Clean Engine Vehicle A Natural Gas Driven Euro-4/SULEV with 30% CO2-Emissions

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ABSTRACT

The goal of the Clean Engine Vehicle project (CEV) was the conversion of a gasoline engine to dedicated natural gas operation in order to achieve a significant reduction in CO₂ emissions. The targeted reduction was 30% compared with a gasoline vehicle with similar performance. Along with the reduction in emissions, the second major requirement of the project, however, was compliance of the results with Euro-4 and SULEV emission limits.

The project entailed modifications to the engine and the pre-existing model-based engine control system, the introduction of an enhanced catalytic converter and downsizing and turbocharging of the engine. As required by the initiators of the project, all components used were commonly available, some of them just being optimized or modified for natural gas operation.

Such technology represents one of the cleanest powertrains for motor vehicles, especially if an effect-based assessment of exhaust emissions regarding greenhouse gases, ozone-forming potential, acid formation and toxic exhaust emissions is considered.

INTRODUCTION

A significant reduction in CO₂ emissions in the mobility sector is a major challenge for the next few decades. In combination with efficient powertrain technologies, the potential of natural gas is excellent for a comparably high and cost-effective reduction in CO₂ and toxic emissions in the near future.

The fuel scenario until 2020 of the European Commission includes the promotion of natural gas for transportation [1]. Although one main reason for this scenario is a reduction in the high European dependence on crude oil, the EU Commission also evaluated the potential for the reduction of CO₂ emissions and pollutants for each fuel under discussion. The fact that natural gas as an engine fuel has a lower negative impact on air quality than any other fuel used today is well known [2].

Table 1 lists some important properties of natural gas. Its properties make it a fuel very well suited to S.I. engines.

Property	Gasoline	Natural gas
Octane number [ROZ]	91-100	<130
Heating value [MJ/kg]	43	47
Max. laminar combust. velocity [m/s]	0.45	0.3 - 0.4
Max. flame temperature [°C]	2'200	1'950
Carbon Weight Fraction [mass-%]	87	75
CO ₂ -efficiency [g CO ₂ /MJ]	73	59

Table 1: Properties of gasoline and natural gas

Since most of the properties of natural gas are similar to those of gasoline, S.I. engines can be converted to natural gas operation quite easily for with the addition of a second fueling system. Such bi-fuel engines have similar engine efficiency in gasoline and natural gas operation, power loss of about 10% and a CO₂ advantage of typically about 20%.

However, natural gas has the potential for increased engine efficiency because of its high knock resistance if the engine is designed for dedicated natural gas operation. Natural gas engines also have the potential for extremely low exhaust emissions if they are operated with state-of-the-art engine control systems. The problematic traffic-related pollutants such as particulates, ozone precursors and benzene in particular are significantly lower compared with diesel, gasoline or even LPG engines.

THE CLEAN ENGINE VEHICLE PROJECT

The Clean Engine Vehicle project consisted in the conversion of an actual gasoline-powered engine to dedicated natural gas operation. This included the optimization of the compression ratio and the development and realization of a suitable catalyst converter and downsizing concept. The aim of the project was to reduce CO₂ emissions in the New European Driving Cycle (NEDC) by 30% compared with a gasoline vehicle of similar performance. Additionally, the vehicle had to comply with the Euro-4 and SULEV emission standards (without deterioration factor).

BASIC VEHICLE AND TEST SETUP

The basic project vehicle was a Volkswagen Polo (MY2000) with a curb weight of 1,020 kg and a naturally aspirated 4 cylinder, 1.0 liter, 2 valve gasoline engine with 37 kW at 5,000 rpm, and 86 Nm at 3,000 rpm. The original compression ratio was 10.7. The engine was equipped with manifold port injection (MPI), exhaust gas recirculation (EGR), and a pre- and main catalytic converter. The vehicle was certified according to Euro-4.

