

Comparison between Internal and External EGR Performance on a Heavy Duty Diesel Engine by Means of a Refined 1D Fluid-Dynamic Engine Model

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ABSTRACT

The potential of internal EGR (iEGR) and external EGR (eEGR) in reducing the engine-out NO_x emissions in a heavy-duty diesel engine has been investigated by means of a refined 1D fluid-dynamic engine model developed in the GT-Power environment. The engine is equipped with Variable Valve Actuation (VVA) and Variable Geometry Turbocharger (VGT) systems. The activity was carried out in the frame of the CORE Collaborative Project of the European Community, VII FP.

The engine model integrates an innovative 0D predictive combustion algorithm for the simulation of the HRR (heat release rate) based on the accumulated fuel mass approach and a multi-zone thermodynamic model for the simulation of the in-cylinder temperatures. NO_x emissions are calculated by means of the Zeldovich thermal and prompt mechanisms.

As a first step, the model has been calibrated and assessed on the basis of experimental tests carried out on four characteristic operating conditions within the WHSC (World Harmonized Stationary Cycle). The considered engine points have been used to identify the optimal engine parameters on the basis of a DoE approach.

As a second step, sweeps simulations centered on the previously considered operating points have been run for iEGR and eEGR in order to study the effects of these latter on combustion and NO_x formation processes at steady-state conditions.

Finally, a ramp from the WHTC (World Harmonized Transient Cycle) has been reproduced with both iEGR and eEGR modes and the engine performance have been compared in terms of fuel consumption, combustion and NO_x emissions.

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INTRODUCTION

The increasingly stringent emission regulations have led the automotive manufacturers to develop innovative techniques for the control of in-cylinder pollutants formation as well as to adopt more and more efficient after-treatment devices. SCR (Selective Catalytic Reduction) device systems are currently a very promising technique adopted in heavy duty diesel engines in order to reduce the NO_x tailpipe emissions [1]. However, major concerns derive from the efficiency reduction that occurs at low temperatures and to the need of urea injection. An effective reduction of in-cylinder NO_x formation could bring benefits in terms of both NO_x tailpipe emission reduction during the engine warm-up phase and urea injection.

The control of in-cylinder NO_x formation is usually performed by means of external Exhaust Gas Recirculation (eEGR), which is a well-established technique and is carried out by means of the recirculation of cooled exhaust gases into the intake manifold [2]. eEGR is effective in reducing the maximum temperature of the burned gases (that is responsible for the thermal NO_x formation), as it leads to an increase in the thermal capacity, to a reduction in the

oxygen concentration and to a dilution of the inducted charge. However, the recently developed Variable Valve Actuation (VVA) systems have allowed for the possibility of adopting additional NO_x reduction techniques. One of these latter is based on the Early or Late Intake Valve Closing (EIVC/LIVC). It allows for performing the Miller cycle and is often combined with eEGR [3, 4]. VVA also allows NO_x formation to be controlled by means of internal EGR (iEGR). Such a goal is fulfilled by means of a modified camshaft that either produces a secondary lift of the intake valve during the exhaust stroke (2IVO) or induces a secondary lift of the exhaust valve during the intake stroke (2EVO) [5]. iEGR can also be performed by means of a conventional camshaft, by adopting an early exhaust valve closing (EEVC) and modulating the valve overlap [5]. Among these techniques, the 2IVO method proved to generally produce the best results in terms of smoke emissions and ISFC [6] and is generally implemented on internal combustion engines [7]. A good potential of combined iEGR / Miller cycle strategies for heavy-duty applications has also been shown by numerical simulations in the past years [8].

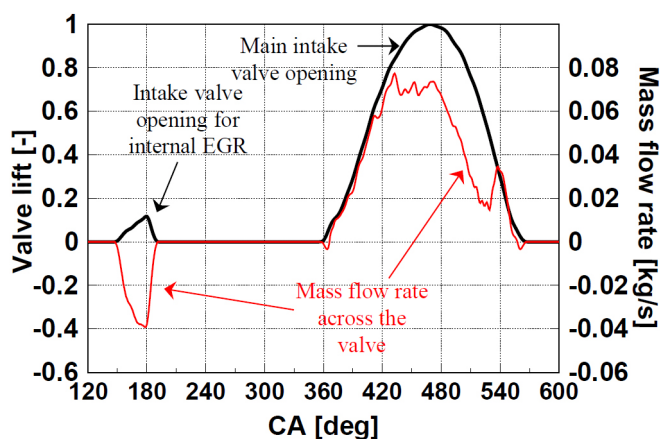


Figure 1. iEGR: example of intake valve lift and mass flow rate across the valve.

