Chapter 6
Effects of EGR on Engines Fueled with Natural Gas and Natural Gas/Hydrogen Blends

Luigi De Simio, Michele Gambino and Sabato Iannaccone

Abstract The exhaust gas recirculation can be used in a stoichiometric engine, for suppressing knocking and increasing efficiency, without a significant impact on pollutant emissions, since charge dilution is obtained with inert gases, allowing closed-loop control operations. However, relatively high EGR rates make worse the combustion process. This chapter deepens the effects of EGR on the performance of gaseous powered engines. In particular, the experimental data have been obtained fueling two engines with NG and NG/H_2 mixtures until 40% by volume of hydrogen, at steady state for different loads, measuring emissions upstream and downstream the three-way catalyst and analyzing the combustion process. A naturally aspirated light-duty spark ignition engine and a turbocharged heavy-duty one were tested. The results obtained with the two engines were consistent with each other. In particular, EGR could be utilized to have high specific power, with reduced thermal stress, but also to increase engine efficiency. Moreover, NG fueling permits a large flexibility in EGR system design, due to very clean engine-out exhaust gas, without visible particles. H_2 added to NG allows to mitigate the effect of EGR in reducing combustion speed. The positive effect of H_2 as combustion booster is more evident at EGR rate increasing. Nevertheless, with EGR, an increment of raw THC emission has been observed. Moreover, for the lower exhaust gas temperatures, oxidation of THC in the catalyst could result less effective. For these reasons, the blends with high hydrogen content, allowing a significant reduction of THC formation directly in the combustion chamber, can be usefully utilized for engines optimization with high EGR rates.

Keywords Natural gas · Hydrogen · Exhaust gas recirculation

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List of Acronyms

ATDC	After top dead center
BMEP	Brake mean effective pressure
BSFC	Brake specific fuel consumption
BTDC	Before top dead center
COV	Coefficient of variation
CR	Compression ratio
ECU	Electronic control unit
EGR	Exhaust gas recirculation
HD	Heavy duty
HR	Heat release
ID	Incubation duration
IMEP	Indicated mean effective pressure
LHV	Lower heating value
LPR	Low-pressure route
MAP	Manifold absolute pressure
MCD	Main combustion duration
NG	Natural Gas
NG_{up}	Natural gas unburned percentage
PWM	Pulse-width modulation
SAFR	Stoichiometric air to fuel ratio
SI	Spark ignition
ST	Spark timing
THC	Total hydrocarbons
TP	Throttle valve position
TWC	Three-way catalyst
WG	Wastegate
WHTC	World harmonized transient cycle test
WOT	Wide open throttle

List of Symbols

CA50	Crank angle for 50% heat release ATDC
EGR%	EGR rate
M	Molecular weight
m	Mass flow rate
P	Power
Wc	Water content
y_{H_2}	Mass fraction of H_2 in the NG/H_2 blend

6.1 Introduction

Comparing stoichiometric and lean burn technologies, the first allows to realize spark ignition (SI) engine with very low emissions, while the second allows lower thermal load and fuel consumption, but with worse emissions (Corbo et al. 1995). In particular, a three-way catalyst (TWC), which allows very low THC, CO, and NO_x emissions with stoichiometric fueling, is not able to reduce NO_x in lean condition. Moreover, simple and cheap after-treatment systems to reduce NO_x are not available. With natural gas engine, also the emissions of unburned methane could be a problem with lean mixtures, due to the lower exhaust temperatures, which could affect methane conversion efficiency. To reduce engine thermal load and improve fuel specific consumption, exhaust gas recirculation (EGR) could be used. EGR technology consists in bringing a part of the exhaust gas to the intake manifold. Exhaust gas, in the case of stoichiometric fueling, is composed by CO_2, N_2, and H_2O; therefore, EGR implies the dilution of the intake charge with inert gases that do not influence closed-loop control for stoichiometric fueling which is necessary for proper working of the TWC.

EGR strategy was initially developed for compression ignition engine, as shown, for example, in the articles Bae (1995) and Kohketsu et al. (1997), to reduce NO_x emission, thanks to the dilution effect (reductions in O_2 available) and thermal effect (increases in the specific heat capacity of in-cylinder working gas) of inert species (Ladommatos et al. 1996), which achieve a reduction of in-cylinder maximum temperature, on which NO_x emissions strongly depend. A lot of commercial diesel engines are nowadays equipped with EGR due to the effectiveness and to the cheapness of this technology.

EGR has been also proposed for optimizing gaseous fueling of compression ignition engine with pilot injection (dual-fuel technology). In these dual-fuel engines, EGR could improve such a complex combustion process, as shown for example in the articles (Dishy et al. 1995) and (Millo et al. 2000) to achieve a more smoothed combustion or in De Simio et al. (2007) to allow the optimization of low loads with a throttle valve. More recently, EGR has been studied as a parameter on which to act for affecting chemical kinetics of the auto-ignition process of natural gas as in Singh Kalsi and Subramanian (2016) and Jamsran and Lim (2016).

Focusing on SI natural gas (NG) engines, EGR strategy has been analyzed with different purposes. Since SI NG engines produce very negligible soot at the exhaust gas, any of the possible EGR routes can be utilized without the inconvenience to dirty the intake manifold or the compressor or to lock the EGR valve. The exhaust gas can be spilled before or after the turbine and can be introduced into the intake manifold before or after the compressor. Moreover, EGR can be cooled or not.

When exhaust gases are taken upstream the turbine and mixed with fresh air downstream the compressor, a high-pressure route is followed, while when exhaust comes from tailpipe to the engine intake, it follows a low-pressure route. Mixed solutions are also possible.

In De Simio et al. (2009), three EGR routes are compared with the scope to achieve the highest possible EGR rate for a six-cylinder, turbocharged gas engine. The best setup, which allows to introduce the highest EGR quantities, to provide the best reduction of the thermal load at rated power, was found to be a cooled low-pressure route (LPR) EGR. However, high–low-pressure route could give the possibility to increase engine efficiency by modulating the power output in the widest un-throttled range operation. In Li et al. (2014), intake dilution of a six-cylinder, turbocharged, intercooler, and large-powered NG engine was obtained with inert addition like CO_2 and N_2 instead of by re-cycling exhaust gas, to evaluate the effect of each component on NO_x emissions. They found that CO_2 is more effective than N_2 in reducing NO_x emission. In addition, with reference to the diluent and to the thermal effects, they are almost comfortable in the view of reducing NO_x emissions for both N_2 and CO_2. In Ibrahim and Bari (2008), a numerical model was developed for a single cylinder research SI engine fueled with NG. The engine employs a stoichiometric mixture with EGR dilution. In particular, it has been predicted the minimum dilution value of EGR that could prevent knocking at high compression ratio (CR). In Ibrahim and Bari (2010), on the same engine, the author found that the engine tolerance to EGR increases with the raise of inlet pressure. In Zhang et al. (2016), a turbocharged SI NG engine for heavy-duty trucks, with TWC and cooled EGR, adopted stoichiometric combustion. EGR was optimized on the whole engine map with a generally positive trend, increasing EGR rate with engine load. The author found that the Euro VI emission standard for NO_x emissions (0.46 g/kWh on the WHTC, world harmonized transient cycle test) can be achieved without difficulties, while engine-out CO emissions could increase, in particular in high-speed/high-load regions with high EGR rates. In Zhang et al. (2017), they found high THC emissions in the late stages of the rural subcycle and in the whole motorway subcycle, due to rapid torque change of the WHTC cycle test and the fact that EGR rate changes accordingly, giving less combustion stability. In Feng et al. (2018), the effects of increasing CR and intake boosting when operating engine with cooled EGR are characterized and compared in a single cylinder research engine equipped with a variable CR mechanism. They assess that combing either intake boosting or increased CR with cooled EGR can further improve engine performance within acceptable regulated emissions. In Yan et al. (2017), on a six-cylinder heavy-duty engine, the author test, in addition to a CR increasing and EGR, the use of late intake valve closing to obtain expansion stroke longer than compression one and further increase engine efficiency. In Sen et al. (2011), the authors investigated the effect of EGR on the cycle-to-cycle variations in a three-cylinder light-duty NG SI engine. The paper investigates the frequency variations of the indicated mean effective pressure (IMEP) time series, highlighting the fact that EGR promotes low-frequency oscillation. On the same engine, the authors in Hu et al. (2009) studied also NG-hydrogen blends combined with EGR. The engine was fueled with mixture up to 40% by volume of hydrogen increasing EGR rate up to 40%, founding, in particular, that the coefficient of variation of the indicated mean effective pressure, which increases with the increase of EGR rate, could be reduced with hydrogen addition.

