Experimental study of two air management strategies for emissions control in heavy duty engines at medium to high loads

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Abstract

Different air management strategies as Miller timing and internal EGR (iEGR) have been studied on internal combustion engines with the objective of decrease NO_x emissions. This paper explores the heavy duty Diesel engine performance by the application of both strategies separately through two different camshaft configurations, mounted and tested in the same engine. In one side, in case of Miller timing, the early intake valve closing is explored and in other side for iEGR, the study is carried out opening the exhaust valve during the intake process. The engine emissions and performance study is achieved through the application of a methodology which begins with the selection of the operating points focusing on medium to high loads. It continues with the exploration of different camshaft profiles by mean of a 1D model. Through the 1D model, two camshaft profiles are selected and tested in the test cell, determining the intake valve closing conditions followed by the identification of the thermodynamic behavior during the compression stroke before the injection. Later on, the combustion

and emissions formation analysis is performed to conclude with the fuel consumption study for each implemented strategy taking in consideration the important influence of each camshaft profile in the pumping loop. A short discussion on transient performance effect of each air management strategy completes the scope of the study.

Abbreviation

$BBDC$	Before Bottom dead centre
$BTDC$	Before Top dead centre
CI	Compression ignition
EGR	Exhaust gas recirculation
$iEGR$	internal Exhaust gas recirculation
ICE	Internal Combustion Engine
IP	Injection Pressure
NoI	Number of injections
PM	Particulate matter
SI	Spark ignition
SOI	Start of injection
TDC	Top dead centre
$Tadb$	Adiabatic Flame temperature

Nomenclature

$BMEP$	Brake mean effective pressure	$[bar]$
$BSFC$	Brake specific fuel consumption	$[g/kWh]$
CAD	Crank angle degree	$[\circ]$
$Comb_{EFF}$	Combustion efficiency	$[-]$
CR	Compression ratio	[-]
Cp	Heat capacity at constant pressure	$[J/kgK]$
EVO	Exhaust valve opening	$[^\circ]$
HRR	Heat release rate	$[J/\circ]$
$IMEP_{GROSS}$	Indicated mean effective pressure gross	$[bar]$
$IMEP_{NET}$	Indicated mean effective pressure net	$[bar]$
$ISFC_{GROSS}$	Indicated specific fuel consumption gross	$[g/kWh]$
$ISFC_{NET}$	Indicated specific fuel consumption net	$[g/kWh]$
IVC	Intake valve closing	$[^\circ]$
m	Mass	$[kg]$, $[mg]$
$\dot{m}$	Mass flow	$[g/s]$
n	Engine speed	$[rpm]$
NO_x	NO + NO2	$[ppm]$
$PMEP$	Pumping mean effective pressure	$[bar]$
p	In-cylinder pressure	$[bar]$
p_{max}	Maximum in-cylinder pressure	$[bar]$
S	Engine stroke	$[m]$
T	Temperature	$[K]$, $[^\circ C]$
V	Volume	$[m^3]$
α	Crank angle	$[\circ]$
τ	Ignition delay	$[\circ]$

Introduction

In recent years the different emission legislation organizations have implemented more stringent emission limits for all kinds of internal combustion engines. This regulations enforce the engine industry to explore different alternatives with aim to reduce and control the emissions level. Particularly Diesel engine has been known like a source of NO_x and Soot emissions, a consequence principally of the short mixing time in the chamber which causes a non-premixed combustion, reaching high combustion temperatures which promotes NO_x formation as described by Kim et al.[1] and the competition between the soot formation and oxidation as discussed by Tree et al.[2]

Now the heavy duty engine industry is assuming new challenges and in the near future the real driving emissions regulation will appear in the same way that in passenger car diesel engines as proposed by Cocker et al.[3] In this actual framework the exploration of "cycle to cycle" emission control techniques is required and it becomes the opportunity to explore different air management strategies which may help the industry to fulfill the future emission normative.

The study of Miller and iEGR separately at medium to high loads helps to understand the behavior of each air management strategy and serve to evaluate the possibility and quantify the real potential of a variable valve actuation system as a tool of "cycle to cycle" emissions control.

Miller strategy

Focusing in NO_x emission, the application of Miller timing in internal combustion engines has been widely studied and it is a demonstrated way to reduce the NO_x emission formation. Recently Verschaeren et al.[4] studied the Miller strategy in a heavy marine engine with an inline system pump injection system. In this study, the Miller strategy is applied in 2 operating conditions at low speed combined with different external EGR rates and obtaining an important reduction of NO_x emission (between 15% to 25%), with a small increment of fuel consumption. Boulouchos et al.[5] proposes after researching in a 9 cylinder heavy duty Diesel engine of 1420 kW of power with Miller timing, the reduction in NO_x emission between 20% to 30% without penalties on fuel consumption and soot emission over the whole load in a low speed operating point.

Benajes et al.[6] as well as Gonca et al.[7] obtained reductions on NO_x emissions between 15% to 25% through modeling and experimental studies in a single cylinder engine using Miller timing (advancing the IVC) and EGR strategies applied combined. This research indicated this important reduction in NO_x emission but a slightly reduction on engine efficiency due to lower compression ratio. In the same way the work performed by Wang et al.[8] where the advancing of IVC was progressively evaluated in a 2 cylinder 1.2L naturally aspirated small diesel engine showing reductions until 13% of NO_x emissions at full load.

Using a simulated 1D model Millo et al.[9] studied the combination between EGR and Miller timing but in this case, due to the mass losses caused by the the earlier intake closing, the intake pressure was increased in order to recover the original intake mass flow rate. In this work was found an important reduction of NO_x (until 35%) without affect or penalize the fuel consumption.

iEGR strategy

The other air management strategy evaluated in this research is the iEGR, which is mainly used to reduce the NO_x formation modifying the oxygen concentration in the combustion chamber as studied by Balaji et al.[10] The strategy is performed by opening the exhaust valve during the the intake stroke, allowing to part of hot exhaust gases to enter to the cylinder if the pressure balance in the intake and exhaust manifolds allows it. This strategy was explored by Millo et al.[11] in a small 2 cylinder Diesel engine obtaining NO_x emission reduction between 25% and 50%.

The main advantage of this form of EGR (internal) could be summarized in the simplicity and economy of producing the exhaust gas re-circulation compared to the complexity of all the components required to realize the external EGR (cooled or not) strategy as proposed by Meistrick et al.[12] and Shi et al.[13]

One fact to take in account when the iEGR strategy is performed is the opposite effect of opening the exhaust valve during the intake stroke. From one side, the oxygen concentration reduction helps to reduce the NO_x formation, but for other side the hot iEGR increases the in-cylinder temperature. This is one of the disadvantages of iEGR that added with the difficulty controlling the amount of iEGR trapped and also the reduction of the intake charge density, may cause excessive PM emissions as is described by Schwoerer et al.[14]

In both mentioned cases the Miller and iEGR camshaft profiles were studied from the single cylinder engine, multi cylinder engine and the theoretical modeling point of view. In this work the objective is to investigate the engine performance under both strategies separately in a 6 cylinder medium duty Diesel engine by changing the camshaft hardware profile in order to have the desired intake valve closing (for Miller timing) and the desired opening of the exhaust valve during intake process (for iEGR profile).

The engine operating conditions were selected from the engine map in the zone of medium to high load and speed, since this operating conditions will become more important in the future normative (real driving normative) and this engine map zone is where is expected to have a high formation of NO_x as observed by Yanowitz et al[15] . The study will be focused on the differences in pollutant emissions and engine performance in the six selected operating conditions under each of three tested camshaft configuration. A detailed study of the pumping loop behavior is performed for each camshaft at part load due to important influence of camshaft profile affecting the specific fuel consumption.

Methodology and objectives

The methodology followed in this research work consists in analyzing the effect of the Miller and iEGR profiles as air management strategies focused on the emissions control through the classification and study of the different engine events in order of occurrence in time. The main objectives of this research work are:

- To study different air management strategies as a cycle to cycle emission control tools exploring different camshafts profiles through the 1D simulation model (GT Power).
- To evaluate experimentally two air management configurations (Miller and iEGR selected from 1D simulations) in medium to high load on a heavy duty Diesel engine.
- To quantify the range of action for each selected air management strategy as emission control tool through the comparison and study of the combustion behavior and engine performance.