The project included investigations on the engine and the chassis test bench. Because of the extremely low emission level, the dilution air of the CVS system was cleaned by additional air filters, dried by cooling down to 0°C for the elimination of water vapor and heated up to 23°C. In addition, the whole raw gas line until the mixing point in the CVS system was heated to 100°C to prevent the water from condensing. The response factor for CH₄ of the propane-calibrated T.HC analyzer (FID) was 1.04. The CH₄ emissions were measured with a methane calibrated GC-FID. The investigations were performed with the following fuels:

	G20	G25	Cal. NG	Swiss NG
Methane	100.0%	86.0%	89.9%	92.0%
Ethane			4.0%	3.0%
C ₃ and higher			2.0%	1.0%
Oxygen			0.5%	
Inert Gases		14.0%	3.5%	3.0%

Table 2: Natural gas test fuels

The G20, G25 and Cal. NG (California certification fuel) fuels were produced according to the appropriate European and Californian regulations by a Swiss gas supplier. The Swiss NG was supplied by a public natural gas fueling station near Zurich.

G20 and G25 were used for the measurements according to European Regulation 70/220/EC and the Cal. NG was used for the SULEV measurements. The Swiss NG was used together with G20 and G25 for the investigation of knocking behavior.

INCREASE IN COMPRESSION RATIO

An increase in the compression ratio was implemented because of the higher octane number of natural gas. This results in a higher maximal compression pressure and temperature as well as because of the greater expansion at a lower exhaust gas temperature. The overall result is increased thermal efficiency or increased engine performance.

In a first step, the influence of the EGR rate, the ignition point, the engine load and speed on combustion was investigated. This was done by combustion, fuel consumption and exhaust gas analysis at some relevant operating points with the original combustion chamber design with a compression ratio of 10.7 (Fig. 1).

Fig. 1: Original piston and combustion chamber with a compression ratio of 10.7

With the help of the measured in-cylinder pressure traces, the combustion parameters were fitted in accordance with the Vibe model [4]. These parameters were used to simulate the engine with a process simulator which calculated the processes in the ducts with one-dimensional gas dynamics and the in-cylinder thermodynamics with a two-zone model. The simulator was used to predict the effect of different compression ratios on the engine performance. Fig. 2 shows the simulated potential for the reduction in the brake-specific fuel consumption (bsfc) for constant engine speed at 2,000 rpm and a constant brake mean effective pressure (bmeP) of 2 bar by varying the compression ratio and the position of combustion.

Fig. 2 Influence of the compression ratio and spark timing at 2,000 rpm and a bmep of 2 bar relating to the bsfc

The simulation predicts the best brake-specific fuel consumption at a compression ratio of 13.5. At higher compression ratios, the pumping losses outweigh the advantages of the increased compression ratio. The increase in the compression ratio at constant bmep shows a potential of about 2-3% compared to the basic compression ratio of 10.7 at a combustion center of 8° after TDC.

However, the highest possible compression ratio of the engine was defined by mechanical limitations at TDC (distance valve - piston, cylinder head - piston). It was not possible to increase the top land of the piston but only to minimize the combustion-chamber bowl. The complete elimination of the combustion-chamber bowl resulted coincidentally in the targeted compression ratio of 13.5.

The modified pistons without combustion-chamber bowl (Fig. 3) were installed and the measurements mentioned above have been repeated.

Fig 3: Modified piston (without combustion-chamber bowl) and combustion chamber with a Compression ratio of 13.5

Fig. 4 shows the simulated and the measured brake specific fuel consumption at two different operation points for the modified compression ratio ($\epsilon = 13.5$).

Fig 4: Calculated and measured influence on specific fuel consumption by increased compression ratio at four different engine states

They are compared with those of the original compression ratio ($\epsilon = 10.7$). It can be shown that the simulation allows good prediction of the brake specific fuel consumption.

DOWNSIZING

Using smaller but supercharged engines can improve the fuel economy of vehicles. Dedicated natural gas engines are much more suitable for supercharging than gasoline engines because of the high octane number of natural gas.

Because of the lower engine charge as a consequence of the injection of the fuel in gaseous form, a naturally aspirated natural gas engine shows a power and torque reduction of about 10 – 15 %. The basic 1.0 l gasoline engine in the vehicle running at 80 km/h in 5th gear had a torque reserve of 58 Nm. This value drops to 49 Nm when the same engine (with the basic compression ratio) is driven with natural gas (see Fig. 5).