In this chapter, using experimental campaigns carried out on both a light-duty and a heavy-duty engine fueled with stoichiometric mixtures, a discussion will be carried out on the effects that the addition of recycled gas and hydrogen have on the completeness of the NG combustion. To evaluate in a direct and effective way the opposite effects coming from these factors, hydrogen which tends to promote combustion and EGR which, on the contrary, tends to oppose it, two parameters have been carefully calculated: the EGR rate and the NG unburned percentage.

6.1.1 EGR Rate

The EGR rate (EGR%), which is the percentage of engine exhaust in the mixture entering the engine, as defined in Eq. (6.1), is often estimated with the formula (6.2), (Kohketsu et al. 1997), in which the increased CO_2 concentration at the inlet manifold, with respect to the ambient concentration (roughly 400 ppm), is directly related to the share of exhaust gas recycled.

$$\mathrm{EGR\%} = \frac{m_{\mathrm{EGR}}}{m_{\mathrm{EGR}} + m_{\mathrm{air}}} \times 100 \tag{6.1}$$

$$\mathrm{EGR\%} = \frac{[\mathrm{CO_2}]_{\mathrm{intake}} - [\mathrm{CO_2}]_{\mathrm{ambient}}}{[\mathrm{CO_2}]_{\mathrm{exhaust}} - [\mathrm{CO_2}]_{\mathrm{ambient}}} \times 100 \tag{6.2}$$

This estimation does not take into account the injected fuel, which alters the mass balance between engine intake and exhaust, or the different density and the mixture composition which influence the molar concentration of the involved species. While the formula could result sufficiently approximated for diesel engine running with high air dilution, for stoichiometric engine, the EGR rate could be underestimated. For NG engines, this underestimation could be up to a 15–20%.

A more approximated evaluation could be obtained performing a mass balance of the gaseous flows between the engine intake and the exhaust, the measurement of composition in each section through sampling analysis, and water production calculation.

The mass flow of EGR gas can be obtained by calculating separately the flow rate of dry gas and that of the water vapor. This separation is made necessary by the fact that from the gas analyzers, it is possible to derive the exhaust composition, on a dry basis, as O_2, CO_2, and consequentially N_2. Air mass flow rate and humidity are commonly measured at the engine intake, respectively, with mass flow meters and hygrometers; therefore, water vapor content can be calculated and consequently the dry air mass flow rate. The water produced during the fuel combustion can be calculated directly from the fuel composition and the fuel mass flow rate. From these parameters, and measuring the dry composition at the intake and the exhaust, it is possible to obtain the EGR rate.

In the following, M is the molecular weight, m the mass flow rate, concentration in bracket the molar fraction, while the suffix *wc* stands for water content.

CO_2 entering the engine both with the intake air and with the recycled exhaust gas can be calculated with the Eq. (6.3):

$$m_{CO_2,\mathrm{air,dry}} + m_{CO_2,\mathrm{EGR,dry}} = [CO_2]_{\mathrm{intake,dry}} \frac{M_{CO_2}}{M_{\mathrm{intake,dry}}} \left(m_{\mathrm{air,dry}} + m_{\mathrm{EGR,dry}}\right) \quad (6.3)$$

Parameter in Eq. (6.3) can be calculated with the followings formulas.

$$M_{\mathrm{intake,dry}} = [CO_2]_{\mathrm{intake,dry}} M_{CO_2} + [N_2]_{\mathrm{intake,dry}} M_{N_2} + [O_2]_{\mathrm{intake,dry}} M_{O_2} \quad (6.4a)$$

$$m_{CO_2,\mathrm{air,dry}} = [CO_2]_{\mathrm{ambient}} \frac{M_{CO_2}}{M_{\mathrm{air,dry}}} m_{\mathrm{air,dry}} \quad (6.4b)$$

$$m_{CO_2,\mathrm{EGR,dry}} = [CO_2]_{\mathrm{exhaust,dry}} \frac{M_{CO_2}}{M_{\mathrm{exhaust,dry}}} m_{\mathrm{EGR,dry}} \quad (6.4c)$$

$$M_{\mathrm{exhaust,dry}} = [CO_2]_{\mathrm{exhaust,dry}} M_{CO_2} + [N_2]_{\mathrm{exhaust,dry}} M_{N_2} + [O_2]_{\mathrm{exhaust,dry}} M_{O_2} \quad (6.4d)$$

In formulas (6.4a, b, c, d), $[CO_2]_{\mathrm{ambient}}$ is the CO_2 molar fraction in ambient air and was assumed to be 4×10^{-4} (i.e., 400 ppm), considering the available literature data with a single significant digit.

The only unknown parameter of Eq. (6.2), the dry mass flow rate of EGR, can be calculated with the Eq. (6.5), obtained substituting (6.4b) and (c) in (6.3):

$$m_{\mathrm{EGR,dry}} = \frac{[CO_2]_{\mathrm{intake,dry}} \frac{M_{CO_2}}{M_{\mathrm{intake,dry}}} - [CO_2]_{\mathrm{ambient}} \frac{M_{CO_2}}{M_{\mathrm{air,dry}}}}{[CO_2]_{\mathrm{exhaust,dry}} \frac{M_{CO_2}}{M_{\mathrm{exhaust,dry}}} - [CO_2]_{\mathrm{intake,dry}} \frac{M_{CO_2}}{M_{\mathrm{intake,dry}}}} m_{\mathrm{air,dry}} \quad (6.5)$$

It is possible to note that approximated Eq. (6.2) can be obtained substituting Eq. (6.5) in (6.1), neglecting water content in EGR, if CO_2 concentration is measured on a dry basis, and assuming: $M_{\mathrm{intake,\ dry}} \approx M_{\mathrm{air,\ dry}} \approx M_{\mathrm{exhaust,\ dry}}$.

As regards the water vapor content in the EGR, as the percentage of dry exhaust gas recycled must be the same of the percentage of water in the exhaust gas recycled, the Eq. (6.6), in which f is a constant, can be written:

$$\frac{m_{\mathrm{EGR,dry}}}{m_{\mathrm{exhaust,dry}}} = \frac{m_{\mathrm{EGR,wc}}}{m_{\mathrm{exhaust,wc}}} = f \quad (6.6)$$

Therefore, the recycled water can be calculated with the Eq. (6.7) obtained from the Eq. (6.6), sorting all the sources of water in the exhaust and calculating water from combustion as the mass of water produced per mass of fuel burned.

$$
\begin{aligned}
m_{\mathrm{EGR,\,wc}} &= f \times \left(m_{\mathrm{air,\,wc}} + m_{\mathrm{EGR,\,wc}} + m_{\mathrm{combustion,\,wc}}\right) \\
&= \frac{f}{1-f}\left(m_{\mathrm{air,\,wc}} + m_{\mathrm{combustion,\,wc}}\right)
\end{aligned} \tag{6.7}
$$

The total mass of exhaust gas recycled is given by the Eqs. (6.5) and (6.7).

6.2 NG Unburned Percentage

To evaluate the effect of EGR and H_2 addition on the combustion a parameter, defined in De Simio et al. (2016), was used.

$$
NG_{\mathrm{up}} = \frac{m_{\mathrm{THC}}}{m_{\mathrm{NG}}} = \frac{m_{\mathrm{THC}}/P}{\mathrm{BSFC}(1-y_{\mathrm{H_2}})} \times 100\,[\%] \tag{6.8}
$$

NG_{up}, which stands for "natural gas unburned percentage," evaluates the exhaust THC mass flow rate with respect to the NG consumption. Unburned H_2, that is not taken into account by THC measurement device, does not affect NG_{up}; therefore, with NG/H_2 blends, a less value of NG_{up} clearly shows a more complete NG combustion, reduces THC emissions achieved with H_2 addition.

In the case of full natural gas running, the parameter coincides with the combustion efficiency; therefore, an increasing of NG_{up} with EGR rate indicates a worse combustion condition.