The methodology is structured basically in 4 steps:

1. Determination of intake valve closing conditions

In order to visualize the relationship between different operating parameters the results of simulated and tested strategies are placed in an engine map which summarizes many settings of engine operation: the YO2 concentration at intake and exhaust as well as the A/F and internal EGR. This engine maps helps to study the trends of each strategy and they are a good tool to compare and select the camshafts profile in the design process (Fig 8.) under a common framework and serve to evaluate the experimental behavior of the selected camshafts profile (Fig 9. and Fig 19). In this maps the red iso-lines defines the YO2 concentration at the intake valve closing while the green iso-lines are the YO2 concentration at exhaust valve opening (a set of parameters which are constant independently of each operating condition). The blue iso-lines are the mass flow at the intake valve closing and they are calculated for each particular standard engine point.

2. Compression stroke in-cylinder conditions

The compression in-cylinder behavior is independent from the injection parameters but totally affected by the gas exchange processes conditions. In case of Miller camshaft profile, advancing the IVC shortens the duration of the intake process which reduces the total air mass flow trough the engine which is equivalent to the reduction of effective compression ratio.

For iEGR camshaft profile the total mass flow is reduced like the Miller one but the reason in this case is the opening of exhaust valve when the intake valve is opened reducing the fresh air quantity in the cylinder. In this part of the analysis, the objective is to determine all the thermodynamic conditions from the intake valve closing until beginning of combustion, in order to compare the influence of each tested camshaft profile in the thermodynamic behavior of the compression stroke.

3. Combustion process study

In direct injection diesel engines, the combustion is divided in three stages: Auto ignition delay time, premixed combustion and diffusion controlled combustion. Through a 0D modeling tool which allows to obtain the heat releases rate (HRR), the objective is to study how the diesel fuel is burned at each stage under the in-cylinder conditions generated for each camshaft.

The 0D modeling combustion tool, called CALMEC, is an in-house one-zone model with a total description completely explained by Payri et al.[16] This combustion diagnosis software uses the in-cylinder pressure signal as its main input. The pressure traces from 50 consecutive engine cycles are recorded in order to compensate the cycle-to-cycle variation due to engine operation. Later, each individual pressure cycle data is filtered using a low-pass filter based on Fourier transformation. Once filtered, the collected cycles are averaged to yield a representative cylinder pressure trace, which is used to perform the analysis.

At this point the first law of thermodynamics is applied between IVC and EVO, considering the combustion chamber as an open system due to blow-by and fuel injection. In addition, the ideal gas equation of state is used to calculate the mean gas temperature in the chamber Fig 12. The main results from the model used in this work are the oxygen concentration, density and HRR as shown on combustion study figures. The 0D model provides too additional information related to each cycle. In particular, the indicated mean effective pressure (IMEP) and the start of combustion (defined as the crank angle position in which the accumulated heat release has reached a value of 2%)

4. Pollutant emissions and engine efficiency

The NO_x reduction, main objective of using the tested camshafts will be evaluated with the direct measurement through the gas analyzer described in the experimental setup. The combustion process study will explain the behavior of each camshaft in the NO_x pollutant formation. Added to NO_x, the other important pollutant emissions (Soot, CO, HC) in Diesel engines will be measured and linked with the combustion process.

In case of the engine efficiency, it will be evaluated from a fuel consumption point of view for all selected operating conditions and the indicator diagram study will serve to find the relation between fuel consumption and pollutants under the influence of each camshaft at part load and full load points.

Experimental setup

Engine and test cell description

In order to explore the different air management strategies the experimental tests were carried out in a DI Diesel engine, whose main characteristics are given in Table 1. The distribution of the test cells elements and the measurement devices location are shown in Fig 2.

The engine is directly coupled to an electric dynamo-meter brake allowing the control of the speed and torque. The in-cylinder pressure was measured by means of a Kistler 6125C glow-plug piezoelectric transducer with a range between 0 and 300 bar, and a sensitivity of -36 pC/bar. The pressure sensor was calibrated using a Kistler 6904A1 pressure generator for continuous and dynamic Pressure measurement. The electrical charge yielded by piezo-electric transducers is converted into a proportional voltage signal by means of a Kistler 5011B charge amplifier.

In all evaluated operating conditions, the in-cylinder pressure was recorded during 50 consecutive engine cycles in order to compensate the engine operation dispersion. A 0.2 CAD sampling interval was used for the angle-synchronous acquisition of the in-cylinder pressure, which was performed using a Yokogawa DL708E oscillographic recorder with 8 modules of 16 bits A/D.

The concentration of the exhaust emissions (NO_x, CO and HC) was measured with a specific gas analyzer Horiba MEXA. Smoke emission was measured with a variable sampling smoke meter AVL 415 using the filter paper method that provides results directly in filter smoke number (FSN), units that were transformed into dry soot mass emissions by means of a suitable correlation described in the ISO 8178.

These exhaust gases analyzers and all the other test cell transducers (speed, pressure, temperature) assure an accuracy, defined as the incertitude of the measurements under identical conditions in the same test session described in Table 2.

Operating points selection

The selection of the operating conditions from all the engine range of function was done focusing on medium to high loads. Four operating conditions at part load and two at full load were selected as is shown in Fig 1. three points of medium speed at medium, high and full load and the equivalent points in high speed. The six points are defined by the torque and speed values which were maintained constant for all simulated and tested air management strategies.

In terms of injection pressure as well as the number and timing of injections the conditions were defined for all the selected operating points as shown in Table 3. This injection settings were maintained constant in all the study (nozzle geometry) for the three tested camshaft profiles at part load, but at full load and due to engine thermo-mechanical constraints the injection settings (injection pressure and timing) were modified for iEGR strategy as observed on Table 3.

Camshafts Profiles design and selection

The standard camshaft profile was measured directly on the engine, installing a linear displacement transducer which allows to measure the cam profile using the lift of intake and exhaust valves as is shown in Fig 3. The measured standard profile is represented on black in Fig 4. and Fig 5.and it is an input on the engine model.

At this point with the aim to explore different air management strategies and select the "Optimal camshaft profile to be manufactured" a 1D engine model is configured and validated through the engine test with the standard camshaft. The air management strategies simulated are: four different Miller profiles changing the intake valve closing were simulated as well as six iEGR profiles varying the exhaust valve aperture and angle location during intake stroke. This profiles candidates are represented on Fig 4. and Fig 5. for each air management strategy

GT Power 1D model description

The engine one-dimensional model is configured in GT Power software as shown in Fig 6. The calibration with experimental data were performed in the full load conditions A100 and B100 as can be observed on Fig 7. The combustion efficiency and heat release rate directly came from the experimental data and the in-cylinder heat transfer process was calculated with the WoschniGT model. At each operating condition, the gross indicated mean effective pressure (IMEP) and in-cylinder pressure trace were firstly calibrated with adjustments on the intake and exhaust pressures achieved by the calibration with accuracy of the of the discharge coefficients in valves and turbine and compressor efficiency. It can be seen that the statistic relative differences are generally less than 5% for the five criteria defined as calibration parameters: net and gross IMEP, BSFC, Pint, Pexh and Pmax and the measured and simulated in-cylinder pressure traces also show good agreements.

After the model calibration the camshafts profile simulations were done in the full load conditions A100 and B100 which are the extreme conditions on the two selected engine speed and the previously calibrated points. The engine maps observed on Fig 8. serve to evaluate the influence of each simulated camshaft profiles on the intake valve closing conditions for variables like the Air fuel ratio, the in-cylinder mass or the oxygen concentration.

Based on the results observed on Fig 8. shows the two selected camshaft profiles to be manufactured plotted on red for Miller and blue for iEGR. The main reasons to select them could be observed at point A100. In case of iEGR, the selected profile is on the limit in the A/F relation in order to avoid an excessive richness and a low quantity of iEGR. For Miller case, the advancing of intake valve closing was selected with the aim of preserve a similar mass at intake valve closing compared to iEGR camshaft as well as not affect the fuel consumption in big measure.

Compared to the standard camshaft the selected Miller profile is changing the angle of intake valve closing from 560 CAD to 530 CAD as is shown in Fig 4. In case of the iEGR camshaft profile it opens the exhaust valve during 90 CAD while the intake valve is open as is shown in Fig 5. allowing to get inside the cylinder the exhaust gases in a process strongly dependent on the intake and exhaust manifolds pressure conditions.

Results

The research results are classified in steady test at part load and full load which will be discussed in the framework described on the methodology and the transient test where is evaluated the affectation of changing air management configuration in the engine Torque time response and other parameters like air mass flow and Opacity.

Part Load Results

Determination of intake valve closing conditions

In the engine maps shown on Fig 9. it can be observed the main modifications on intake valve closing conditions caused by the implementation of the Miller and iEGR air management strategies. The Miller profile affects directly the trapped mass at intake valve closing reducing the fresh air mass flow and consequently increasing the richness. In case of the iEGR profile it increases in all cases the residuals which is the main objective of this profile affecting too the trapped mass as was expected due to the increment of internal temperature.

in-Cylinder conditions at compression stroke

As discussed on previous section the first notorious consequence of producing Miller cycle by advancing IVC is a reduction on the trapped air mass, this explains the results in terms of density on the four evaluated engine points compared to the standard camshaft profile density behavior as shown in Fig 10.

