

4.1. Natural gas performances

A new set of tests with natural gas has been made both with direct ignition and prechamber ignition. For these preliminary tests, the spark timing was held constant at 8 CA_{BTDC} for prechamber operation and 27 CA_{BTDC} for the direct ignition mode.

The relative air to fuel ratio was $1.81<\lambda<1.84$ for direct ignition and $1.64<\lambda<1.68$ for prechamber ignition. The pressure traces (*Figure 2*) show how the two ignition modes influence the combustion.

Due to the important delay in spark timing the peak pressure is lower with prechamber ignition. The first pressure peak corresponds to the end of the compression and the second one results from the combustion in the cylinder after ignition by the hot gas jets issuing from the prechamber. An earlier and much higher combustion induced pressure peak is observed with direct ignition.

This difference can also be observed on the heat release rate cycle (*Figure 3*) where a sharper raise of the heat release is observed with the prechamber with a higher rate of combustion.

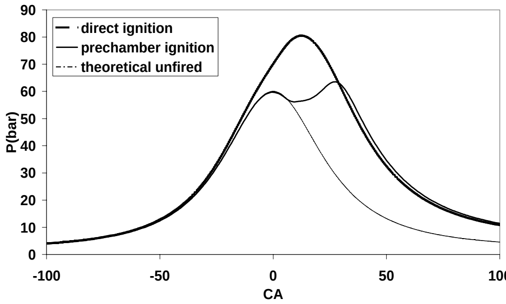

Figure 2. Comparison between direct ignition and prechamber ignition: main chamber pressure cycle for natural gas.

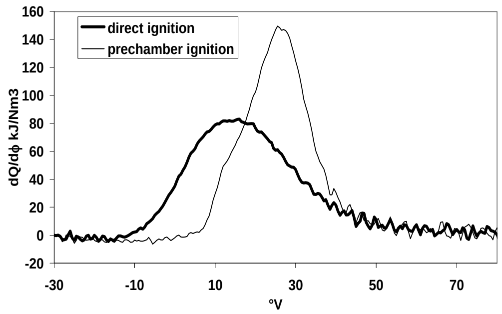

Figure 3. Comparison between direct ignition and prechamber ignition: main chamber heat release rate cycle for natural gas.

This more rapid combustion leads to a more stable operation and then to a lower coefficient of variance of pmi (*Figure 4*). But the late spark timing ignition in the prechamber shifts the combustion process in the expansion phase. Following this, the maximum peak pressure is lower but all the heat released produces a positive mechanical work.