6.3 Experimental Setup

Tests with some NG/H_2 blends were carried out on two engines: a light duty and a heavy duty. To the HD SI turbocharged engine (main characteristics in Table 6.1) was added a cooled LPR EGR, in which the exhaust gases come from the turbine outlet at the compressor inlet. The EGR rate was set with a valve using a pulse-width modulation (PWM) signal. An electronic control unit (ECU) enabled to

Table 6.1 Main characteristics of the reference HD SI NG engine

Type	Six-cylinder in-line
Displacement	7800 cm^3
Bore × stroke	115 × 125 mm
CR	11:1
Rated power	200 kW @ 2100 rpm
Max. torque	1100 Nm @ 1100–1650 rpm
Boost pressure	0.8 relative bar, with wastegate valve
NG feeding system	Electronic timed multi-point injection

set the main operating parameters (air/fuel mixture, spark timing, wastegate valve duty cycle, etc.). The apparent heat release (HR) curves were calculated from a mean over 100 consecutive pressure cycles to identify the incubation duration (ID), angular interval between the spark and 10% HR; the main combustion duration (MCD), angular interval between 10 and 90% HR, and the crank angle for 50% heat release (CA50), as the angle at which 50% of heat is released. The spark timing (ST) was set to have a CA50 close to 8–10° after top dead center (ATDC), which is commonly considered a condition for the best thermal conversion efficiency at fixed engine brake torque.

For the light-duty naturally aspirated engine, which main characteristics are shown in Table 6.2, the exhaust gas was taken before the catalyst and reintroduced after the throttle valve.

Moreover, the exhaust gas recycled has been cooled using the engine liquid coolant. The EGR control valve was similar to that used for the HD engine, but with feedback on working position to be sure of the absence of any uncertainty in the valve position. This was necessary, because the EGR route does not always exhibit a strong difference in the pressure between the inlet and the outlet section. Due to the complex fluid dynamic of a reciprocating engine and the mutual influence that have both the intake and the exhaust pressure on the EGR line, also the position of the throttle valve strongly affected the EGR rate. Therefore, the rate of EGR can be significantly different if a given position of the valve is reached in closing or opening maneuvers and also if the given position is reached more slowly or more quickly. For this reason, to find a prefixed EGR rate, the position of the EGR valve has been varied for further approximation very accurately, using the feedback position signal. In particular, the points at low EGR rate were more difficult to perform, for finding the required EGR rate. This behavior, not observed on the HD turbo engine, made the experimental activity on LD engine longer than that on the HD one.

The installed EGR valve is closed with a PWM duty cycle of 97.5%, and fully open at 0%. It worked properly. In fact, at a fixed PWM signal, no EGR variation was observed over the time and no EGR rate was measured when the EGR valve was closed. Anyway, a specified EGR rate did not correspond to a given position of the valve. In Fig. 6.1, the results at different EGR rate, for two engine throttle valve positions (TP), are reported for three repetitions (series 1–3). Since there is not a

Table 6.2 Main characteristics of the reference LD SI NG engine

Type	Four-cylinder in-line
Displacement	1596 cm^3
Bore × stroke	80.5 × 78.4 mm
CR	10.5:1
Rated power	76 kW @ 5750 rpm
Max. torque	144 Nm @ 4000 rpm
Boost pressure	Naturally aspirated
NG feeding system	Electronic timed multi-point injection

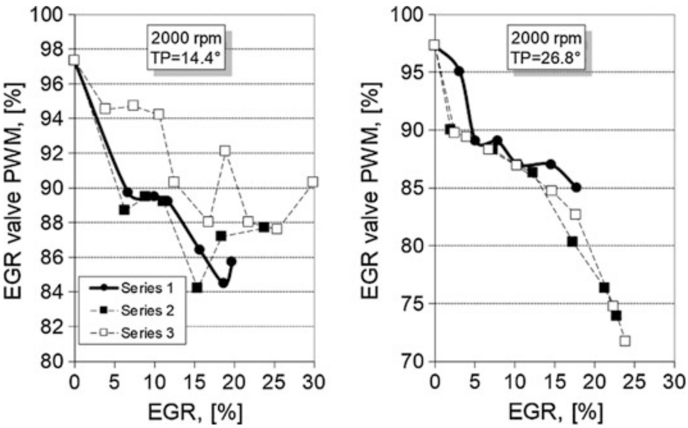


Fig. 6.1 EGR rate at different PWM values at 2000 rpm for all the test conditions

Table 6.3 Instrumentation for emissions and performance measurement

Unit	Type	Range	Accuracy
Air flow	ABB sensy flow P (HD)	1200 kg/h	±1% of reading
	Laminar flow meter cussons (LD)	0 ÷ 350 l/s	±1% of reading
Fuel	Micro motion elite	50 kg/h	<1% of reading
THC	Multifid 14 EGA	0 ÷ 10,000 ppm C_3	0.5% of range
CO	URAS 14 EGA	0 ÷ 10%	<1% of range
NO_x	CLD ecophysics	0 ÷ 5000	<1% of range
CO_2	URAS 14 EGA	0 ÷ 20%	1% of range
O_2	MACROS 16 EGA	0 ÷ 25%	0.5% of range
In-cylinder pressure	Piezoelectric pressure transducer	0 ÷ 250 bar	0.1% of range

close correlation between the measured EGR rate and the PWM duty cycle set, this means that the amount of EGR in steady state does not depend mainly on the position of the EGR valve, but it depends also on the story of the EGR valve changing. Also for the LD engine, test cases refer to the optimal ST phasing.

A list of the main instrumentation used for both HD and LD engines is reported in Table 6.3.

The experimental activity has been carried out with NG and two blends containing a nominal 20 and 40% of hydrogen by volume, with and without EGR, at steady state in different engine conditions, measuring emissions upstream and downstream the catalyst and analyzing the combustion behavior through the study

Table 6.4 Tested fuel composition

Fuel	CH_4	H_2	C_2H_6	C_3H_8	$CO_2 + N_2$
	% vol	% vol	% vol	% vol	% vol
NG	85.36	0.00	9.47	1.75	3.40
NG/H_2 20%	69.89	18.20	6.86	1.40	3.65
NG/H_2 40%	50.55	37.87	6.80	1.33	3.44

Table 6.5 Tested fuel characteristics

Fuel	SAFR	LHV	y_{H_2}	H/C	CO_2	H_2O
	kg/kg	MJ/kg	kg/kg		g/MJ	kg/kg
NG	15.89	46.27	0.00	3.71	57.07	2.01
NG/H_2 20%	16.16	47.56	0.02	4.09	53.89	2.15
NG/H_2 40%	16.56	49.44	0.06	4.79	48.70	2.36

of pressure cycles inside the combustion chamber. An EGR mass percentage until a maximum of 25–30% was tested.

All the fuels were analyzed, and the results are shown in Table 6.4, while the main characteristics of the fuels, like stoichiometric air to fuel ratio (SAFR) or water production through combustion, have been calculated and reported in Table 6.5.

In the following, the experimental activity on the HD engine and on the LD engine and a comparison among the results is shown.

6.4 Experimental Activity on the Heavy-Duty Engine

A first experimental activity was carried out only with NG, at SAFR condition with closed-loop control, at different engine speeds and loads, adding EGR, without changing the ST. The main results of this activity are reported in Fig. 6.2, as the effect on ID and MCD. Each point of the lines in Fig. 6.2 is an average value obtained from points with different loads (from 170 to 850 Nm), but at the same speed. It resulted in quite representative of the behavior of the engine at each speed, being the relative standard deviation of the data at different load less than 5 and 15%, respectively, for ID and MCD. It is possible to note that both ID and MCD are not strongly affected by the speed at each EGR rate. From the above two mean trend lines, main combustion duration is almost not affected by the speed at EGR increasing.

Therefore, as a first wider approximation, to make possible a simple evaluation of EGR effect on combustion development, it can be assumed that ID and MCD are almost not affected by engine load and speed at each EGR rate. In Fig. 6.2, also the relationship between combustion parameters and EGR rate with relative correlation factors is reported.

