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Research Paper

A novel approach for exhaust gas recirculation stratification in a spark-ignition engine

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ABSTRACT

Over the years, various stratified exhaust gas recirculation (EGR) concepts have been proposed and studied to increase the total amount of EGR that could be trapped within the cylinder so as to decrease NO emissions while lessening the negative impact of high dilution. The different concepts normally involved modifying the intake manifold so as to sequentially feed the cylinder with air and EGR or to create an asymmetrical EGR supply in the intake ports. The present work proposes a novel proof of concept aimed at achieving a stratified EGR in a spark-ignition engine. For the first time, stratified EGR is done by directly injecting exhaust gas through the cylinder head using solenoid valves controlled by an ECU. By this way, the proposed approach allows EGR injection at any time during the intake and compression stroke. The proposed concept was experimentally implemented by modifying a single-cylinder spark-ignition engine. Such an approach allows evaluating EGR stratification with respect to combustion, fuel consumption and emissions, at higher speed and load than previously reported with other stratified concepts. Three different EGR injection timings were evaluated and compared to a homogeneous EGR strategy. It has been possible to simultaneously lower fuel consumption by 6 % at 2100 RPM and high EGR rate while producing 2.6 % less NO emission when compared to a homogeneous EGR. Moreover, by injecting EGR directly in the cylinder allows enhancing the combustion process, as measured by a shorter combustion duration, by 17 % at 2500 RPM with a 20 % EGR-rate. The proof of concept has shown that stratified EGR is possible at high load and that EGR injection timing needs to be retarded with increasing engine speed for better performance.

1. Introduction

Gasoline spark-ignition engines are subjected to stringent pollutant emissions and fuel economy regulations, and exhaust gas recirculation (EGR) is a key technology that provides various pathways to decreasing the environmental impact of the internal combustion engine. For example, EGR influences the combustion process in three different ways [1]: 1) it increases the heat capacity of the exhaust gas mixture (through the thermal effect), by lowering the combustion temperature, and thus decreasing nitrogen oxide (NO) emissions; 2) it has a dilution effect as it decreases the oxygen concentration in the mixture, and 3) it has a chemical effect on soot formation through the suppression of the formation of large aromatic sheets [2]. Moreover, the use of hot EGR reportedly decreases particulate emissions in both port fuel and direct injection engines by increasing the temperature of the mixture, which fosters fuel evaporation and the fuel–air mixing process [3]. EGR also improves fuel economy by reducing pumping loss at part load [4], and can also be used to suppress end-gas auto-ignition and to mitigate knocking if cooled [5]. For example, this latter function has been demonstrated successfully with a spark-ignition engine fed with propane with which cooled EGR was effective to mitigate end-gas autoignition [6] and thus knocking. Even hot EGR has been shown to be beneficial in allowing the use of natural gas mixed with hydrogen in a high compression ratio spark ignition engine, to create an efficient and clean natural gas engine [7]. Overall, EGR has been valued as a low-cost alternative that can help achieve high-efficiency spark-ignition engines [8].

In a conventional or external EGR system, the exhaust gas is taken from the exhaust system and is transported to the intake manifold, next to the throttle body, in order as to create a homogeneous air-exhaust gas mixture. In that configuration, the pressure difference between the exhaust and intake is the driving force required to recirculate the hot gas while a valve allows controlling the amount of exhaust gases (EG)

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Nomenclature

ρ_{N2}	Nitrogen density at reference state (kg/m^3)
σ_{IMEP}	Standard deviation of the indicated mean effective pressure (kPa)
$\left(\frac{dP}{d\theta}\right)_{max}$	Maximum pressure rise during combustion (kPa/deg)
BMEP	Brake mean effective pressure (kPa)
C	Sonic conductance (m^4·s/kg)
COV of IMEP	Coefficient of variation of indicated mean effective pressure (%)
dt	Time interval per angle degree (s)
dP	In-cylinder pressure variation (Pa)
dV	In-cylinder volume variation (m^3)
$\frac{dQ_{net}}{d\theta}$	Instantaneous net heat release (J/deg)
EGR-rate %	EGR rate (%)
EI	Emission index of specie i (g/kg of fuel)
$\overline{IMEP}$	Average indicated mean effective pressure (kPa)
IMEP	Indicated mean effective pressure (kPa)
M_i	Molar weight of specie i (kg/mole)
M_f	Molar weight of isooctane (kg/kmole)
m_{N2}	Nitrogen mass (kg)
$\dot{m}_{air}$	Mass of air (kg)
$\dot{m}_f$	Mass of fuel (kg)
$P_{cyl}(\theta)$	In-cylinder pressure at each crank angle (Pa)
P_{N2}	Nitrogen pressure (Pa)
RPM	Engine speed (rev/minute]
SFC	Specific fuel consumption (kg/kW-h)
V	In-cylinder volume (m^3)
$\dot{W}$	Engine power (W)

Greek symbols

α	number of carbon atoms in a molecule of gasoline
θ	crank angle degree
γ	heat capacity ratio
λ	air-to-fuel equivalence ratio