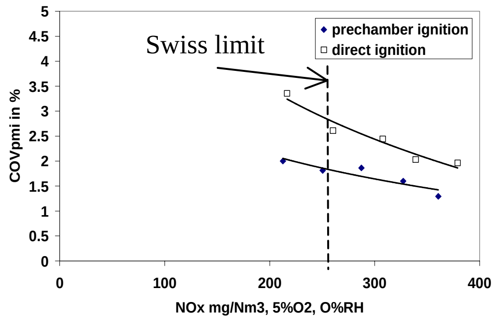

Figure 4. Coefficient of variance of pmi in % for pure natural gas

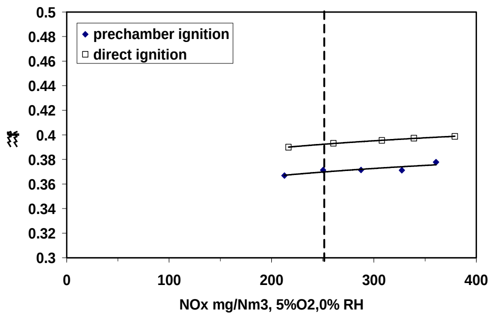

Figure 5. Fuel conversion efficiency for natural gas

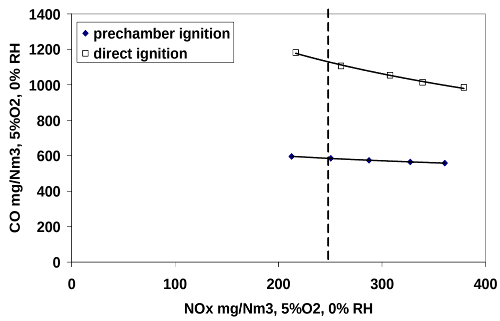

Figure 6. CO emissions for natural gas

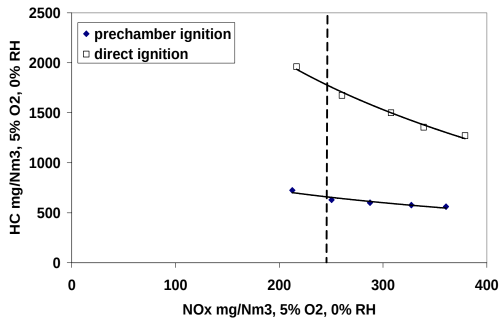

Figure 7. HC emissions for natural gas

The balance of those two phenomena explains the lower fuel conversion efficiency of 2% (*Figure 5*). The prechamber allows a reduction of at least 40% of the CO emissions (*Figure 6*) and 55% for the HC emissions (*Figure 7*) for the same NOx emissions. On the one hand, the hot gas jets issuing from the prechamber to ignite the main combustion chamber create a more homogeneous and a faster combustion so that partial oxidation has less time to occur. On the other hand, less fuel air mixture is going into crevices volumes thanks to a lower pressure in the cylinder.

4.2. Synthetic biogas performances compared to natural gas

The first set of experiments made with synthetic biogas was to see the lower limit in stability and in NOx emissions with this fuel. The spark timing in the prechamber was 8 CA_{BTDC} and $1.54<\lambda<1.55$. The second set of experiments objective was to observe the influence of the spark timing on the performances from 8 to 13 CA_{BTDC}.

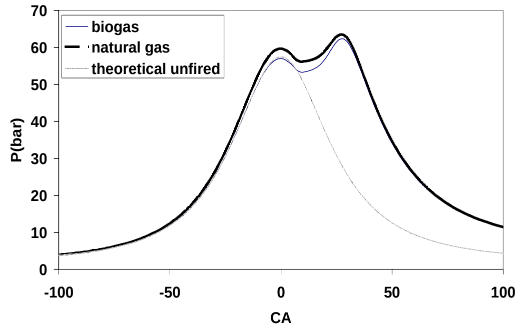

Figure 8. Comparison between natural gas and synthetic biogas pressure cycle

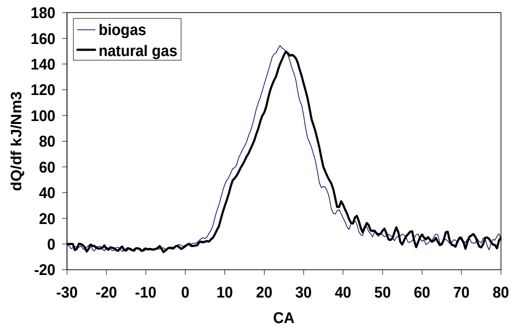

Figure 9. Comparison between natural gas and synthetic biogas heat release rate cycles

The main cylinder pressure trace (*Figure 8*) shows that the operation with synthetic biogas is similar to the one with natural gas but with a lower peak pressure due to a higher heat capacity of the gas mixture with the CO_2. The heat release rate is very similar to the one for natural gas (*Figure 9*). The prechamber seems to operate in the same way with synthetic biogas as for natural gas.

As the combustion process is the same, the engine stability in terms of variance coefficient is similar to the natural gas operation (*Figure 10*).

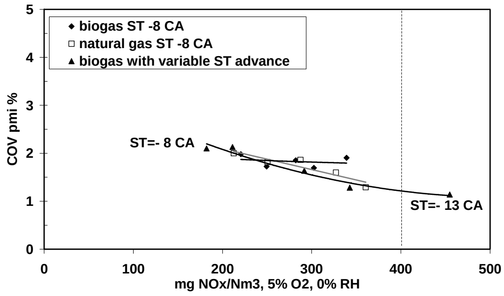

Figure 10. Comparison between natural gas and synthetic biogas, coefficient of variance of pmi

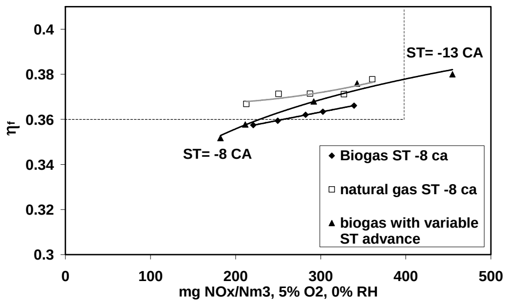

Figure 11. Comparison between natural gas and synthetic biogas, fuel conversion efficiency

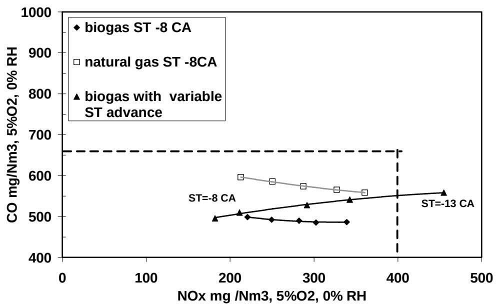

Figure 12. Comparison between natural gas and synthetic biogas, CO emissions

Due to the lower peak pressure, the fuel conversion efficiency is 1% lower at constant spark timing (*Figure 11*). But on the other hand the CO emissions (*Figure 12*) and the HC emissions (*Figure 13*) are 15% and 8% lower respectively.