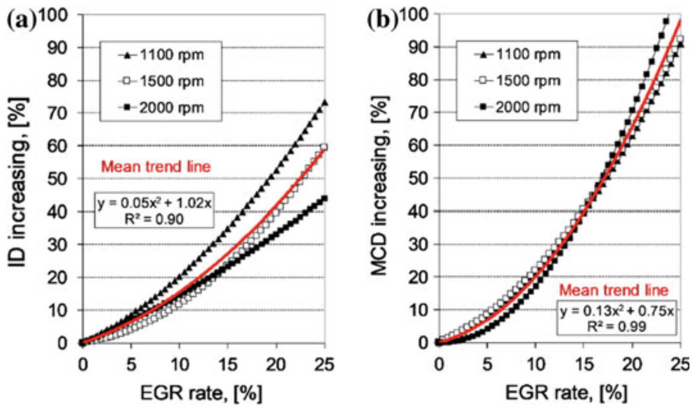


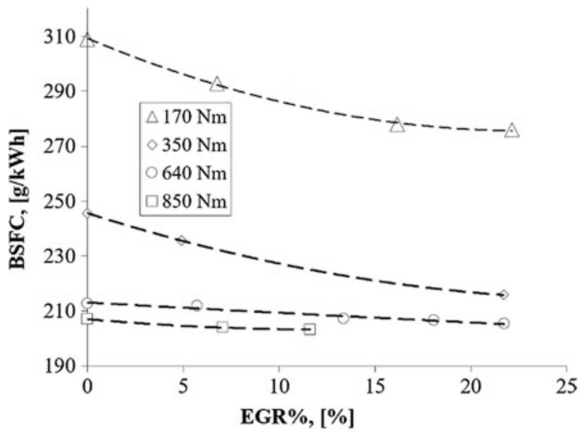
Fig. 6.2 Effect of EGR on ID (**a**) and MCD, (**b**) for the HD engine, without changing the ST

6.4.1 Effect of EGR on HD Engine BSFC

The tests on the HD engine have revealed the possibility to reduce fuel consumption by means of EGR, when a spark timing optimization is made, to compensate the lower flame combustion speed and achieving the maximum thermal efficiency at a given engine brake torque. The results, reported in Fig. 6.3, are consistent with this trend, especially at low loads.

The fuel consumption reduction can be related to both a lower pumping work and a lower heat loss (thermal exchanges through walls, sensible heat of the exhaust gases). A lower pumping work implies a lower active work at the same engine torque, while a lower heat loss requires lower fuel to burn to retain the same in-cylinder pressure.

Fig. 6.3 EGR effect on BSFC at 1100 rpm for the HD engine (ST optimized)



The fraction related to the pumping work could be very small or negligible. Actually, the indicated work (W_i) can be expressed as:

$$W_\mathrm{i} = W_\mathrm{a} + W_\mathrm{p}$$

where W_a is the active work ($W_\mathrm{a} > 0$) and W_p is the pumping work.

Generally, $W_\mathrm{p} < 0$, although at full load and with turbocharger, it is possible to have $W_\mathrm{p} > 0$. If $W_\mathrm{p} < 0$, a positive effect of the EGR on W_p would imply a decrease of $| W_\mathrm{p} |$ due to higher throttle valve opening angles, while it would imply an increase, if $W_\mathrm{p} > 0$, due to higher boost pressure (i.e., higher mass flow rate to the turbine). At the same operating conditions, and therefore at the same W_i, a positive effect of the EGR on W_p would cause a W_a reduction. Such a W_a decrease would determine the same reduction of consumption, assuming negligible the effects of a different cycle development/peak pressure positioning, for pressure cycles with optimized ST.

Figure 6.4 shows some indicated pressure cycles during the phases of the change of charge. It is notable that at low load (*a*), although the mean pressure into the cylinder increases with EGR, there is no effect of the EGR on the W_p, while at mean load (*b*), the W_p increases from negative to positive values, due to a higher boost pressure and therefore of the net work acting on the piston also during the change of the charge. However, this work is of small entity (about 0.5%) respect the W_a.

Therefore, it is reasonable to think that the lower fuel consumption found in the case of EGR is mainly due to a reduction of the heat exchange. This reduction is always present with EGR, and it allows to increase the indicated active efficiency and therefore to decrease the natural gas quantity for each cycle necessary to reach the required indicated torque.

To verify and deepen the experimental results, the EGR effect on the consumption has been analyzed through a mono-dimensional numeric model. To do this, the engine has been modeled both with and without EGR, on all the operating conditions. The EGR% has been set at the maximum possible value, for each

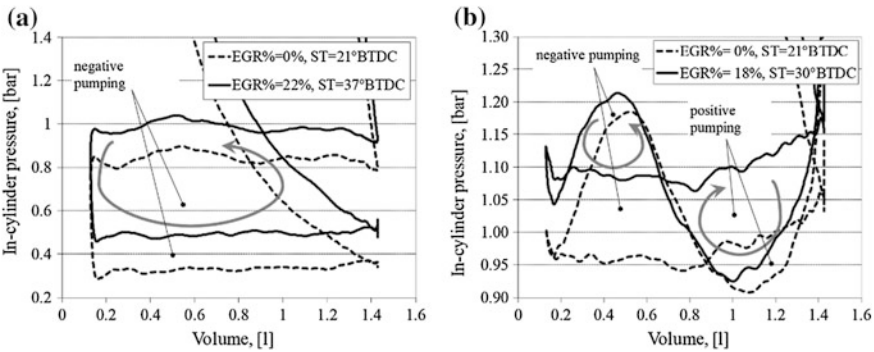


Fig. 6.4 EGR effect on pumping cycle at 1100 rpm and 170 Nm (**a**) or 640 Nm, (**b**) for the HD engine (ST optimized)

condition, without overcoming the 25% threshold. At low load, this result has been reached increasing the throttle valve opening, while at medium–high loads acting also on the wastegate valve (WG), to increase the turbocharging and so allow to introduce EGR, besides the air, to obtain the required torque. The model has ignored the EGR effect on the combustion efficiency, and it has been considered the optimization of the spark timing, both with and without EGR, to have an optimal burner center gravity, considering a lower combustion speed, with EGR, according to the trend of Fig. 6.2. More details about the numeric model construction are reported in De Simio et al. (2009). The results of the model, useful to analyze the BSFC, are reported in Figs. 6.5, 6.6 and 6.7.

Figure 6.5 shows the BSFC percentage reduction of the engine with LPR EGR compared to the base configuration (without EGR). The possibility of obtaining a reduction in consumption of up to 7% in any operating condition of the engine is evident.

The effect of the pumping on fuel consumption reduction ($I_{Wp/\Delta BSFC}$) has been evaluated as:

$$I_{W_p/\Delta BSFC} = \frac{\Delta W_a}{\Delta BSFC} \times 100\,[\%] \tag{6.9}$$

where ΔBSFC represents the consumption percentage reduction in the case of LPR with respect to the base case, and ΔW_a represents the active work percentage variation in the case of LPR with respect to the base case. As before mentioned, the Eq. (6.9) was written assuming that for a given W_a percentage reduction, and there

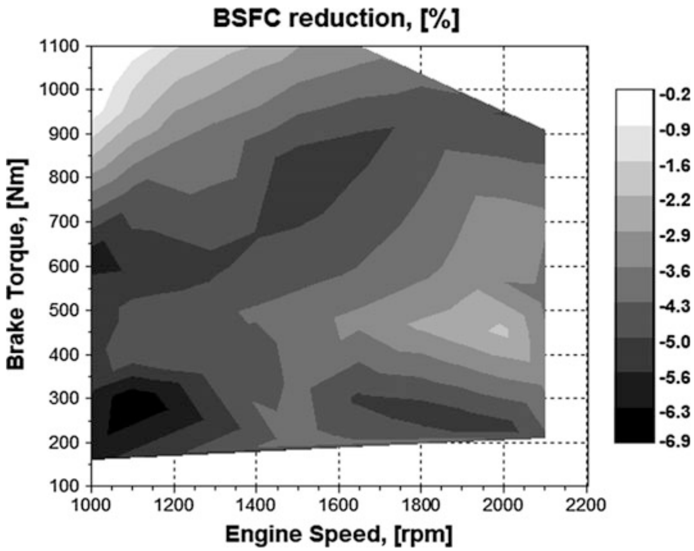


Fig. 6.5 Calculated BSFC percentage reduction achievable switching from the base case to the LPR EGR case on the whole HD engine map (ST optimized)