As the intake valve was closed earlier, there is an slight expansion of the in-cylinder charge which is the cause of a lower initial pressure on the compression process. This is equivalent to reduce the compression ratio CR and the effect can be noticed at the end of the compression stroke where the maximal pressure is reduced in all part load points compared with the standard camshaft Fig 11. These results in terms of in-cylinder pressure are equivalent to those obtained by Benajes et al.[17] With the reduction in terms of pressure and despite the density decrease a reduction of mean in-cylinder temperature was expected as is shown in Fig 12.

In case of iEGR camshaft, the density behavior shown in Fig 10. is mainly explained by the in-cylinder temperature which is increased by the internal recirculation of exhaust gases causing a density reduction as shown on Fig 12. Considering a similar intake manifold pressure at beginning of the compression process for both iEGR and standard camshafts, there is a pressure reduction at the end of the stroke despite the temperature increase caused by iEGR profile as observed on Fig 11. This pressure reduction is explained by the density decrease added to the in cylinder air dilution with CO2 and H2O (with higher CP with respect to N2 and O2).

Combustion process study

The combustion process is now described to well understand the phenomena that occurs inside the combustion chamber and its influence on the engine efficiency and the pollutant formation. The combustion results are classified at medium load on Fig 13. and medium-high load Fig 14. On the combustion process different variables have been selected as a tracer of the phenomena: the oxygen concentration, the density, the HRR and the adiabatic temperature are plotted as function of mass fraction burned which evolves from 0% to 100%.

In the case of oxygen concentration could be observed that is always higher with the standard profile in four tested engine conditions for the same mass fraction burned. The application of the air management strategies not only affect the trapped mass and consequently the density, but also affects the evolution of the oxygen concentration. This effect is remarked on the medium speed conditions A50 and A75 in the case of iEGR camshaft profiles.

Centering now on the HRR and based on the mixture process both strategies slow down the combustion speed as is shown on the reduction of maximum peak on HRR, again with the iEGR profile and in the medium speed points this decrease is remarked.

The last combustion variable selected as indicator was the adiabatic flame temperature, this is the temperature reached in the spray reaction surface and it is one of the main controls in the chemical processes on the formation of NO_x and soot. It is not a measured variable due to difficulty to measure in an engine without optical access, but on Fig 13. and Fig 14. it is shown the theoretical one calculated from the chemical equilibrium having in account the combustion residuals products and the dissociation effects.

The adiabatic temperature decreases with the Miller profile due to reduction on the in-cylinder temperature at the same oxygen concentration at combustion beginning but in case of iEGR in spite of the temperature increase the reduction of oxygen concentration causes a decrease even bigger than Miller profile. This is an effect over again remarked on the medium speed conditions A50 and A75.

Pollutant emissions and engine efficiency

The NO_x emission was reduced in all the points tested with Miller camshaft as well as iEGR as is shown on Fig 15. and Fig 16. due to the reduction of adiabatic flame temperature during combustion process specially before the end of injection where most of thermal NO_x is formed as proposed by Benajes et al.[18] The Miller and the iEGR camshafts allow the reduction of flame temperature acting on two different ways on the combustion process:

- With Miller camshaft, the main modification is the reduction on the mean temperature of the combustion chamber due to lower pressure at start compression stroke. This reduction of temperature helps to have a cooler combustion which leads to reduce NO_x formation.
- In case of iEGR camshaft, the reduction of NO_x emissions is mainly explained by the reduction of oxygen concentration. This reduction is achieved by the internal re circulation of exhaust gases which contain less oxygen than fresh air. At the end the combustion temperature is reduced despite the fact that the end compression temperature shown in Fig 12. is higher.

Exploring in detail the results in terms of NO_x emissions, this pollutant reduction with iEGR camshaft is more notorious than Miller which is not affected by the engine speed as much as the iEGR does.

Consequently with the results in NO_x emissions the well known trade off is completed by increase in soot emissions. The strong reduction of NO_x achieved by the iEGR camshaft is compensated by a notorious increment in soot emission, specially at medium engine speed

(A points on Fig 15. and Fig 16.). Focusing on these results A50 and A75 points shows a combustion deterioration for iEGR camshaft leading to do not reach the soot oxidation temperatures when the mass fraction burned is near to 100%. Picket et al.[19] describes that the soot formation occurs inside the envelope of the diffusion flame on the high temperature and fuel-rich region after the lift-off length, then the increase on the ambient temperature added to lower concentration of O2 with iEGR camshaft promotes this formation in the same way than developed work by Ladommatos et al.[20]

The CO and HC pollutants are summarized on the combustion efficiency shown on Fig 15. and Fig 16. the Miller and the iEGR camshaft profiles preserve the combustion efficiency reached by the standard profile. Despite the combustion efficiency is preserved, there is an increase in CO emission in all the part load operating conditions under the two tested camshaft profiles as observed in Figure 17. This increase is notorious in the iEGR case at medium speed but is under the limits of the emission standard. HC emissions are very low and they are not almost affected by the camshaft profile.

Engine efficiency

The engine efficiency was evaluated from fuel consumption point of view. As shown on Fig 15. and Fig 16. the ISFC gross, ISFC net and BSFC are compared for all air management configuration at four tested engine conditions. Both tested strategies at the medium speed points (A points) reduces the engine efficiency in a percentage between the 2% and the 8%, showing the same trend as reported in previous works by Benajes et al.[21] where the changes introduced by the camshafts in the gas exchange conditions deteriorates the combustion process. When the analysis is translated to high speed (B points) an opposite effect is found, there is a reduction in the fuel consumption BSFC which cause the improving of the engine efficiency even with the increase on the ISFC gross caused by the combustion deterioration. This behavior could be explained by the ISFC NET where the pumping loop is added.

To better understand this fuel consumption behavior the indicators diagrams under part load tested operating conditions are shown on Fig 18. In this plot the pumping loop is zoomed in order to observe the modifications caused by each camshaft profile since it is this loop that fully explains the behavior in fuel consumption and consequently in the engine efficiency.

At medium speed (A points), there is a high volumetric efficiency then the engine work used to perform the gas exchange process is not so big than at high speed. the table 4 summarize the pumping loop value under each camshaft configuration. In this case Miller and iEGR improves the pumping loop changing from spending engine work to contribute to global BMEP in a low percentage not representative on the BSFC.

At high speed (B points), the table 5 shows the same trend of medium speed points. The pumping loop is improved but in this case reducing the amount of engine work spent to do the gases exchange process contributing in a big percentage to global BMEP and notoriously reducing the ISFC net. This is the main reason to have an improvement on the BSFC then with the two air management strategies tested efficiency at high speed is increased.

Full Load Results

The behavior at the extreme conditions of full load (A100 and B100) under the two air management configurations will be analyzed having in mind than the injection settings were modified in case of iEGR profile due to thermo-mechanical excessive conditions in the exhaust manifold. The injection settings modification are described on table 3.

Determination of intake valve closing conditions

As the two air management profiles were selected in these full load conditions through the simulation, experimentally now is possible validate the results obtained by the 1D GT Power model as is shown on Fig 19.

Combustion process study

The combustion study is performed in the same way than partial load by analyzing the oxygen concentration, the density, the HRR and the adiabatic flame temperature as observed on Fig 20.

In both cases the A100 and B100 the injection settings were defined for each profile as described on Table3, then the comparison must have in account the increase in injection pressure for iEGR camshaft profile. In a similar way than part load points the application of the air management strategies not only affect the trapped mass and consequently the density, but also affects the evolution of the oxygen concentration during combustion.

The main difference in combustion process between part load and full load conditions is remarked in the HRR and Tadb behavior. Centering on the HRR and based on the mixture process the rise up caused by the injection pressure increase on standard profile iEGR contrast with the HRR on standard profile under Miller injection conditions which has a lower injection pressure.

Focusing now on adiabatic flame temperature the improvement on mixture process due to injection pressure increases the adiabatic flame temperature for iEGR then the effect of this air management strategy is mitigated. In case of Miller strategy the results are similar than explained in part load.

Pollutant emissions and engine efficiency

The Miller and the iEGR camshaft profiles preserve the combustion efficiency reached by the standard profile as described on Fig 21. then the CO and HC pollutants are not increased in high measure.

