Applied Thermal Engineering 29 (2009) 2043–2050

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Influence of EGR unequal distribution from cylinder to cylinder on NOx–PM trade-off of a HSDI automotive Diesel engine

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ARTICLE INFO

Article history:
Received 26 May 2008
Accepted 22 October 2008
Available online 14 November 2008

Keywords:
Diesel engine
Pollutant emissions
Exhaust gas recirculation (EGR)
Unequal cylinder-to-cylinder EGR distribution
EGR maldistribution
NOx–PM trade-off

ABSTRACT

The influence of cylinder-to-cylinder variation in EGR distribution on the NOx–PM trade-off (while varying EGR rate) is studied on an automotive high-speed direct injection Diesel engine. Experiments have been conducted on an engine test bench with and without air-EGR mixer and demonstrate that variations in cylinder-to-cylinder EGR distribution results in a deteriorated NOx–PM trade-off (increased NOx emissions level at a given PM emissions level, or increased PM emissions level at a given NOx emissions level) compared to the well mixed configuration with equal EGR rate for all the cylinders. A qualitative study as well an original experiment is conducted to explain this emissions increase induced by unequal distribution of EGR. When recirculating hot exhaust gases, the emissions increase is due to cylinder-to-cylinder variations in intake gas composition and temperature.

1. Introduction

Future emissions regulations like EURO 6 in Europe force Diesel engine manufacturers to find ever more complex ways to reduce exhaust gas pollutant emissions, in particular NO*x* and particulate matter (PM) emissions. Exhaust gas recirculation (EGR) into the engine intake is an established technology to reduce NO*x* emissions [1,2]. The decrease of NO*x* emissions with EGR is the result of complex and sometimes opposite phenomena occurring during combustion [3–10].

At the same time, the decrease in combustion temperatures and oxygen concentration while increasing EGR rate reduces both soot production in the spray core and soot oxidation in the diffusion flame around the jet [11]. Thus the final impact of EGR on PM emissions is complex and is the result of contradictory phenomena. In the conventional Diesel high-temperature combustion (HTC), the increase of EGR rate (at constant boost pressure) is accompanied by an increase of PM emissions, resulting in a trade-off between NO*x* and PM emissions while varying EGR rate [4,12–15].

Moreover, practical EGR systems often lead to EGR unequal distribution from cylinder to cylinder, air and EGR being imperfectly mixed. This phenomenon has been studied by various researchers [16–24]. By measuring CO_2 instantaneous concentration at each inlet port during the intake stroke owing to mid-infrared laser absorption spectroscopy, Green [16] has demonstrated that even when operating at a steady condition, the engine's EGR system can produce large temporal variations in the EGR concentration within the flow of fresh charge during the intake stroke, that are different for each cylinder. Furthermore, CFD analyses [17–18,20–21] have demonstrated that a standard engine's EGR system results in a highly stratified concentration field within the inlet manifold. Many experimental and numerical studies [17,19,23–24] have proposed improved inlet manifolds or air–EGR connections in order to improve cylinder-to-cylinder EGR distribution.

If some studies have shown that cylinder-to-cylinder variations in EGR can lead to higher NO*x* and PM emissions compared to a configuration where the EGR is equally distributed among all cylinders [17], the influence of on the NO*x*–PM trade-off (while varying EGR rate) has not been experimentally studied in details or explained. Thus, the aim of this study is to quantify and explain the influence of this phenomenon on the NO*x*–PM trade-off (while varying EGR rate at constant boost pressure) of an automotive HSDI Diesel engine.

2. Experimental set-up

2.1. Test engine and operating conditions

Experiments are conducted on a 2.0 l constant-moderate-swirl, water-cooled, turbocharged inter-cooled HSDI Diesel engine, equipped with a variable geometry turbine (VGT) and a high pres-

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doi:10.1016/j.applthermaleng.2008.10.017

Nomenclature

(CO_2)	CO_2 concentration (%)
D	volume flow ($m^3\,s^{-1}$)
$\dot{m}$	mass flow ($kg\,s^{-1}$)
$\dot{n}$	molar flow ($mol\,s^{-1}$)
NO*x* (g/h)	NO*x* emissions ($g\,h^{-1}$)
P	pressure (bar)
PM (g/h)	PM emissions ($g\,h^{-1}$)
T	temperature (K)
X_{egr}	EGR ratio (%)
Greek letters	
ρ	density ($kg\,m^{-3}$)
Subscripts	
a	air
cool	engine coolant
ex	exhaust
egr	related to recirculated exhaust gases
in	inlet
mix	after mixing with recirculated exhaust gases