Fig. 5 Calculated torque at full load for the naturally aspirated gasoline and natural gas engine and for turbocharged natural gas operation

Moreover, if a gearbox designed for the 1.4 l engine of the same vehicle is used, the torque reserve is reduced to 39 Nm. This would mean that it would be virtually impossible to accelerate from 80 km/h in 5th gear. It would in fact be possible to save fuel with this powertrain, but at the cost of the “fun” of driving – one of the main reasons why people do not buy enhanced environmentally friendly cars.

In the project, it was possible to compensate the power losses by supercharging the 1.0 l engine so that it achieved the performance of the comparable 1.4 l engine. In this way, the torque reserve could actually be boosted beyond the original value. Simulations indicated that a torque reserve of about 80 Nm can be achieved at 80 km/h in 5th gear, even with the modified gear box.

Investigations on the supercharged engine (maximum intake pressure 1.6 bar) with increased compression ratio ($\epsilon=13.5$) showed that the knock behaviour was still controllable (Fig. 6)

Fig. 6 Torque and specific fuel consumption with different natural gas qualities at 3,000 rpm and 600 mbar boost pressure

The engine can be operated in the range of the best specific fuel consumption even with natural gas from the Swiss grid. At this operation point, the knock limit is only 4 - 6 degrees earlier than with the G25 gas.

The turbocharging included a boost pressure controller with a best efficiency strategy. The turbine bypass valve was open during throttled engine operation and closed only for operating points where boost pressure is needed. Further investigations on flame propagation and knocking behavior are in progress.

The downsizing concept also included modifications to the gear box ratios. Because the performance of the similar 1.4 l gasoline engine could be matched, we used the gear box of the vehicle with this 1.4 l engine.

Fig. 7 shows the influence on torque at a constant speed of 80 km/h in 5th gear and the calculated reduction of the specific fuel consumption (ΔBe) with the two gearboxes.

Fig. 7 Driving curve (5th gear) with original and modified gear box ratios

The simulated reduction of specific fuel consumption in this operation point was 11% with the modified gearbox compared to the original gearbox. However, supercharging also has an effect on the exhaust gas emissions by extending the light-off time of the catalysts by a few seconds. Turbocharging also results in a lower exhaust gas temperature level after the turbine.

ENGINE CONTROL SYSTEM

The engine was equipped with a Bosch ME7 engine control system. This system transforms the position of the accelerator pedal to a desired value for the engine torque. All necessary engine parameters to achieve the desired torque are calculated by the ECU and adjusted accordingly [7]. Transient procedures such as the intake manifold pressure behavior are simulated in models. This allows controlled transient operations of the engine to be performed. This system is able to ensure comparably low engine-out exhaust emissions, especially in dynamic operation.

The engine control system was modified by Robert Bosch GmbH for dedicated natural gas operation and extended for the turbocharging with a boost controller. Therefore, additional functions for the gas supply and injection had to be included and some gasoline-specific functions such as the evaporation control system or the gasoline wall wetting model could be removed.

After the realization of the final powertrain setup, the engine control system had to be applied on the engine and chassis test bench with regard to exhaust emissions, fuel consumption, engine performance and drivability.

CATALYTIC CONVERTERS

The original vehicle in gasoline operation was equipped with a metallic catalytic converter close to the engine (pre-catalytic converter) and an underfloor main catalytic converter. The original pre-catalytic converter was replaced in the project by a new one with a slightly higher volume, a higher cell density with lower wall thickness and a higher precious metal loading.

While a stoichiometrically driven gasoline engine is operated in some cases in fuel-rich conditions ($\lambda < 1$), a stoichiometrically driven natural gas engine can be operated "always" at $\lambda = 1$. This is because there are no wall wetting dynamics that have to be compensated for and due to the lack of fuel evaporation which could be used for increasing the charge density. Further, the maximum laminar velocity of a methane flame is closer to $\lambda = 1$ than the one of a gasoline flame.

Because of the stable molecule structure of methane, the temperature of the catalytic converter is even more important for natural gas than for gasoline operation. This two main differences (always $\lambda = 1$, temperature of the catalytic converter) between gasoline and natural gas engines leads to other requirements for catalytic converters for natural gas engines, mainly regarding precious metal loading, oxygen storage capacity and placement.