With respect NO_x emissions, the adiabatic flame temperature described on combustion study explains the behavior shown on Fig 21. where are compared the two air management strategies with the standard miller one. In this case and due to injection pressure increase with iEGR profile, the NO_x reduction is not the highest with this profile.

In terms of soot emission the notorious increase caused by the iEGR profile in part load points is reduced due to improvement on mixture process achieved by the raise of the injection pressure. In case of Miller profile the richness near to the lift off is reduced because it decreases the density and the temperature of the gas in the combustion chamber. Despite this reduction in the soot formation conditions, there is a soot increment which confirms the affectation in the soot oxidation process in the same way than part load points.

Engine efficiency

The engine efficiency behavior at full load is explained in a similar way than part load points, in case of the A100 operating condition the fuel consumption shown on Fig 21. is not so affected by the Miller strategy which contrast with the iEGR profile where the increment on injection pressure helps to improve the mixture process making a faster combustion reducing the ISFC gross reflected on the BSFC.

In case of B100 the high speed point, the reduction on the pumping loop as explained on part load section added to the increase on injection pressure in the case of iEGR profile helps to have a notorious improvement in the ISFC gross, ISFC net and consequently on BSFC as observed on Fig 21.

Transient Study

The transient behavior comparison is described on Fig 22. In this case only the results of A50 and A75 operating condition are discussed due to the modification in the injection settings for iEGR strategy at full load. The well known tip-in transient test as proposed by Hagena et al.[22] and Kirchen et al.[23] is performed instantaneously changing the pedal position from 0% to 50% and 75% respectively.

The Torque, air mass flow and Opacity are recorded during 20s under each air management configuration obtaining the next results:

- The first studied variable is the Torque, in this case with both tested air management strategies (Miller and iEGR) there is an increase on the T90 time which is the required time to achieve the 90% of target Torque. The maximum increment is of 1.5s in iEGR strategy case in A75 mode.
- In case of air mass flow, the transient study is coherent with the results obtained in the steady test. There is a reduction of fresh air flow into the engine when is operated in any of different camshaft profiles different to the standard one.
- A notorious increase on opacity with the iEGR strategy and in less quantity with Miller profile is observed, the reduction in the A/F relation due to reduction in air mass flow leads to increase the richness of the mixture causing the raise of the opacity. In case of A50 engine point the increment is under 30% which is an acceptable increase of the opacity peak. The A75 engine condition must be re-calibrated in order to avoid the peak near to the 50% which is the maximum allowed peak value (Emission standard ISO 8178).

Conclusions

An experimental research about the engine performance under three different camshaft configuration has been carried out in a medium duty diesel engine. Two air management strategies with aim of NO_x reduction as the Miller profile or the iEGR were tested and the main conclusions after the study of the in-cylinder gas thermodynamic conditions, combustion process, pollutant formation and transient performance are now detailed:

- Important reduction in NO_x emissions was achieved with Miller timing as well as iEGR camshaft with a complementary increase of smoke and a preservation of combustion efficiency.
- The iEGR air management strategy shows the best potential in the NO_x emissions reduction being penalized by the increase of smoke and fuel consumption in medium speed points.
- Both tested air management strategies Miller as well as iEGR reduces the in-cylinder gas pressure and density compared with the standard camshaft being remarkable the reduction in the in-cylinder pressure for Miller profile.
- In terms of engine efficiency based on the engine fuel consumption, both camshaft have demonstrated the important influence of the pumping loop in the high speed operating conditions (B points) both in partial loads and under full load.
- The torque transient behavior is not drastically affected by the implementation of any of the air management strategies unlike the air mass flow which finally affects increasing the transient opacity.

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Table 1: Engine characteristics

Cylinders	6 in line
Strokes	4
Bore	110 mm
Stroke	135 mm
Displacement	7800 cm3
Nominal CR	16.5:1
Fuel injection system	Common-Rail

Table 2: Test cell instrumentation accuracy

Variable measured	Device	Manufacturer and model	Accuracy
In-Cylinder pressure	Piezoelectric sensor	Kistler 6125C	± 1.5 bar
Intake/Exhaust pressure	Piezorresistive sensor	Kistler 4045A10	± 25 mbar
Different Temperatures	Thermocouple	TC direct K type	± 2.5°C
Crank Angle,Engine Speed	Encoder	AVL 364	± 0.02CAD
Cam profile	Linear transducer	WA HBM	± 0.002mm
NO_x,CO,HC,O_2 Conc	Gas analyzer	Horiba Mexa One	4%
Smoke emission	Smoke meter	AVL 415	± 0.025 FSN
Diesel fuel mass flow	Fuel balance	AVL733S	± 0.2%
Air mass flow	Air flow meter	Sensyflow MT400	± 0.1%

Table 3: Injection settings of selected operating conditions at medium,high and full load

Inj Settings Partial Load

	NoI [-]	IP [bar]	SoI(Main) [CAD]
A50	2	340	-2
A75	3	375	-1
B50	2	590	-2
B75	2	680	-1.5

Inj Settings Full Load

	NoI [-]	IP [bar]	SoI(Main) [CAD]
A100 Miller	3	500	1
A100 iEGR	3	735	3
B100 Miller	2	820	3.5
B100 iEGR	2	950	3.5

Table 4: Pumping mean effective pressure comparison for A modes

Camshaft	A50		A75	
	PMEP [bar]	PMEP in BMEP [%]	PMEP [bar]	PMEP in BMEP [%]
Standard	-0.03	-0.3	0.082	0.5
Miller	0.04	0.4	0.22	1.4
iEGR	0.18	1.8	0.42	2.8

Table 5: Pumping mean effective pressure comparison for B modes

Camshaft	B50		B75	
	PMEP [bar]	PMEP in BMEP [%]	PMEP [bar]	PMEP in BMEP [%]
Standard	-1.38	-16.1	-1.68	-13.9
Miller	-0.82	-9.5	-0.93	-7.7
iEGR	-0.81	-9.4	-0.74	-6.1