The present work aims at investigating into the performance of internal and external EGR for a 6 cylinder 8.7L diesel engine by means of a refined 1D fluid-dynamic engine model developed in the GT-Power environment. The investigation has been carried out over several engine operating conditions, ranging from low-load/low-speed to high-load/high speed points. The activity has been carried out within the CORE (CO₂ reduction for long distance transport) Collaborative Project of the European Community, VII FP, with the support of CRF (Centro Ricerche Fiat).

iEGR has been obtained by means of a secondary opening of the intake valve during the expansion stroke (2IVO approach) thus producing a backflow of the gases to the intake runner. The gases are reintroduced in the combustion chamber in the subsequent cycle. Compared to external EGR, internal EGR has a greater potential to allow for a cycle-to-cycle control of the exhaust gas fraction and NO_x formation. However, main concerns are related to the lack of cooling of the recirculated exhaust gases, which could penalize the NO_x reduction potential compared to external EGR.

A zero-dimensional heat release simulation model, based on the accumulated fuel mass approach, a multizone thermodynamic model and a NO_x formation submodel, that had been previously developed by the authors, have been integrated in the GT-Power simulation tool in order to allow for an accurate prediction of the combustion processes, of the in-cylinder temperatures and of the NO_x formation.

The implementation of the heat release simulation model was already presented and discussed in [9], whereas the integration of the multizone thermodynamic approach into the GT-Power engine model is detailed in the present paper.

ENGINE SETUP AND EXPERIMENTAL ACTIVITY

The main engine specifications are reported in [Tab. 1](#).

The engine is a six cylinder 8.7 L heavy-duty diesel engine, equipped with a Common Rail injection system and a VGT turbocharger. The engine is also equipped with Fiat Multiair® VVA (Variable Valve Actuation) system, which is capable of controlling the intake valve lift in terms of both opening and closing angles. This system allows

for realizing innovative NO_x reduction techniques, such as the early intake valve closing (EIVC) and the internal EGR. The latter is performed by means of a modified cam geometry, which allows a second intake valve lift to be actuated at the end of the expansion stroke. This causes a backflow of gases to the intake runner, which are hence reintroduced in the combustion chamber in the subsequent cycle. A qualitative example of iEGR is shown in [Fig. 1](#), which reports a normalized intake valve lift profile and the corresponding mass flow rate across the valve. The amount of iEGR trapped mass is controlled by means of the electric closure angle of the secondary opening lift, which will be referred to as “ ϕ_2 ”.

Table 1. Main engine specifications.

Displacement	8.7 L
Stroke	135 mm
Bore	117 mm
Connecting Rod	200 mm
Compression ratio	15.8:1
Number of cylinders	6
Injection system	Common rail, Bosch CRIN3, 1800 bar
Turbocharger	VGT type
Intake valve lift	VVA system

The experimental data have been acquired at a stationary test bench at CRF.

The test engine was instrumented with piezoresistive pressure transducers and thermocouples to measure the pressure and temperature at different locations, such as upstream and downstream from the compressor, turbine and intercooler, and in the intake manifold. Thermocouples have also been used to measure the temperatures in each of the six exhaust runners. Piezoelectric transducers have been installed to measure the pressure time-histories in the combustion chamber of the six cylinders.

The experimental tests that have been used for the model calibration, using both eEGR and iEGR techniques, have been carried out at 4 steady-state operating conditions of the WHSC (World Harmonized Stationary Cycle), i.e., modes 8, 9, 10 and 5 (see Tab. 2).

Table 2. Definition of the WHSC modes analyzed in this study.

Mode	Speed [%]	Load [%]
8	45	25
9	55	50
5	35	100
10	75	100

The main engine parameters (injection pressure, start of injection of the main pulse, intake manifold pressure) have been optimized on the basis of a Design of Experiment (DoE) approach, in order to find the best trade-off amongst fuel consumption, NO_x and soot emissions.

The main tests specifications are reported in Tab. 3 and Tab. 4 for the eEGR and iEGR modes, respectively. In the tables, p_f denotes the injection pressure, SOI_{main} the start of injection of the main pulse, p_{int} the intake manifold pressure, X_r the eEGR rate and ϕ_2 the electric closure angle of the iEGR intake valve lift. The values of p_f , SOI_{main} , p_{int} and ϕ_2 were expressed as differences with respect to the Case 1 for each mode, due to confidentiality reasons. The values of external EGR rate were evaluated on the basis of the CO₂ concentration measured in the intake manifold, downstream from the EGR mixing node, on the basis of the procedure reported in [10].