The concept used in the Clean Engine Vehicle project consisted of a new pre-catalytic converter with a higher precious metal loading and the standard main catalyst.

The substrate for the new pre-catalytic converter was supplied by Corning GmbH with the following specifications (Table 3):

Specification	pre-catalyst
Substrate material	Cordierite-ceramic
Cell density (cpi)	600
Wall thickness (mil; 1 mil = 10^{-3} inch = 0.0254 mm)	4
Cell geometry	quadratic
Geometric Surface Area (m^2/l)	3.45
Open Front Area (%)	80
Hydraulic Diameter of the cell tubes (mm)	0.928

Table 3: Properties of the substrate of the new pre-catalytic converter

This pre-catalytic converter represents the state-of-the-art regarding the substrate specifications for mass-production and a good compromise between active surface, mechanical stability and back pressure [5].

Generally, the coating of catalytic converters for natural gas engines is different from that of gasoline engines because the engine-out exhaust emissions are quite different.

The engine out CO, NMHC and Nox emissions of a natural gas engine are much lower, but the CH₄ emissions are much higher than those of a gasoline engine. Therefore and because of the stable molecule structure of CH₄, the coating has to be designed for a high CH₄ selectivity. In our project, the coating of the new pre-catalyst was produced by Engelhard Technologies GmbH according the specifications in Table 4.

Specification	pre-catalyst
Wash coat	Al ₂ O ₃ -based
Precious metal coating	Pd-rich
Loading (g/cu.ft)	300

Table 4: Coating of the new pre-catalytic converter

This higher precious metal loading of the pre-catalytic converter was necessary to achieve the Euro-4 and SULEV emission limits with the supercharged engine. However, it was possible to comply with the Euro-4-limits with a "gasoline coating" (80 g/cu.ft) on the same substrate with the naturally aspirated engine.

The coated pre-catalytic converter was canned with an InteramTM 100 HD Mat (4070 g/m²) from 3M [6] and equipped with flanges on both sides to change them without replacement of the complete exhaust line. This solution was very helpful for the project but included some disadvantages regarding the thermal mass during warm-up of the catalytic converter.

The catalytic converters were first driven for about 300 km and then tested regarding their conversion efficiency in different test cycles.

Fig. 8 Ceramic substrate of the new pre-catalytic converter

Exhaust gas temperature

The cold start emissions of a dedicated natural gas vehicles are much lower than those of gasoline vehicles. Nevertheless, cold start behavior still remains an important factor in emission reduction because of the cold catalytic converter at the start of the test.

The cold start emission behavior depends on the temperature profile of the exhaust gas during the warm-up of the engine. Figure 9 shows this temperature profile before the pre-catalytic converter for the engine without and with supercharging in the NEDC.

Fig. 9 Exhaust gas temperature profile in the NEDC without and with supercharging after cold start

The pre-catalytic converter of the engine without supercharging was installed closer to the engine exhaust valve than that of the supercharged engine. The turbocharger absorbs heat from the exhaust gas during cold engine start because of its thermal inertia. This is the reason why the exhaust temperature of the engine with supercharging rises more slowly after a cold start, even with the wastegate fully open during warm-up.

While the pre-catalytic converter temperature of the naturally aspirated engine reached 300 °C only about 12 seconds after the start of the engine, the engine with supercharging reached the same temperature about 30 s later. The temperature before the catalytic converter is often more than 100 °C lower during the European driving cycle because of the increased heat losses and the conversion of the exhaust gas enthalpy by the turbine. This loss in temperature mainly causes problems for the catalytic methane conversion. The stable molecule structure of methane requires higher temperatures for the conversion than other hydrocarbons do.

Conversion efficiency

The level of the engine-out emissions is important for the tailpipe emissions during warm-up. A combination of low engine-out emissions (which need a sophisticated engine control system) and appropriate catalytic

converters is needed to fulfill the requirements of the Euro-4 and SULEV regulations. For Euro-4, the T.HC emission is the most challenging pollutant because methane is included in the T.HC limit. For the SULEV requirements, the NO_x emissions are hard to achieve but not the hydrocarbon limits, because of the non-methane assessment. The corresponding CO limits do not cause any problems because of the typically low CO emissions of natural gas vehicles.