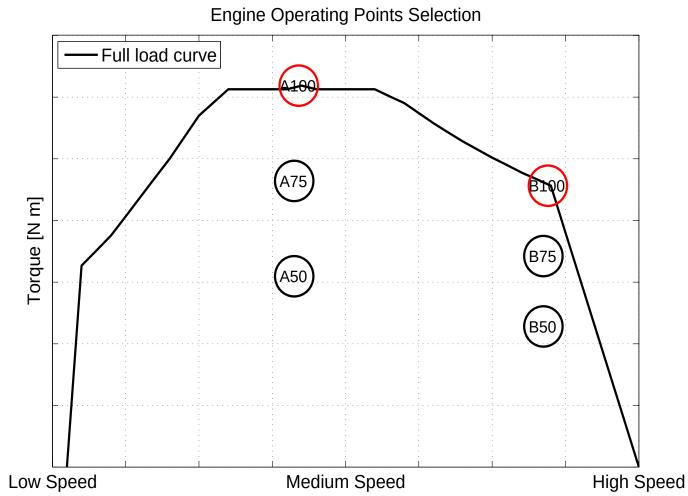

Figure 1: Engine points selection

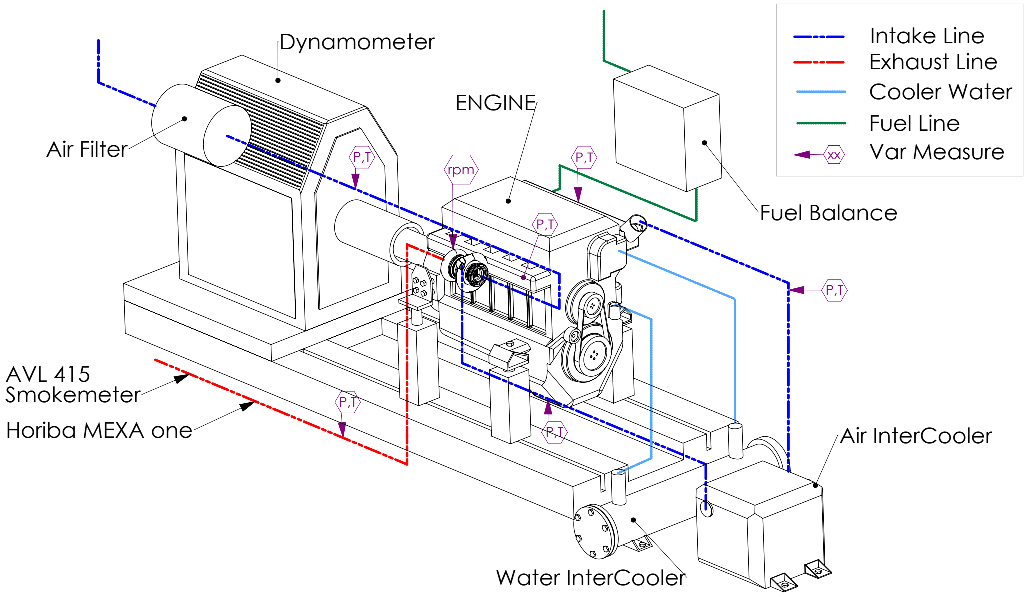

Figure 2: Test cell configuration and instrumentation

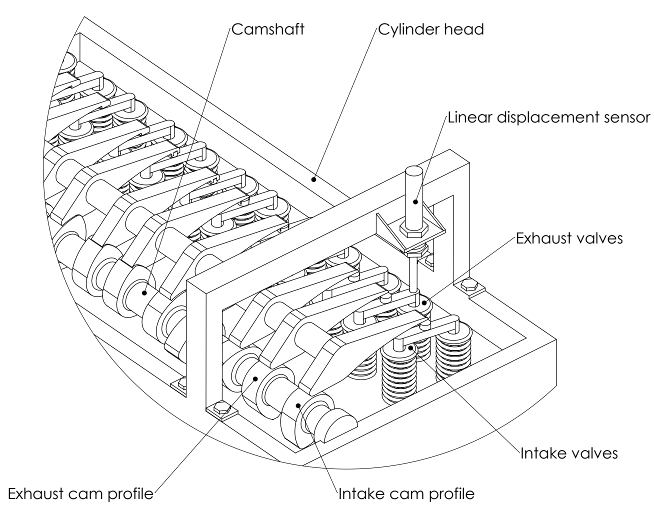

Figure 3: Camshaft profile measurement

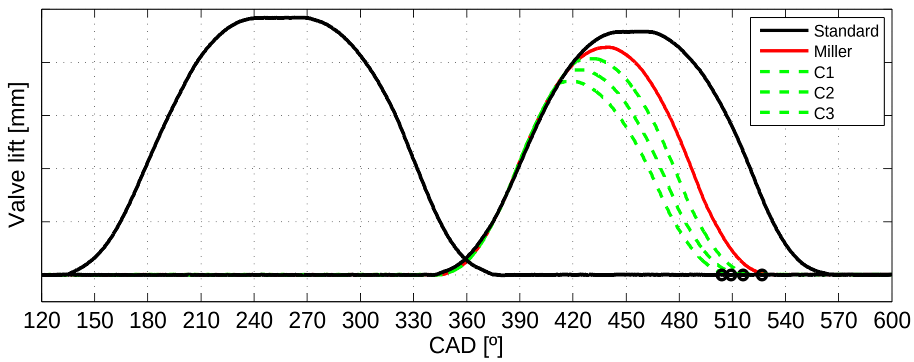

Figure 4: Miller camshaft profile and simulated options

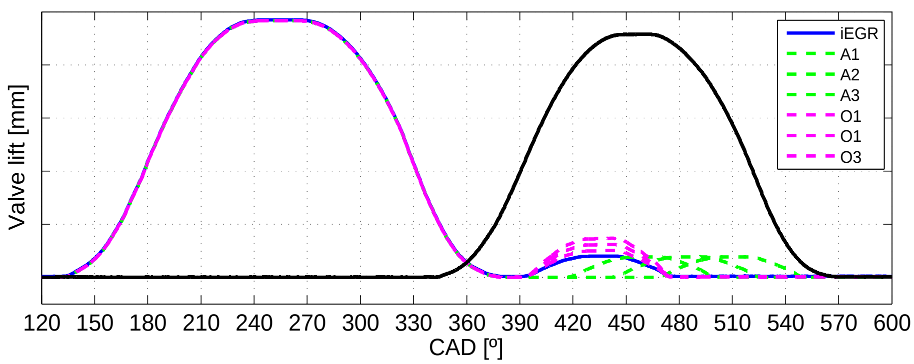

Figure 5: iEGR camshaft profile and simulated options

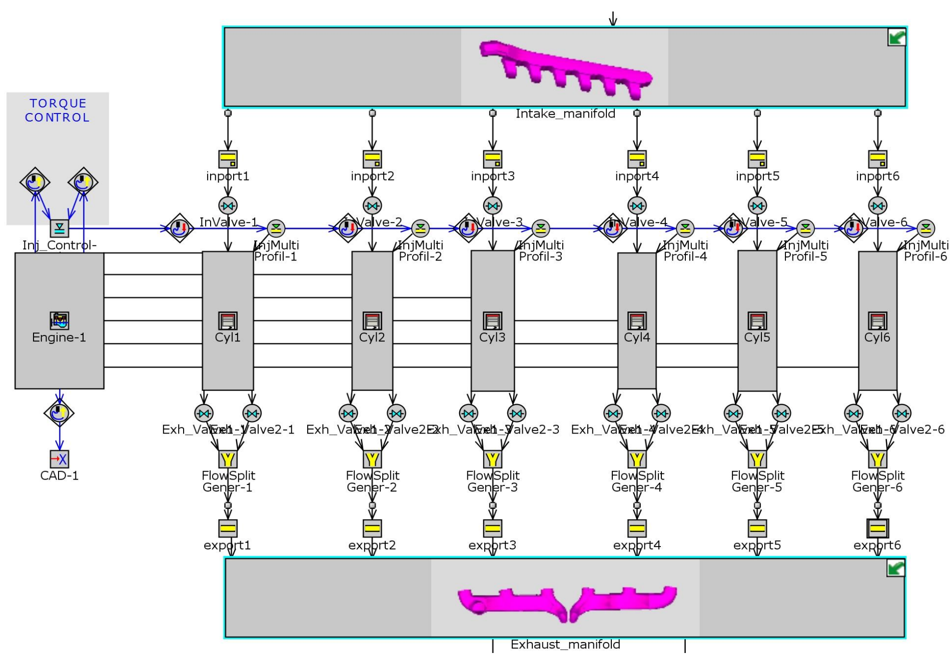

Figure 6: GT Power 1D Engine model

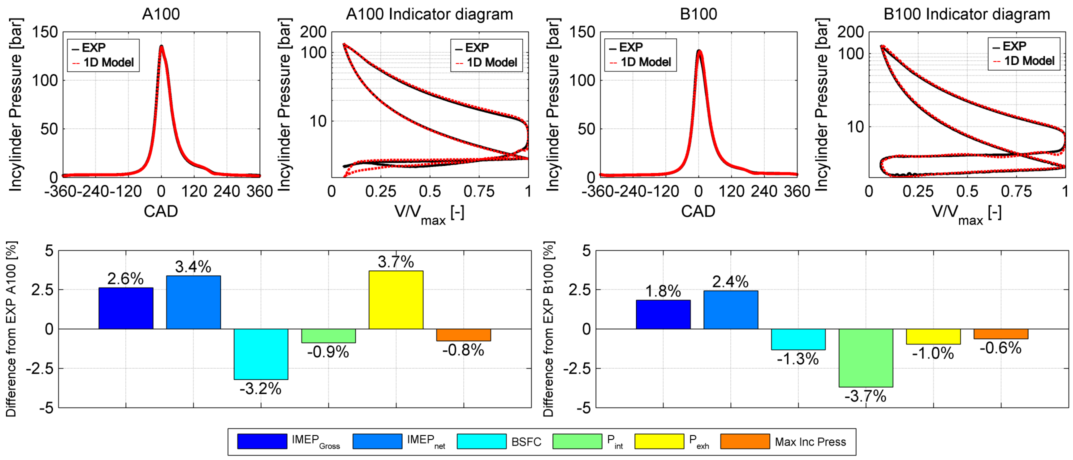

Figure 7: GT Power 1D validation

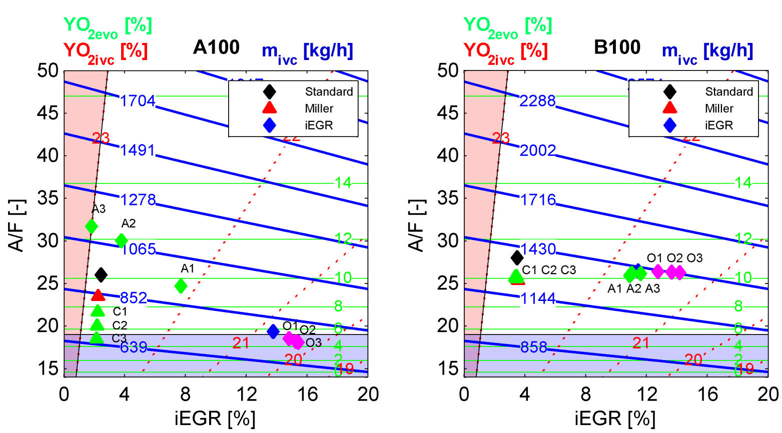

Figure 8: Intake valve closing conditions for Simulated strategies

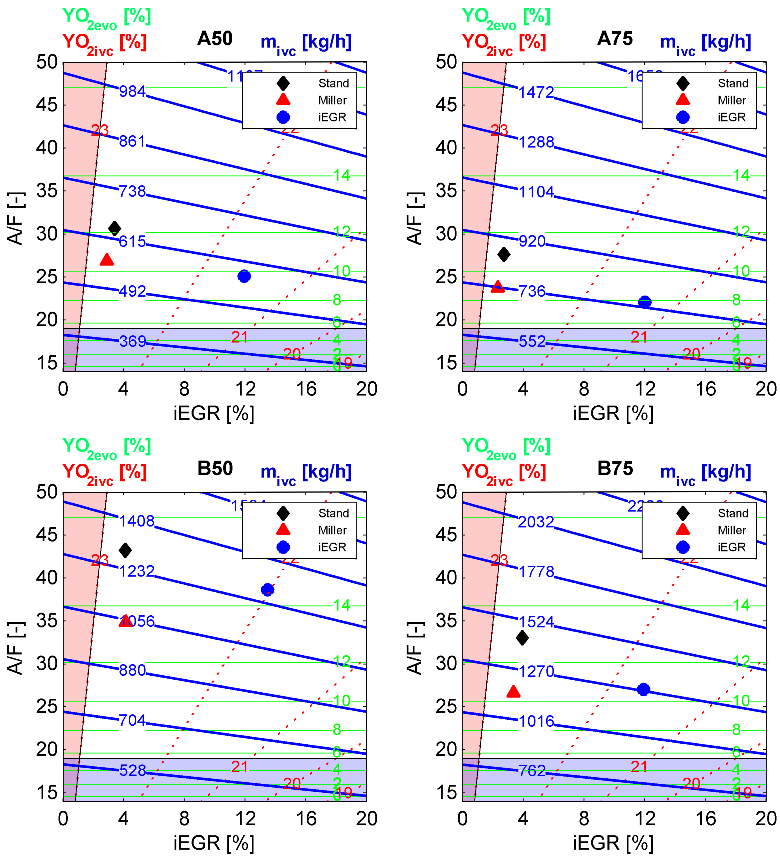

Figure 9: Air management influence on in-cylinder conditions at intake valve closing on part load points