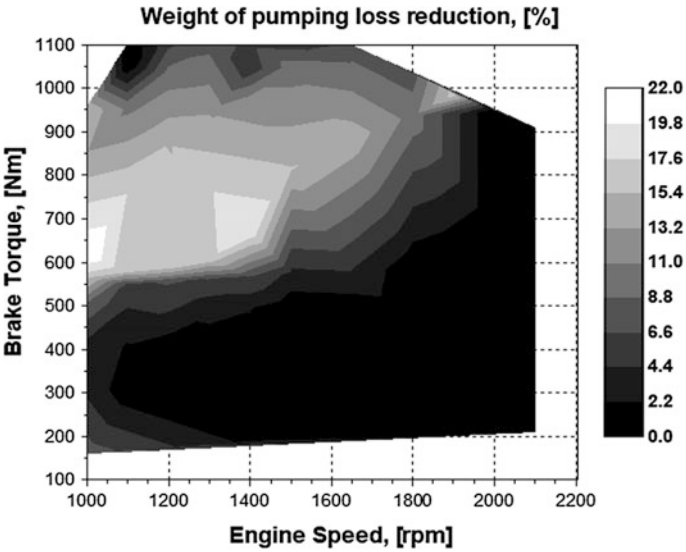


Fig. 6.6 Calculated weight of pumping loss reduction on BSFC percentage reduction achievable switching from the base case to the LPR EGR case on the whole HD engine map (ST optimized)

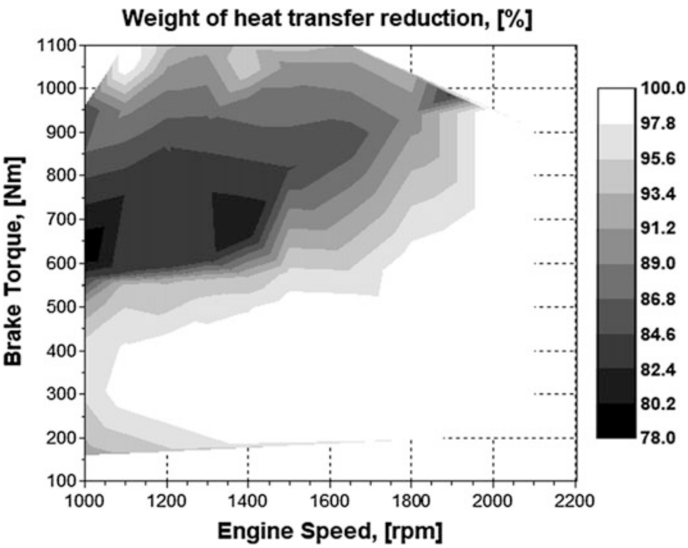


Fig. 6.7 Calculated weight of heat transfer reduction on BSFC percentage reduction achievable switching from the base case to the LPR EGR case on the whole HD engine map (ST optimized)

should be about the same percentage of consumption reduction. Therefore, the positive influence of pumping work reduction on the BSFC reduction is directly related to the active work reduction percentage (i.e., $I_{Wp/\Delta \mathrm{BSFC}}$ is 0% if there is not a reduction in the active work in the EGR case, and it is 100% if the active work percentage reduction is the same as BFSC one). This parameter is shown in Fig. 6.6. It is notable that the influence of the pumping work on the fuel consumption reduction does not exceed 22%, while exists a zone, at high speeds and low/medium loads, where the pumping work is even increased when the EGR is used. Therefore, in that zone, the effect of EGR on the consumption is zeroed because of the EGR flow requires more active work, due to the increase of gas speed through engine valves, which is not balanced by lower-pressure drops through the throttle valve.

The effect of the lesser heat exchange on fuel consumption reduction ($I_{Q/\Delta \mathrm{BSFC}}$) has been valued, instead, as complement to 100%, with Eq. (6.10):

$$I_{Q/\Delta \mathrm{BSFC}} = 100 - I_{W_\mathrm{p}/\Delta \mathrm{BSFC}}\,[\%] \tag{6.10}$$

Equation (6.10) assumes negligible the effect of unburned fuel, and combustion phasing on the BSFC. Therefore, lower BSFC, in the case of EGR, which is not related to a lower pumping work, must be related to lower in-cylinder wall heat losses, which implies less fuel for retaining the engine torque at the same level without EGR. These conditions, that have been considered in the model, could be realistic if not high EGR rate is achieved, being small the worsening of combustion and the difference in cycle development, once the ST is adapted. This effect, reported in Fig. 6.7, clearly shows that the lower fuel consumption calculated with EGR is mainly due to the reduction in heat exchange with the walls. The influence of minor thermal exchange on the consumption reduction ranges between 75 and 100%. In particular, where the pumping work increases with EGR, it is the same possible to have a reduction of the fuel consumption, thanks to less heat exchanges.

6.4.2 Combined Effect of EGR and H_2 Addition

The main activity was performed changing the ST with EGR and adding H_2. The engine was tested with the WG valve fully open and at wide open throttle (WOT). Engine speed was set at 1100 and 1500 rpm.

At increasing EGR, the substitution of part of the inlet air with exhaust gases causes a BMEP reduction at fixed SAFR, Fig. 6.8. Comparing the three blends, only negligible reductions of BMEP were observed with H_2 admission, due to the small reduction of the heat content of the stoichiometric mixture. In the same figure, also the manifold absolute pressure (MAP) is reported. Although the less air entering the engine, MAP is not much influenced by EGR, because the gas mass flow rate expanding into the turbine is more or less the same at EGR increasing,

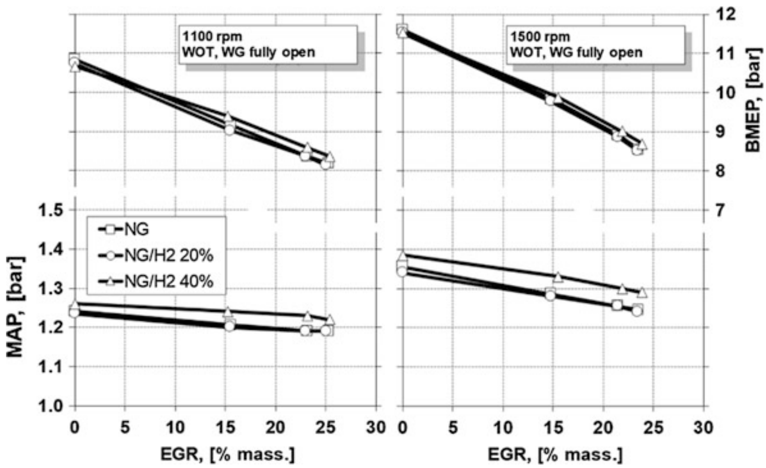


Fig. 6.8 BMEP and MAP at the two tested speeds varying EGR rate and H_2 for the HD engine

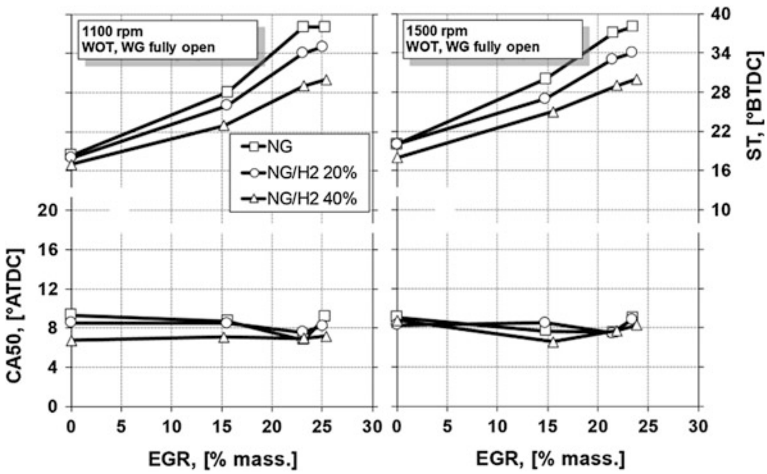


Fig. 6.9 ST and CA50 at the two tested speeds varying EGR rate and H_2 for the HD engine

even if at lower temperatures. As a result, the boost pressure decreases, but less than the load does.