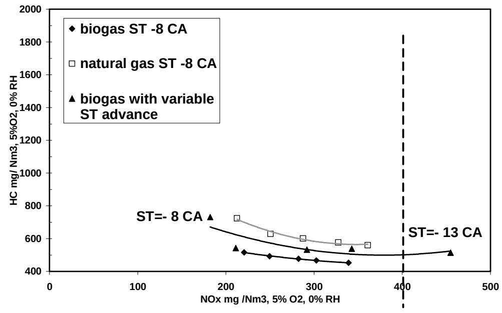

Figure 13. Comparison between natural gas and synthetic biogas, HC emissions

With those conditions (ST 8 CA_{BTDC}) the CO and NOx emissions always fulfill the legislation requirements but with a reduction in efficiency of 1%, $0.355<\eta_f<0.366$. One way of recovering this difference in efficiency is to increase the spark-timing advance (*Figure 11*).

4.3 Influence of spark timing advance

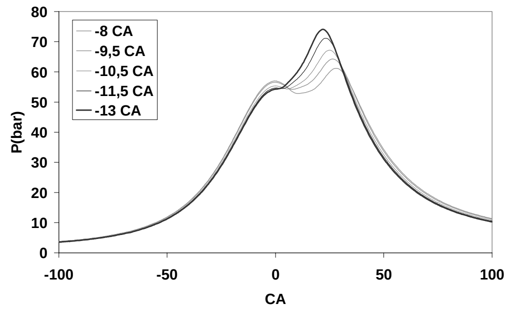

Figure 14. Influence of spark timing on main cylinder pressure cycle

When the spark timing advance increases, the maximum pressure of the cycle is higher (*Figure 14*) and comes sooner but the entire work is still during the expansion phase. The higher peak pressure induces a higher combustion temperature in the main combustion chamber and also promotes the NOx thermal formation (*Figure 15*). Due to this higher pressure, more fresh mixture is trapped in the crevice volume during the early stage of the combustion process. This leads to an increase of the CO emissions throw secondary oxidation. Moreover, the fresh fuel-air mixture has less time to penetrate in the prechamber and the hot combustion gas jets from the prechamber are weaker. Following this, the jets are less efficient to oxidize hydrocarbons trapped in crevices.With an earlier spark timing, the COV of pmi is decreasing (*Figure 10*), giving a good general stability. As the stability of the end of combustion in the expansion phase is better, the HC emissions are lower.

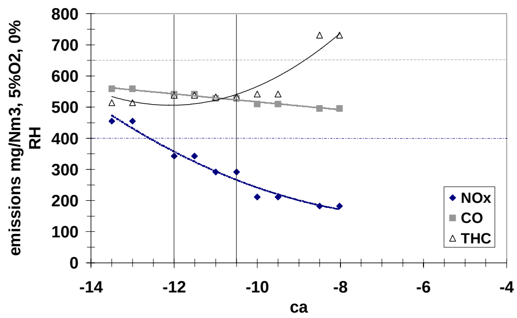

Figure 15. Influence of spark timing on the engine exhaust gas emissions

The positive aspect of a larger ignition advance is a higher fuel conversion efficiency (*Figure 16*) due to a higher peak pressure. To maintain a value of η_f higher than 0.365, one has to use a spark timing between 10.5 and 12 CA_{BTDC} and the emission limit for NOx and CO will still be fulfilled.

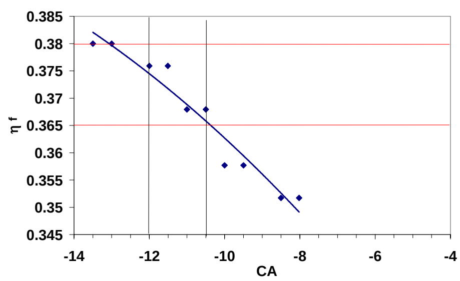

Figure 16. Influence of spark timing on the engine efficiency

5. Conclusion

The results presented in this paper indicate that the cogeneration engine fitted with unscavenged combustion prechamber operates well with a synthetic biogas fuel as the advantage of prechamber ignition compared to direct ignition is kept. The use of simulated biogas for the same rated brake power output of 150 kW and the same NOx emissions, reduces the CO emissions by 15%, and the HC emissions by 8%. The fuel conversion efficiency varies between 0.36 and 0.37 (with the given volumetric compression ratio of 12). The combustion process is essentially unchanged. The velocity and the heat release rate is the same but the peak pressure is lower due to a higher heat capacity of the gas mixture. With a spark timing of 8 CA_{BTDC}, the emissions of CO and NOx are well below the Swiss limits. Increased efficiencies in a range of 0.365 to 0.375 for an ST varying between 10.5 and 12 CA_{BTDC} can be achieved while remaining within the Swiss limits but with an increase in CO and NOx emissions. Another solution could be to increase the volumetric compression ratio. These results open significant perspectives for prechamber ignition as a way to a cleaner and more rational use of biogas without any major economic burden.