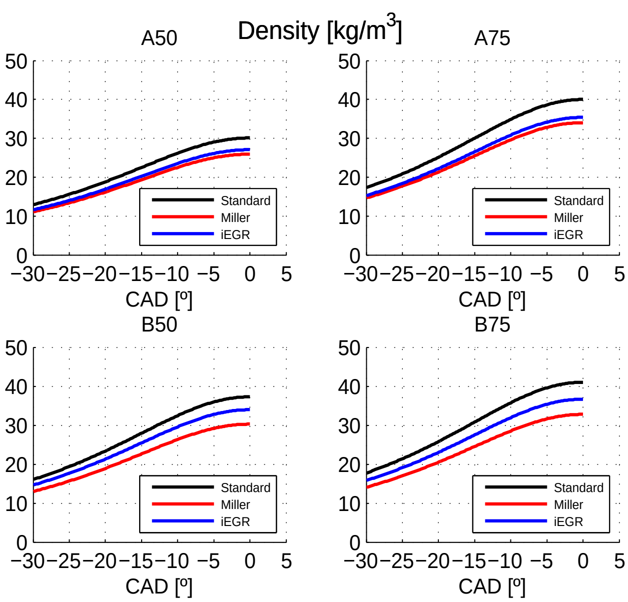

Figure 10: Density evolution at compression stroke before injection

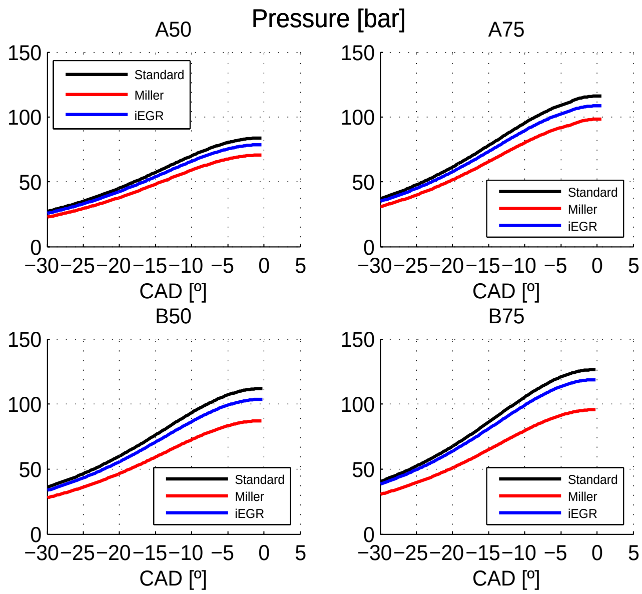

Figure 11: In-cylinder pressure at compression stroke before injection

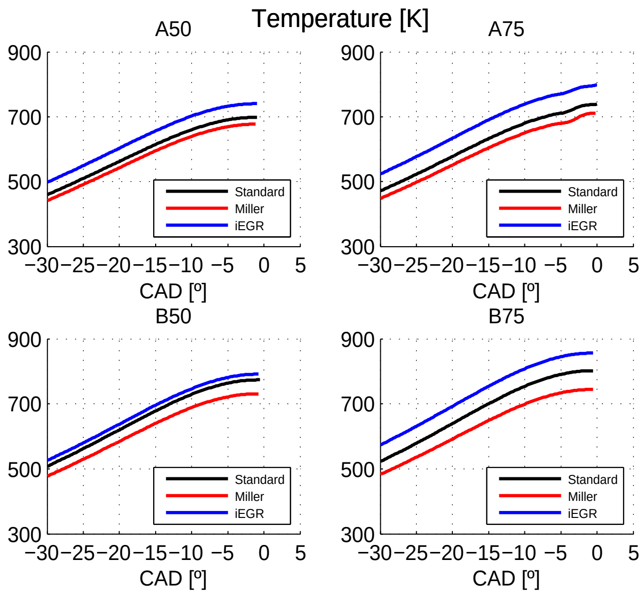

Figure 12: Temperature at compression stroke before injection

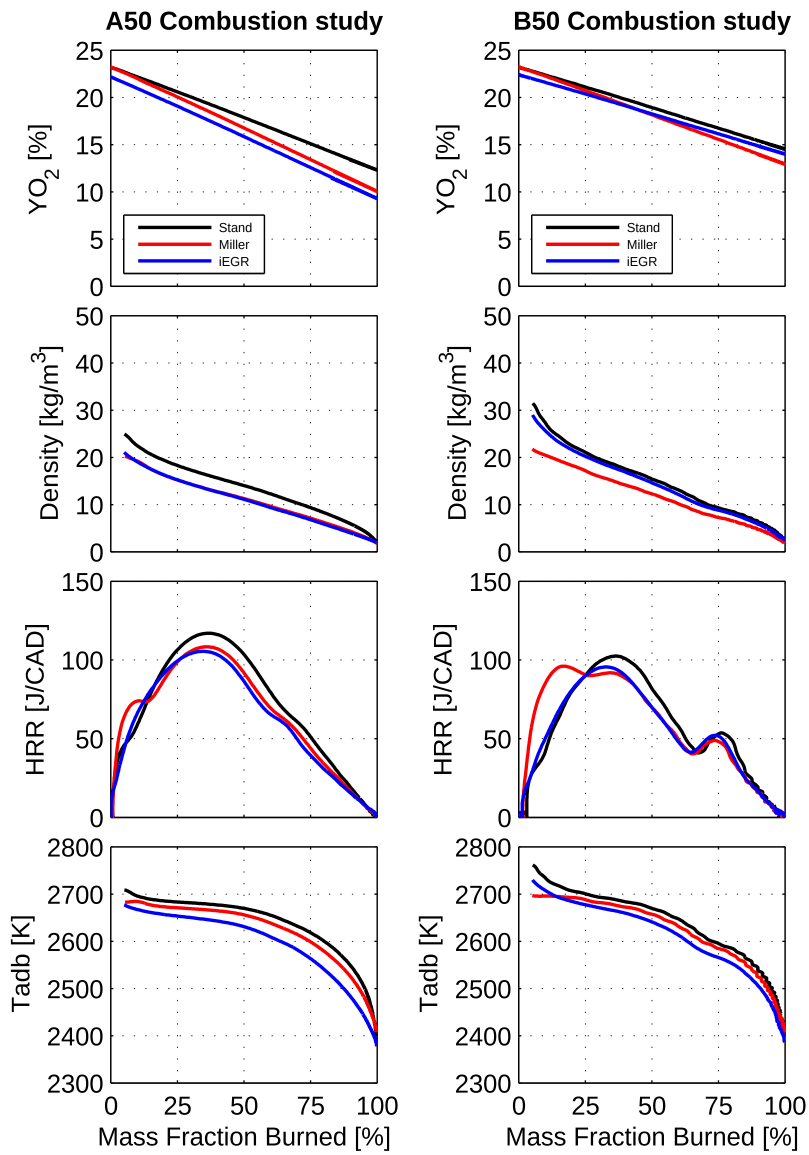

Figure 13: Combustion study at 50% of Load

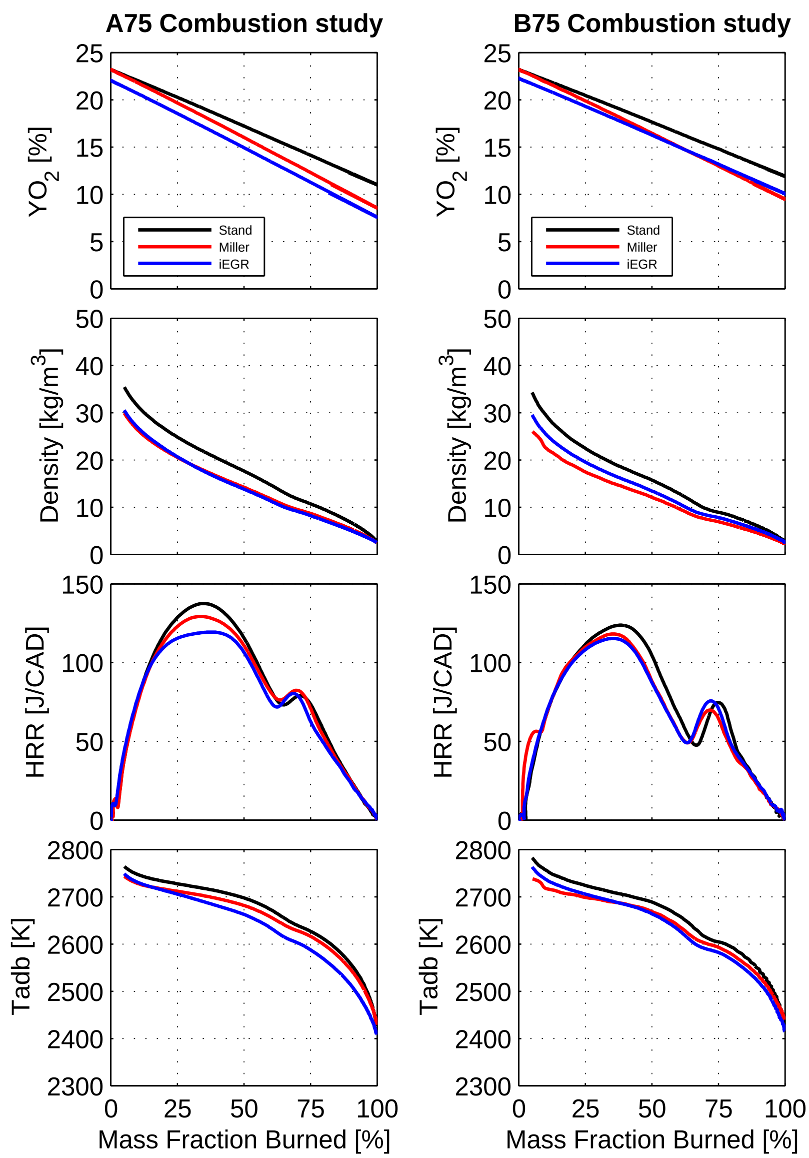