The ST for proper CA50 phasing has to be increased with EGR and decreased with H_2, Fig. 6.9.

The addition of H_2 causes the combustion to be faster, in particular at lower range speed during the first stage of combustion (ID). On the contrary, the combustion propagation becomes worse with high EGR rate. In these last conditions, a more positive effect of H_2 on the start of the ignition process can be noted,

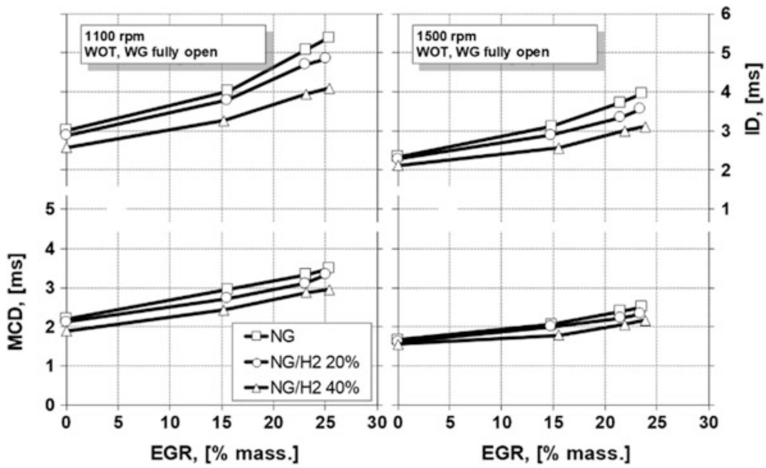


Fig. 6.10 ID and MCD duration at the two tested speeds varying EGR rate and H_2 for the HD engine

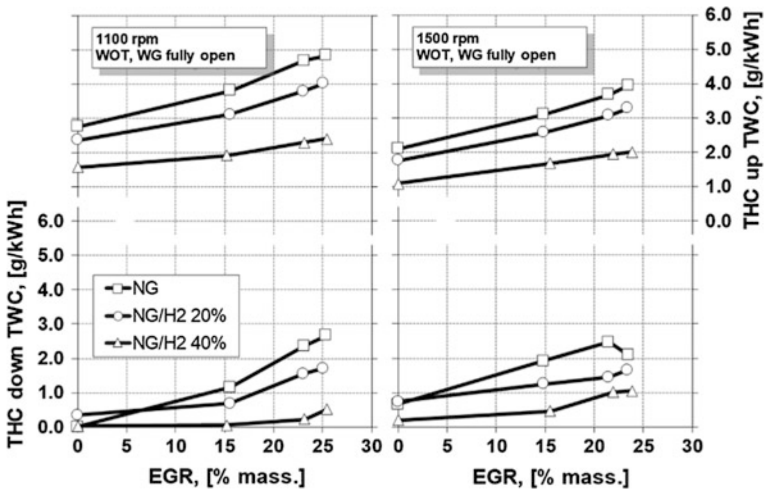


Fig. 6.11 THC upstream and downstream the catalyst at the two tested speeds varying EGR rate and H_2 for the HD engine

Fig. 6.10. In the same figure, a less evident effect on the MCD appears, especially at higher speed, which denotes that the effect of turbulence intensity on combustion speed increasing is higher than that of H_2 addition. The increase in the EGR rate worsens THC emissions, both upstream and downstream of the TWC for all three tested fuels; however, the H_2 content in the mixture helps to limit this drawback, as shown in Fig. 6.11.

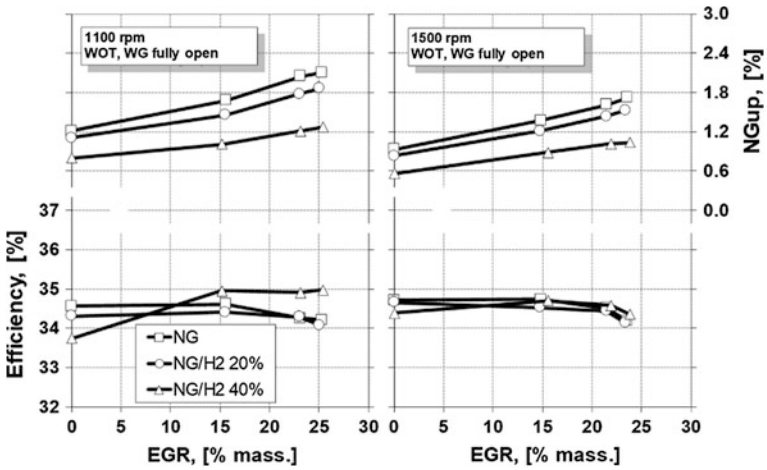


Fig. 6.12 Efficiency and NG_{up} at the two tested speeds varying EGR rate and H_2 for the HD engine

Substituting a share of NG with H_2 contributes to a have a THC reduction, because the unburned H_2 is not recognized by flame ionization detector instrument. Moreover, a more complete combustion also contributes to reduce THC. Therefore, the reduction of THC with H_2 content, Fig. 6.11, together with the NG_{up} decreasing with H_2 content, Fig. 6.12, highlights the positive effect of H_2 on combustion development and propagation, which contrasts the worsening combustion conditions due to the EGR. Nevertheless, the better combustion with H_2 increasing does not improve the overall engine efficiency, (Fig. 6.12). This because from one hand the improvement of combustion efficiency is of small entity, while on the other hand, a more complete combustion, reducing the thickness of the quenching zone, causes an increasing of the heat wall transfer, in an opposite way with respect to what EGR does.

The EGR rate greatly reduces NO_x formation (upstream the TWC) through lower combustion temperature. On the other hand, a slight NO_x increasing inside the combustion chamber can be observed with H_2 in the blend, Fig. 6.13, due to faster combustion, which implies higher reaction temperature. At the TWC outlet, NO_x emissions are quite constant with fuel mixture composition.

6.5 Experimental Activity on the Light-Duty Engine

The experimental tests have been performed at a fixed throttle valve position, adjusting spark timing to have the CA50 at the optimal position. However, due to EGR flow instability, ST was adjusted with minor precision with respect to the HD case. Part-load conditions have been tested to have a pressure drop between the inlet and the outlet of EGR line. Engine speed was set at 2000 rpm.

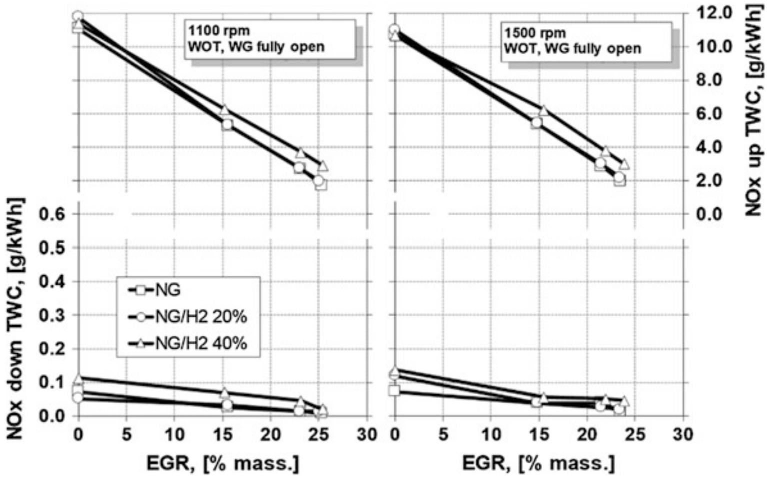


Fig. 6.13 NO_x upstream and downstream the catalyst at the two tested speeds varying EGR rate and H_2 for the HD engine

No efficiency improvement was found with EGR. In Fig. 6.14, it is possible to compare the efficiency at EGR increasing with the case without EGR. Data were obtained as follows. The solid line represents the case without EGR, where at a BMEP increasing corresponds a TP increasing. Starting for a given TP at part load without EGR, the EGR rate was increased, acting only on the EGR valve. An increase in the EGR, at fixed TP, results in a reduction in the supplied torque. At each BMEP, engine efficiency with EGR resulted similar to that of the case without EGR, until at higher EGR rate it becomes to decrease. This behavior was imputed to the worse combustion progress, due to the presence of inert gas in the combustion chamber, that counterbalances the positive effect of lower heat losses and higher throttle valve position, at parity of BMEP. Therefore, there was a difference with respect to the tests on the heavy-duty engine, where the EGR permits a reduction of BSFC. The problem could reside in the weight that unburned hydrocarbons have on the global engine efficiency. With smaller bore engines, the effect of quenching zone is bigger with respect to larger engines, and therefore, the effect of combustion efficiency reduction, correlated with EGR rate increasing, cancels the positive effect of the EGR on BSFC.