It can be observed that the exhaust gas temperature in the FTP-75 cycle at the same speed is up to 50 °C higher than in the NEDC (Fig. 10). This is mainly due to the more dynamic speed pattern of the FTP-75 cycle and a different gear-change strategy.

Fig. 10 Exhaust gas temperature in the NEDC and FTP-75 cycle

Figure 11 shows that the T.HC emissions can be reduced significantly by appropriate catalyst technology even at the exhaust gas temperatures mentioned above in the European driving cycle. The T.HC emissions in this cycle consist to more than 95 % of methane.

Fig. 11 Engine-out and tailpipe T.HC emission in the NEDC

Figure 12 shows the T.HC emissions in the FTP-75 cycle.

Fig. 12 Engine-out and tailpipe T.HC emission in the FTP-75 cycle

The engine-out T.HC emissions in the FTP-75 cycle are generally higher, but the conversion efficiency of the catalyst is also higher because of the higher exhaust temperatures. Overall, the tailpipe T.HC-emissions are lower than in the NEDC.

The engine-out NOx emissions depend greatly on the engine load. Figure 13 shows them in the NEDC:

Fig. 13 Engine-out and tailpipe NOx emission in the NEDC

Because of the comparatively low engine load in the NEDC, especially during the first part, the engine-out NOx emissions are lower than in the FTP-75 cycle (see Fig. 14). The conversion efficiency of the catalysts is very high during the whole cycle. The EGR is not active in the first part of the cycle.

Fig. 14 Engine-out and tailpipe NOx emission in the FTP-75 cycle

The engine-out NOx emissions are higher in the FTP-75 cycle but lower at tailpipe.

FINAL RESULTS

A CO₂-reduction of 31%, compared to a gasoline vehicle with similar performance could be demonstrated in the European driving cycle (Fig. 15).

Fig. 15 CO₂-reduction path – targets and measured values

The main CO₂ advantage of about 20 % is given by the conversion to natural gas operation. The increase in the compression ratio and the realization of the downsizing concept resulted in a further CO₂ reduction of about 10%.

Table 5 and 6 show the final CO, T.HC, NMHC and NO_x emissions compared with those of the gasoline engine in the NEDC and FTP-75 cycle.

NEDC	CO [g/km]	T.HC [g/km]	NMHC [g/km]	NO _x [g/km]
Gasoline	0.12	0.07	0.05	0.06
Natural gas G20	0.14	0.10	0.00	0.00
Natural gas G25	0.15	0.09	0.01	0.00
Target	1.0	0.10	---	0.08

Tab. 5 Exhaust emissions in the NEDC

FTP-75 cycle	CO [g/km]	T.HC [g/km]	NMHC [g/km]	NO _x [g/km]
Gasoline	0.114	0.028	0.020	0.064
Natural gas	0.081	0.047	0.004	0.001
Target	0.620	---	0.006	0.012

Tab. 6 Exhaust emissions in the FTP-75 cycle (SULEV)

All measurements were performed with the same ECU calibration.

CONCLUSIONS

Natural gas as a fuel has the potential to reduce environmental impact significantly and to increase powertrain efficiency, if appropriate technology is used. Enhanced natural gas technology does not show the typical conflict of aims between CO₂ emissions and pollutants as many gasoline and diesel technologies do.

The technologies needed for clean natural gas vehicles are available on the market. Similar CO₂ emission and pollutant levels are achievable with gasoline and diesel vehicles, but only at much higher costs.

The emission reduction of natural gas vehicles compared with gasoline and diesel vehicles is even higher in real-world cycles than in the official cycles, because natural gas engines are always operated at a stoichiometric air to fuel ratio and the load in real world cycles is typically higher, resulting in higher catalyst temperatures.

While the SULEV regulations consider the different impact of the hydrocarbons regarding their ozone forming potential, the European regulations for passenger cars do not distinguish. A separation of CH₄ and NMHC in the European passenger car regulations, as is already done for heavy duty applications, would simplify the development of natural gas vehicles without a significant ecological disadvantage.

The durability of the catalytic converter was not part of the current project, but will be investigated in a subsequent project.

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