Figure 14: Combustion study at 75% of Load

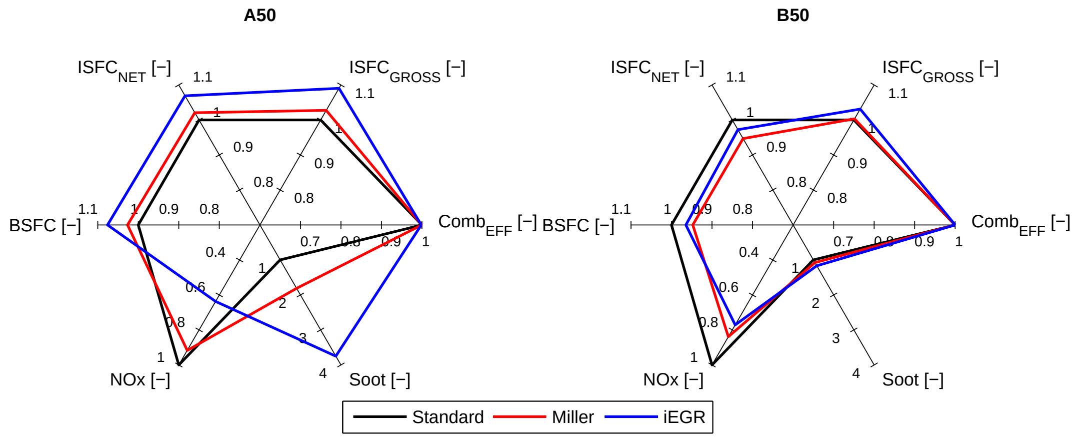

Figure 15: Pollutant Emissions and fuel consumption at 50% of Load

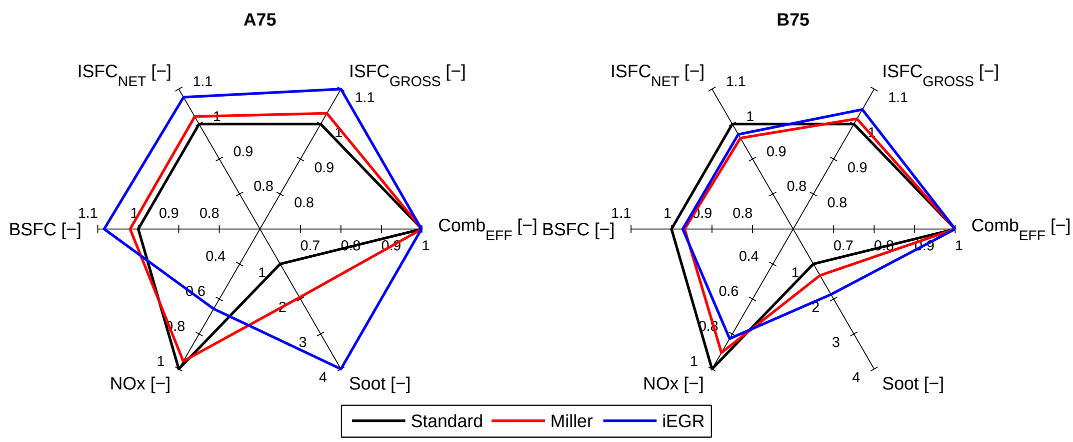

Figure 16: Pollutant Emissions and fuel consumption at 75% of Load

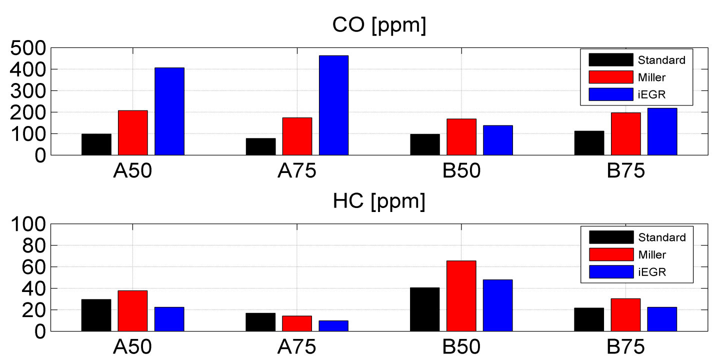

Figure 17: CO and HC emissions in part load operating conditions

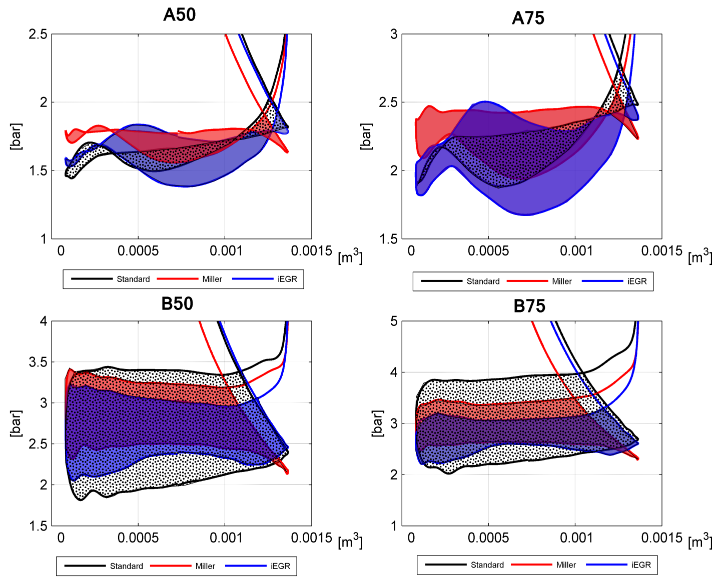

Figure 18: Pumping loop diagram at part load