Nomenclature

CA	crank angle
CLD	Chemi-luminescence Detector
CO	carbon monoxide
CO_2	carbon dioxide
COV	coefficient of variance
FID	Flame Ionisation Detector
HC	Hydrocarbons
IR	Infra-Red Detector
NOx	nitrogen oxides
Nm^3	normal m^3 (1,013 bar and 273.15 K)
O_2	oxygen
Q	heat
RH	relative humidity
ST	spark timing
pmi	indicated mean effective pressure
rpm	rotations per minute
λ	relative air to fuel ratio
η_f	fuel conversion efficiency
BTDC	Before Top Dead Center

Acknowledgements

This work is financially supported by the Swiss Federal Office of Energy, in collaboration with the engine manufacturer Liebherr Machines Bulle S.A. and the cogeneration group manufacturer DIMAG S.A.

References

Bucksch, S., Egebäck K.E., 1999, “The Swedish program for investigations concerning biofuels”, *The science of the total environment*, 235, 1999, pp. 293-303.

Henham, M. K., 1998, “Assessment of a simulated biogas as fuel for spark ignition engine”, *Energy Conversion & Management*, Vol. 39, 1998, pp. 293-303.

Huang, J., Crookes R. J., 1998, “Spark ignition performance with simulated biogas – a comparison with gasoline and natural gas”, *Journal of Inst. of Energy*, Vol. 71, 1998, pp. 197-203.

Le Conseil Fédéral Suisse 1999, “Ordonnance sur la protection de l'air (OPair) du 16 décembre 1985” (Etat au 12 octobre 1999).

Muller, G. P., 1995, “Landfill gas application development of the Caterpillar G3600 spark ignition engine”, *Journal of Engineering for Gas Turbine and Power*, Vol. 117, 1995, pp. 820-825.

Nellen, C., and Boulouchos, K., 2000, “Natural gas engines for cogeneration: highest efficiency and near zero emissions through turbocharging, EGR and 3-way catalytic converter", *SAE technical paper*, 2000-01-2825, 2000

Röthlisberger, R. P., Leyland, G., Favrat, D., and Raine, R. R., 1998, "Study of a Small Size Cogeneration Gas Engine in Stoichiometric and Lean Burn Modes: Experimentation and Simulation", *SAE Paper*, SP-1391, 982451, 1998

Röthlisberger, R. P., Raine, R. R., Kleemann, R., and Favrat D., 2000, "Experimental Results and Modelling of Carbon Monoxide Emissions from a Natural Gas Fuelled Spark-Ignition Cogeneration Engine", *ImechE Int. Conf. on Computational and Experimental Methods in Reciprocating Engines, IMechE Conference Transactions*, ISBN 1-86058-275-3, pp. 127-138, 2000.

Röthlisberger, R. P., 2001, "An Experimental Investigation of a Lean Burn Natural Gas Prechamber Spark Ignition Engine for Cogeneration", Swiss Federal Institute of Technology of Lausanne, Thesis No. 2346, 2001.

Röthlisberger, R. P., Favrat, D., 2002a, "Comparison between direct and indirect (prechamber) spark ignition in the case of a cogeneration natural gas engine, part I: engine geometrical parameters", *Applied Thermal Engineering*, Vol. 22, 2002, pp. 1217-1229.

Röthlisberger, R. P., Favrat, D., 2002b "Comparison between direct and indirect (prechamber) spark ignition in the case of a cogeneration natural gas engine, part. II: engine operating parameters and turbocharger characteristics", *Applied Thermal Engineering*, Vol. 22, 2002, pp. 1231-1243.

Röthlisberger, R. P., Favrat, D., 2003a, "Investigation of the prechamber geometrical configuration of a natural gas spark ignition engine for cogeneration; part I, numerical simulation", accepted for publication in the Int. J. Th. Sc., 2003.

Röthlisberger, R. P., Favrat, D., 2003b "Investigation of the prechamber geometrical configuration of a natural gas spark ignition engine for cogeneration; part II, experimentation", accepted for publication in the Int. J. Th. Sc., 2003.

Stone, C. R., Gould, J., 1993, "Analysis of biogas combustion in spark ignition engines by means of experimental data and computer simulation", *Journal of Inst. of Energy*, Vol. 66, 1993, pp. 180-187.