Table 3. Summary of the experimental eEGR sweep tests for the different modes.

Mode 8	Unit	Case 1	Case 2	Case 3	
p_f	bar	Ref	+200	+200	
SOI_{main}	deg BTDC	Ref	-1.2	-0.8	
p_{int}	bar	Ref	+0	+0	
X_r	%	1.3	9.6	17.4	
Mode 9	Unit	Case 1	Case 2	Case 3	Case 4
p_f	bar	Ref	+100	+200	+300
SOI_{main}	deg BTDC	Ref	-0.7	-0.7	-0.7
p_{int}	bar	Ref	+0.09	+0.18	+0.11
X_r	%	0.8	6.8	13.4	17.9
Mode 10	Unit	Case 1	Case 2	Case 3	Case 4
p_f	bar	Ref	0	0	0
SOI_{main}	deg BTDC	Ref	+0.1	+0.4	+0.8
p_{int}	bar	Ref	+0	+0	+0
X_r	%	2	4.7	9.2	13.5
Mode 5	Unit	Case 1	Case 2	Case 3	Case 4
p_f	bar	Ref	+0	+100	+200
SOI_{main}	deg BTDC	Ref	-2.3	-3.3	-0.3
p_{int}	bar	Ref	-0.02	-0.05	-0.1
X_r	%	0.4	2.9	5.4	8.3

Table 4. Summary of the experimental iEGR sweep tests for the different modes.

Mode 8	Unit	Case 1	Case 2	Case 3	
p_f	bar	Ref	+200	+200	
SOI_{main}	deg BTDC	Ref	-0.8	+0.6	
p_{int}	bar	Ref	+0.0	+0	
ϕ_2 iEGR	deg	Ref	+85	+95	
Residual fraction cyl1 (simulated)	%	3.8	20.5	28.6	
Residual fraction cyl3 (simulated)	%	3.7	13.6	17.0	
Mode 9	Unit	Case 1	Case 2	Case 3	Case 4
p_f	bar	Ref	-100	-200	-400
SOI_{main}	deg BTDC	Ref	-2.5	-2.8	-5.8
p_{int}	bar	Ref	+0.09	+0.17	+0.19
ϕ_2 iEGR	deg	Ref	+35	+50	+50
Residual fraction cyl1 (simulated)	%	3.8	7.1	15.5	15.6
Residual fraction cyl3 (simulated)	%	3.6	6.4	10.8	10.8
Mode 10	Unit	Case 1	Case 2	Case 3	Case 4
p_f	bar	Ref	-200	-200	-400
SOI_{main}	deg BTDC	Ref	+0	-2	-1.4
p_{int}	bar	Ref	+0.03	+0.11	+0.03
ϕ_2 iEGR	deg	Ref	+40	+50	+60
Residual fraction cyl1 (simulated)	%	3.7	6.0	13.9	19.6
Residual fraction cyl3 (simulated)	%	3.7	5.5	10.2	12.4
Mode 5	Unit	Case 1	Case 2	Case 3	Case 4
p_f	bar	Ref	-300	-400	-600
SOI_{main}	deg BTDC	Ref	-0.3	-0.1	+0.9
p_{int}	bar	Ref	+0	+0	+0.04
ϕ_2 iEGR	deg	Ref	+30	+40	+45
Residual fraction cyl1 (simulated)	%	2.8	6.4	9.3	16.8
Residual fraction cyl3 (simulated)	%	2.8	5.4	6.9	10.0

1D ENGINE SIMULATION MODEL

The engine was modeled at ICEAL-PT (Internal Combustion Engines Advanced Laboratories) of Politecnico di Torino with GT-POWER v7.3, a 1D simulation tool licensed by Gamma Technologies [11]. The model has been built according to the geometry marked by the technical drawings of the engine. Each system component has been properly characterized in terms of its specific geometric features such as length, diameter, bend radius, etc. Moreover, the model was validated

The geometric features of the inlet and exhaust pipes have been accurately measured at the engine test bench at CRF, in order to correctly reproduce the real experimental setup of the experimental tests which were used for the model calibration.

Figure 2. (cont.) GT-Power engine model map.

The HRR simulation model was presented by the authors in [12, 13, 14] and was integrated in the GT-Power model in [9]. However, a brief summary of the method is reported hereafter.

where $m_{f,inj}^*$ is the total injected fuel mass and H_L the lower heating value of the fuel.