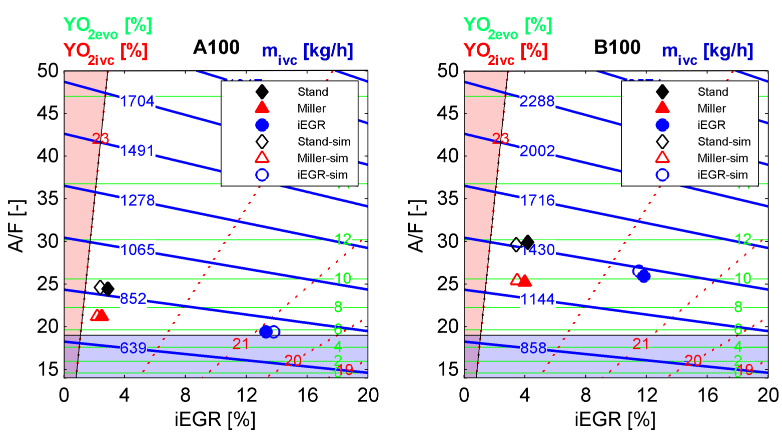

Figure 19: Air management influence on in-cylinder conditions at intake valve closing on full load points

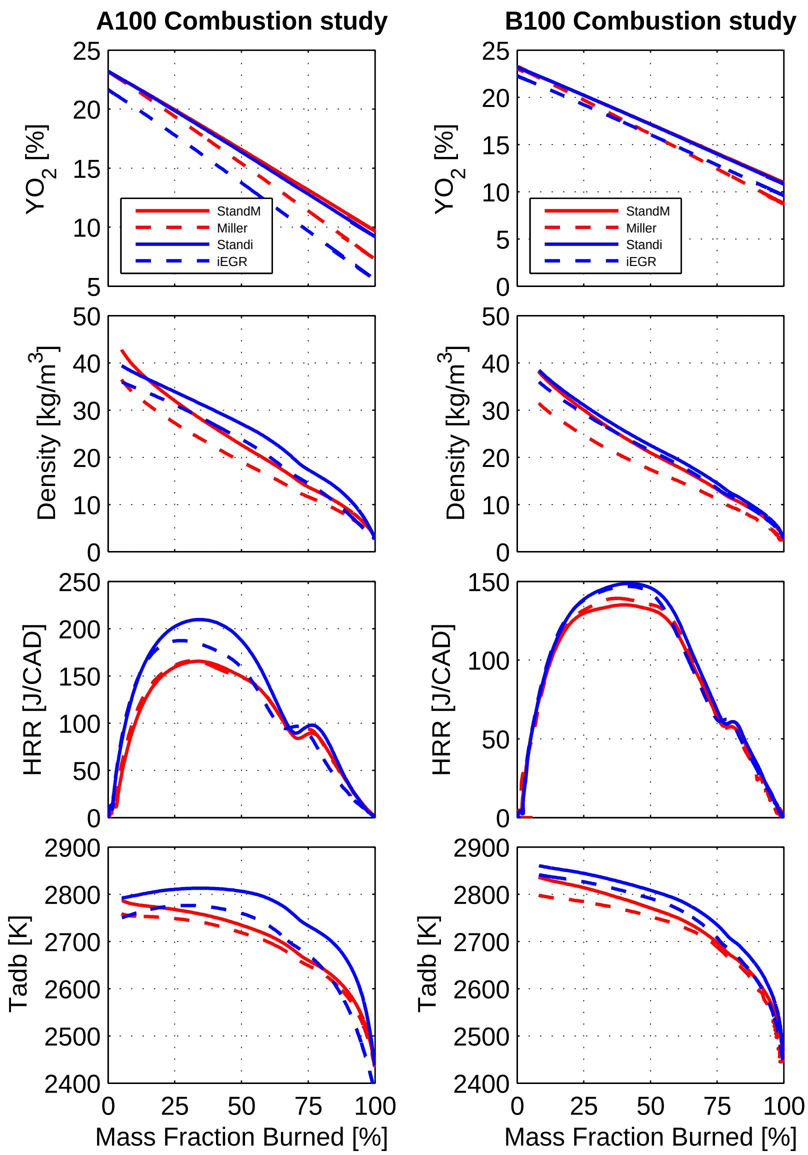

Figure 20: Combustion study at full Load

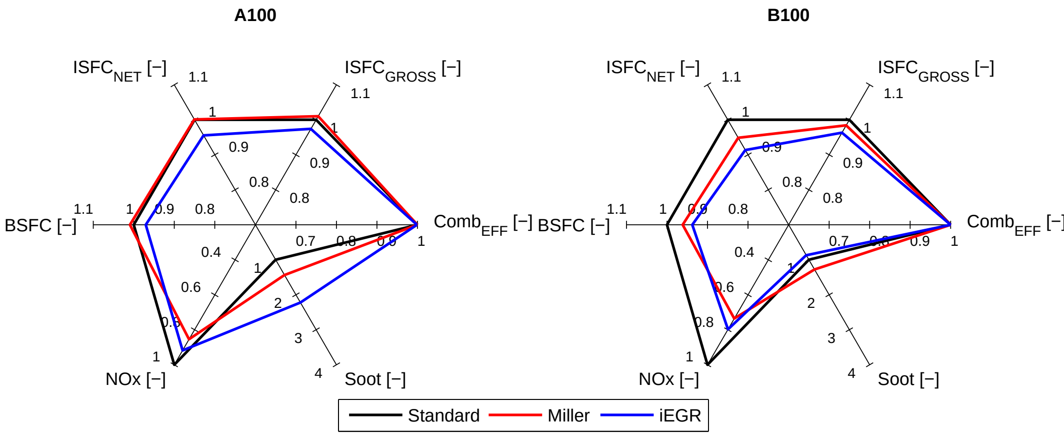

Figure 21: Pollutant Emissions and fuel consumption at full of Load

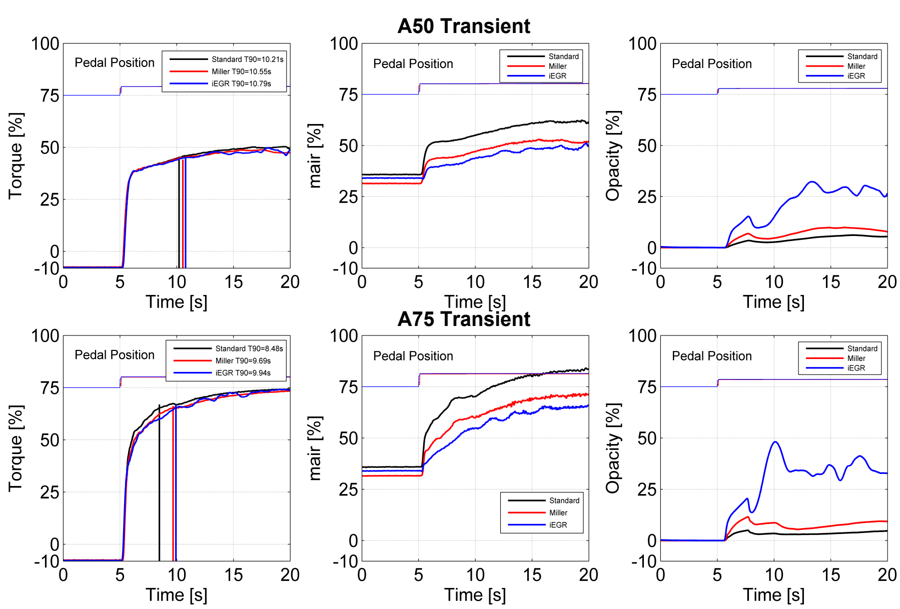

Figure 22: Transient response at A50 and A75 engine operating condition