Similarly, it is possible to compare the NO_x emission at EGR increasing with respect to the case without EGR. For a given load, less NO_x upstream the TWC were produced with EGR, Fig. 6.15, due to the inert gas which reduces peak temperature in the combustion chamber.

On the contrary, a higher EGR rate determines an increasing of THC upstream the catalyst, as shown in Fig. 6.16, due to the inert gas which makes worse the combustion conditions. Anyway, the presence of H_2 gives a positive effect on THC emissions, both without and with EGR.

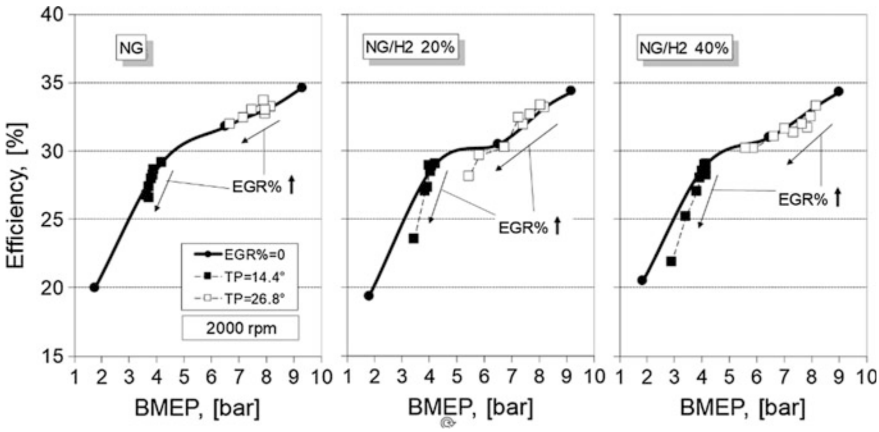


Fig. 6.14 Efficiency at the two part-load conditions varying EGR rate and H_2 for the LD engine

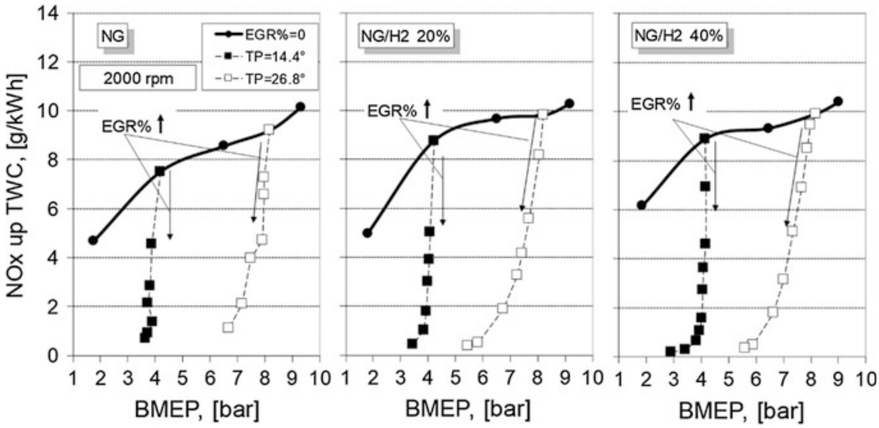


Fig. 6.15 NO_x upstream TWC at the two part-load conditions varying EGR rate and H_2 for the LD engine

In Fig. 6.17, at the two considered TP, the effect of EGR on NG_{up} for the three fuels at 2000 rpm is shown. The blend of NG and 40% by volume of H_2, compared to the NG case without EGR, shows a lower NG_{up} until the EGR rate was lower than 15–20%. This result clearly highlights the possibility of improve combustion condition with H_2 addition.

In the same conditions of Fig. 6.17, THC conversion efficiency is reduced at EGR increasing, as shown in Fig. 6.18. Anyway, NG/H_2 40% shows THC downstream the TWC lower than NG case (without EGR) until 15–20% of EGR rate, Fig. 6.19.

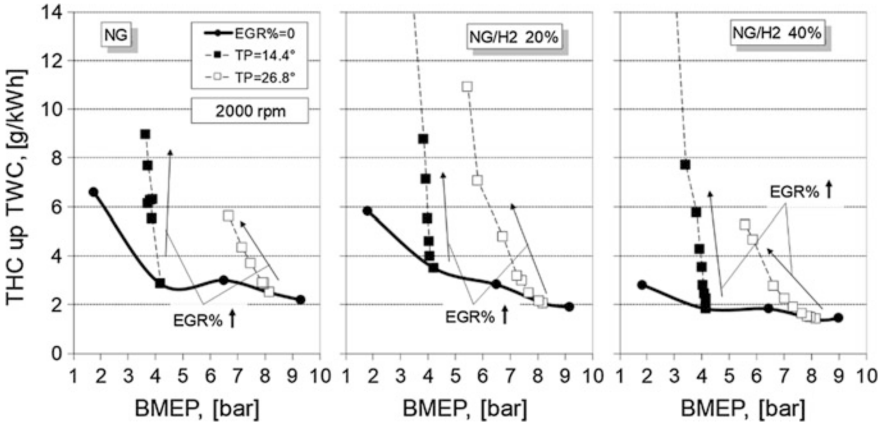


Fig. 6.16 THC upstream TWC at the two part-load conditions varying EGR rate and H_2 for the LD engine

Fig. 6.17 NG_{up} at the two part-load conditions varying EGR rate and H_2 for the LD engine

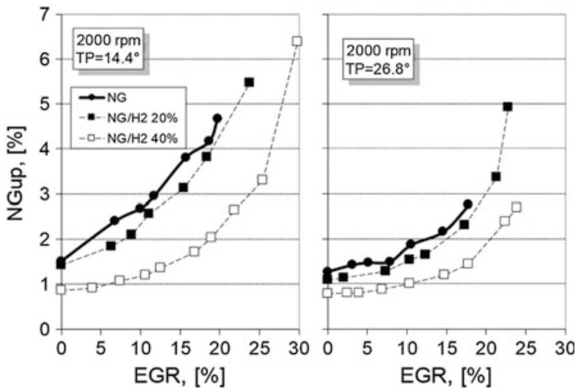
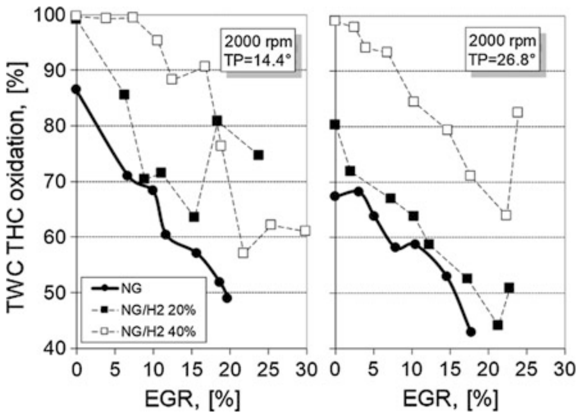


Fig. 6.18 TWC conversion efficiency on THC at the two part-load conditions varying EGR rate and H_2 for the LD engine



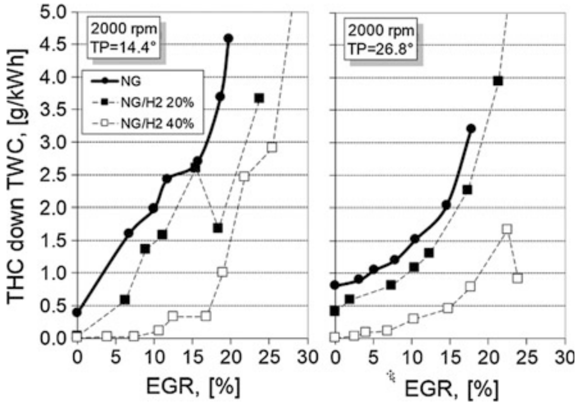


Fig. 6.19 THC downstream TWC at the two part-load conditions varying EGR rate and H_2 for the LD engine

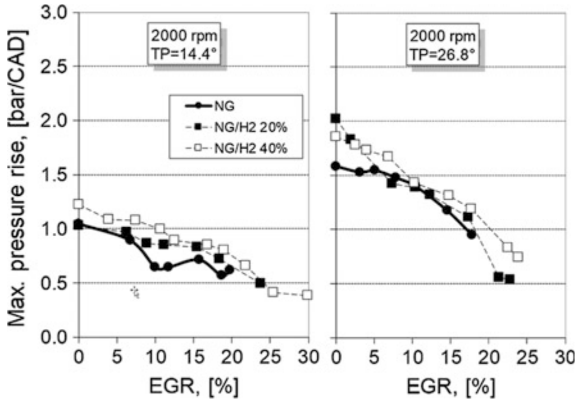


Fig. 6.20 Maximum pressure rise rate at the two part-load conditions varying EGR rate and H_2 for the LD engine

EGR benefit on smoothing combustion is highlighted in Figs. 6.20 and 6.21, where, respectively, maximum pressure rise rate and coefficient of variation of indicated mean effective pressure are shown.