Abbreviations

BMEP	brake mean effective pressure
BSFC	brake specific fuel consumption
DI	direct injection
EGR	exhaust gas recirculation
EUDC	extra urban driving cycle
FSN	filter smoke number
HP	high pressure
HSDI	high speed direct injection
HTC	high temperature combustion
LTC	low temperature combustion
PM	particulate matter
ROHR	rate of heat release
UDC	urban driving cycle
VGT	variable geometry turbine

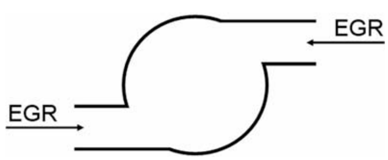

Fig. 1. Axial section (with respect to inlet duct) of air–EGR mixer.

sure (HP) water-cooled EGR loop (recirculated gas are taken upstream of the turbine and introduced downstream of the compressor). Engine specifications are given in Table 1.

When opening EGR valve to increase EGR flow rate at a fixed VGT vanes position, boost pressure is reduced because of a reduced exhaust gas flow through the turbine. Boost pressure is maintained constant by closing VGT vanes when opening EGR valve. Thus, both EGR flow rate and boost pressure are controlled simultaneously.

An air–EGR mixer was developed to ensure that air and recirculated gases are perfectly mixed to suppress cylinder-to-cylinder variations in EGR quantity. EGR gases are introduced into the main inlet duct tangentially, thus creating a vortex (Fig. 1). Moreover, when using the air–EGR mixer, the volume of inlet duct between air–EGR mixer and inlet manifold is increased in order to limit the temporal variations in the EGR concentration. The influence of EGR unequal distribution on NO*x*–PM trade-off is thus obtained by comparing engine-out NO*x* and PM emissions with and without the air–EGR mixer (Fig. 2). Inlet air temperature T_a (after intercooler) is controlled separately.

Table 1
Specifications of test engine.

Type	Turbocharged (VGT), inter-cooled
Compression ratio	18:1
Number of cylinders	4
Number of valves per cylinder	4
Combustion chamber type	Re-entrant bowl-in-piston
Injection system	Common-rail piezoelectric 2nd generation
Number of injection holes	7
Injection nozzle diameter (mm)	0.150
Maximum injection pressure (bar)	1600
Fuel	Diesel

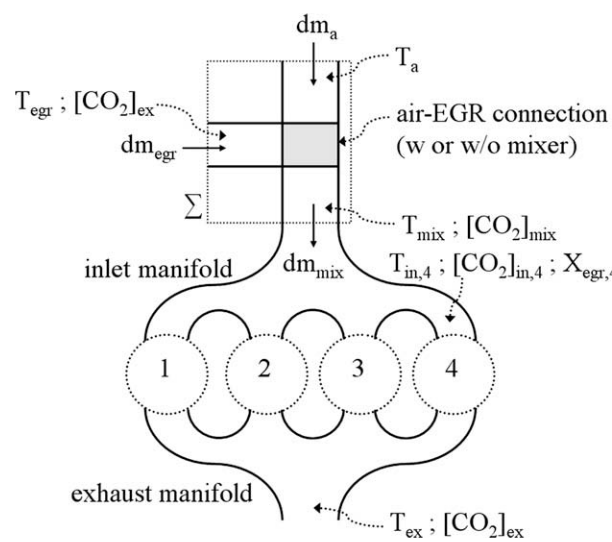

Fig. 2. Configuration of engine inlet.

Table 2
Operating conditions.

Point	Engine speed (rpm)	Pilot quantity (mg/stroke)	Principal quantity (mg/stroke)	BMEP (bar)	P_{rail}(bar)
A	1450	1.5	17.7	5.5	700
B	1870	1.2	14.5	4.0	750

The study is conducted at part load and low load conditions (operating points *A* and *B*, respectively), such as those encountered in the European emissions test cycle for light-duty vehicles – composed of four urban driving cycles (UDC) and one extra urban driving cycle (EUDC). The corresponding engine speed, pilot and main injection quantities, brake mean effective pressure (BMEP) and rail pressure are given in Table 2. For each operating point, injection quantities are held constant, and thus the BMEP is little changed for the various operating conditions (use of air–EGR mixer, variation of EGR rate).