The fuel burned mass fraction $x_{b,f}$ can be obtained by dividing the curve of chemical energy release Q_{ch} by the total chemical energy introduced into the combustion chamber:

$$x_{b,f} = \frac{Q_{ch}}{m_{f,inj}^* H_L} \quad (5)$$

The combustion rate coefficient of the main injection, i.e., K_{main} , can assume two distinct values corresponding to the premixed and diffusive phases respectively:

$$K_{main} = K_{main,p} \quad (t < t_{SOC,main} + \Delta t_{main,diff}) \quad (6)$$

$$K_{main} = K_{main,diff} \quad (t \geq t_{SOC,main} + \Delta t_{main,diff}) \quad (7)$$

The above procedure has been adopted in order to obtain a more accurate prediction of the premixed and diffusive burning phases of the main injection.

It can be observed that the approach requires the injection rate as an input. This is built considering a trapezoidal profile, and taking into account the nozzle opening delay (NOD), the nozzle closure delay (NCD), the slope of the injection profile and the maximum value of the injection rate [9]. These quantities are available through specific correlations, which are functions of the energizing time (ET) and injection pressure, worked out on the basis of the injector experimental data.

The K_j and τ_j coefficients were appropriately tuned in [9] on the basis of the Q_{ch} curves derived from the experimental in-chamber pressure trace, and physically consistent correlations have been derived and implemented in the GT-Power code.

For a generic injection pulse, the model starts from the rail pressure, SOI (Start of Injection), ET (Energizing Time) and DT (dwell time), which are known from the engine maps, then evaluates the NOD, NCD, the opening slope and the maximum rate of injection, and finally calculates the simplified injection rate, that is used to evaluate the Q_{fuel} terms in Eqs. (2-3). The heat release is then evaluated on the basis of Eq. (1) for each injection shot, using the abovementioned correlations for the K_j and τ_j coefficients.

It was demonstrated in [9] that the proposed HRR simulation model provides more accurate results compared to a conventional approach based on the interpolation of experimental HRR profiles, especially in transient conditions. Moreover, this approach allows the effects of different injection strategies (in terms of injection pressure, injection timings and quantities) on the combustion process to be accurately evaluated.

Multizone Thermodynamic Model

The multizone thermodynamic model that was integrated in GT-Power is a refinement of the real time three-zone combustion model presented by the authors in [15], which was in turn derived from the simplification of the premixed-diffusive multizone diagnostic model previously developed by the authors and presented in [16]. The equations have been developed in order to be solved in closed form, following the approach explained in [15], thus allowing for attaining a very low computational time for the thermodynamic model, with benefits in terms of the simulation time required by the complete GT-Power engine model.

The combustion chamber has been divided into: a vapor-fuel zone (f), an unburned gas zone (u), made up of fresh-air, residual gas and EGR, and n burned gas zones (b,j) obtained from a globally stoichiometric combustion process. The diffusive burned gas zones are generated at specific crank angles, which are selected so as to divide the heat release curve into a user-defined number of equal portions. The last generated zone b,n is fed with gas from the vapor-fuel and unburned gas zones so as to reach stoichiometric conditions, whereas it is assumed that the other burned gas zones b,j evolve at constant mass.

The splitting of the burned gases into several zones leads to a better prediction of the temperature gradients in the combustion chamber, which in turn allows a more accurate calculation of the NOx concentration to be obtained.

The total trapped mass is needed in order to estimate the mass and oxygen concentration of the unburned gas zone. The latter is constituted by the trapped air mass, trapped EGR mass and residual gas mass:

$$m_{trap}^* = m_a^* + m_{res}^* + m_{EGR}^* \quad (8)$$

where the asterisk denotes a constant quantity in time.

The cycle-by cycle and cylinder-by-cylinder values of the three mass contributions are evaluated by GT-Power and are available in the thermodynamic model.

The combustion process can be modeled as follows. The in-cylinder content from the IVC (Intake Valve Closing) to the SOI (Start of Injection), is made up of unburned gas, which is considered as a homogeneous mixture of fresh air, EGR and residual gas. A vapor-fuel zone is generated after the SOI, as a consequence of the heating of the injected fuel. The parcels from the vapor fuel zone react with the parcels from the unburned gas zone at stoichiometric ratio, thus generating the burned gas zones (b, j) after the SOC (start of combustion). A user-defined number of burned gas zones is generated, so as to split the mass fraction profile into equal portions. The number of these zones is selected as a trade-off between the computational effort and the accuracy in the estimation of the NOx emissions. The optimal value is of the order of 10-12 [16].