With H_2, the pressure rise is generally slightly higher than the NG case, nevertheless the ST reducing to retain the same peak pressure position. Maximum pressure rise rate is reduced at EGR rate increasing, Fig. 6.20. This behavior could be useful, for instance, to develop turbocharged engine, with high power density, fueled with blend at high H_2 content, lowering maximum in-cylinder pressures and temperatures with high EGR rate. The use of high EGR rate is possible when H_2 is added to NG thanks to the higher capability of the engine to tolerate EGR, as shown in Fig. 6.21 by the effect of EGR on the coefficient of variation (COV) of IMEP. NG/H_2 40%, with an EGR rate up to 15–20%, assures the same combustion stability of the of the NG case without EGR.

In Fig. 6.22, the EGR effect on the ST and the angle of the CA50 is shown, while in Fig. 6.23, the effect on ID and MCD is represented. The comparison of

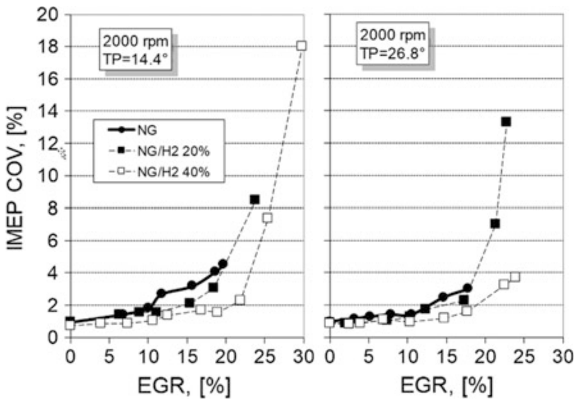


Fig. 6.21 IMEP COV at the two part-load conditions varying EGR rate and H_2 for the LD engine

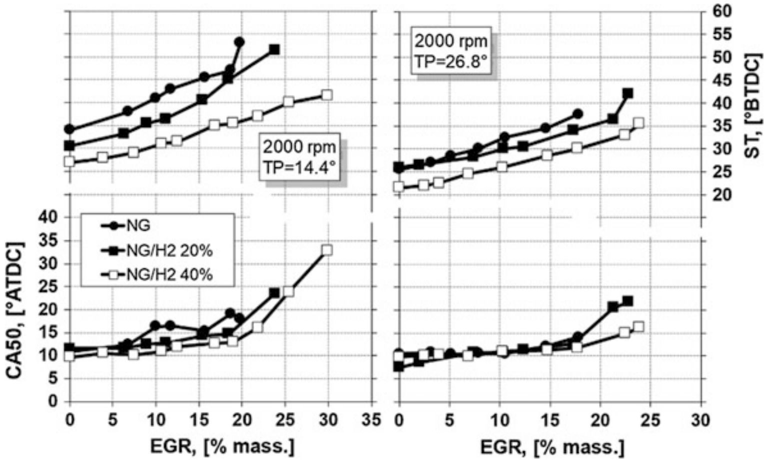


Fig. 6.22 CA50 and ST at the two part-load conditions varying EGR rate and H_2 for the LD engine

Figs. 6.22 and 6.23, referred to the LD engine, with Figs. 6.9 and 6.10 referred to the HD engines evidences a great congruence among the results for the two different engines.

For both the engines, a higher ST was set at EGR rate increasing to retain the CA50 close to the optimal position. Some differences can be noted at EGR rate higher than 20%. In fact, for the LD engine, it was not possible to retain the optimal CA50 position at EGR rate higher than 20%. This is mainly due to the lower LD engine load percentage, with respect to the HD engine. The connected very poor combustion conditions would have required a ST higher than the maximum allowed by the calibration tool software of the LD engine.

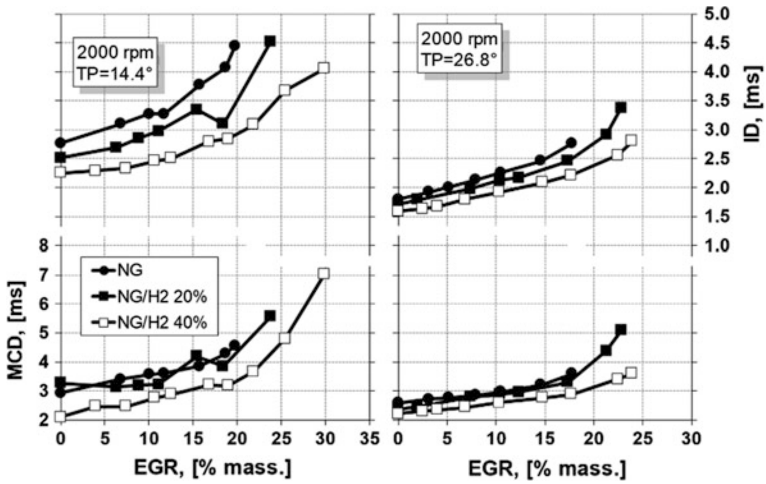


Fig. 6.23 MCD and ID at the two part-load conditions varying EGR rate and H_2 for the LD engine

6.6 Conclusions

The chapter analyses the effect of EGR and hydrogen enrichment on the performance and emission of a light-duty and a heavy-duty NG engine. EGR system for NG engines can be of different types (LPR, HPR, HLPR), due to almost soot free engine exhaust, which does not give fouling of intake system, including compressor and intercooler. EGR could be utilized to have high specific power, with low thermal stress, but also to increase efficiency. The efficiency increasing with EGR is mainly due to the reduction of thermal loss, rather than the decrease of pumping losses. Nevertheless, this last aspect could be of some interest for turbocharged engine: The main advantage could be to run un-throttled and with the wastegate closed, controlling load in a certain range (from maximum to lower torque) by adjusting the EGR rate. On the other hand, combustion propagation is worsened by EGR, which increase fuel consumption. Therefore, when unburned fuel effect exceeds the benefits associated with lower heat exchanges, efficiency gains cannot be achieved. This was the case of the LD engine compared to the HD engine. The results of the combustion process observed on the two engines were consistent with each other, although the increase of THC emissions by means of EGR is less evident on heavy-duty engine, for which the quenching zones have less effect on worsening emissions. Therefore, EGR technology is most suitable for optimizing efficiency of large bore engines.

The effect of EGR in reducing combustion speed is mitigated by hydrogen content in blend with NG. The positive effect of H_2 on combustion propagation is more evident at EGR rate increasing and mainly in the first stage of combustion.

EGR and hydrogen influence pollutant emissions, both upstream and downstream the catalyst. In particular, the lower temperature in combustion chamber with EGR reduces NO_x emissions formation upstream the catalyst, and since the mixture is still stoichiometric, also NO_x conversion efficiency in the three-way catalyst is very effective. Nevertheless, with EGR, the lower combustion temperature and high inert gas levels in the combustion chamber give an increment of THC. Moreover, for the lower exhaust gas temperatures and higher inert concentration, oxidation of THC in the catalyst could result in less effectiveness; for this reason, the blends with high hydrogen content, allowing a significant reduction of THC formation directly in the combustion chamber, can be a suitable opportunity for engines optimized with high EGR rates.

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