Abbreviations

$\frac{\partial f}{\partial x_j}$	Derivative of calculated parameter f with respect to measurement x_j
0-10MFB	0-10% mass fraction burned
10-50MFB	0-50% mass fraction burned
10-90MFB	10-90% mass fraction burned
50-90MFB	50-90% mass fraction burned
A/F	Air-to-fuel ratio
BDC	Bottom dead center
CFD	Computational fluid dynamics
CO	Carbon monoxide
EG	Exhaust gas
EGR	Exhaust gas recirculation
H-EGR	Homogeneous EGR
MFB	Mass fraction burned
N	Number of observations
N_2	Nitrogen
NO	Nitrogen oxide
S.T.	Spark timing
STR50	Stratified EGR with injection beginning 50 deg. Before TDC
STR180	Stratified EGR with injection beginning 180 deg. Before TDC
STR215	Stratified EGR with injection beginning 215 deg. Before TDC
s	Global variance
TDC	Top dead center
THC	Total unburnt hydrocarbon
U_A	Uncertainty of type A
U_B	Uncertainty of type B
U_C	Combined uncertainty
WOT	Wide open throttle
x_i	Molar concentration of specie i
x_{CO}	Molar concentration of CO
x_{CO2}	Molar concentration of CO_2
x_{HC}	Molar concentration of THC
$\overline{x}$	Mean value of all tests
$\overline{x}_N$	Mean value of each test

recirculated. Contrary to homogeneous EGR, the stratification approach attempts to separate the exhaust gas from the air within the cylinder in order to achieve an EG gradient within the mixture. One way stratification is achieved is by creating an asymmetrical EG supply using an independent intake runner per intake valve. The configuration allows feeding each intake valve with different air-EG mixtures, in a 2 intake valves per cylinder engine. Such an approach was studied by Fuyuto et al. [9], using a single-cylinder diesel engine having 2 intake valves and where one intake port was fed with a 50 %-50 % air-EGR mixture, while the second intake port was for air. In that case, a 20 % decrease in soot production was observed versus homogeneous EGR. Recently, a similar approach using a helical intake port on a single-cylinder diesel engine allows creating EGR stratification in the cylinder [10]. They reported lower NO_x and particle mass at the exhaust with the stratified approach when compared with a conventional EGR. Park et al. [11] also studied a stratified EGR concept in a diesel engine based on a two-step piston design; in this set-up, a higher EGR concentration is present in the upper part of the piston (first step), at which the fuel spray is also aimed, and the combustion first appears, while a lower EG concentration is found at the bottom of the second piston cavity (second step). This approach allows to simultaneously decrease NO and soot emissions. Choi et al. [12] twinned a similar two-step piston with an asymmetrical EGR supply. The experimental results showed a simultaneous decrease of NO_x (7 %) and particulate matter (23 %), while positive effects on CO and HC were also noted. Another recent approach has been proposed and studied numerically by Bao et al. [13] and consists on the insertion of a small pipe, within the helical intake port, to deliver EGR at the intake valve of a diesel engine. The influence of the EGR pipe outlet circumferential position around the valve stem on the stratification was studied. From numerical results, it has been found that based on the pipe outlet location, different types of stratification can be achieved while creating an in-cylinder radial stratification with a high EGR concentration near the cylinder wall allows reducing soot and NO_x emissions.

Stratified EGR approach was also developed by Han and Cheng [14] for a spark-ignition engine. In their configuration, both intake ports are split into two using a partition, and are designed to generate a strong tumble. With this configuration, a lateral EG confined on each side of the fuel–air mixture is obtained in the cylinder. Under a high dilution level (close to 35 %), their stratified EGR approach produced a higher gross indicated mean effective pressure (IMEP), a lower coefficient of variation (COV) of gross IMEP and a faster fully developed combustion, as compared to homogeneous EGR. Woo et al. [15] also used a spark-ignition engine and achieved in-cylinder EGR stratification by timing the beginning and duration of the EG supply in the intake manifold with respect to the intake valve opening. This approach thus allowed to supply EG under closed intake valves up to the first half of the intake stroke or from the intake valve opening to the bottom dead center (BDC) of the intake stroke, etc. They observed a shorter combustion duration when a stratified EGR was used, whereas NO_x emissions were similar between the different stratification levels achieved by such a system. Other configurations, based on a two-intake port spark-ignition engine, include using only a single port to alternatively let EG and intake air into the cylinder during the intake stroke [16] or to use both intake ports for intake air supply during part of the intake stroke, after which the fresh air supply is stopped and an external port is activated and fed EGR in only one of the intake ports [16]. In all cases, a radial stratification of different intensities was achieved based on numerical simulations. They reported an 88 % decrease in NO_x emissions experimentally as compared to homogeneous EGR when EG was first supplied to the cylinder for part of the intake stroke followed by a supply of air.

In general, the above techniques aim to achieve EGR stratification in a bid to increase the amount of EG within the cylinder without the negatives associated with a homogeneous high EGR level; these include an increase in the local heterogeneity in the fuel-to-air mixture, combustion instabilities, a lower fuel efficiency, and in some cases, misfires [17]. Moreover, as the engine approaches the wide-open throttle (WOT) condition, a homogeneous EGR decreases the engine power as EGR is detrimental to the intake air density entering the cylinder [1]. The different strategies that have thus far been proposed to achieve EGR stratification mainly involve using the engine intake ports while experimental results with spark-ignition engines were achieved at low load or low speed. For this general approach, Choi et al. [12] mentioned the difficulty modifying intake ports to achieve EGR stratification. Other authors, such as Dong et al. [17] and Han & Cheng [14], have identified, based on CFD (computational fluid dynamics), that if a swirling flow is used to achieve stratification, then factors such as heat transfer between the phases, centrifugal forces and geometrical aspects will tend to decrease the stratification in the cylinder. One of the drawbacks of using the intake port to generate a stratified in-cylinder EGR is similar to the conventional EGR approach: the presence of EGR in the intake port decreases the amount of air induced inside the cylinder and thus limits the power of the engine. Hence, the benefits of EGR at high load is difficult to achieve. One way to circumvent this problem is to directly inject EGR in the cylinder without going through the intake port. By doing so, EGR can be added directly in the cylinder, independent of intake valve opening and closing timing. Thus, at WOT, maximum air and fuel can feed the cylinder and a stratified EGR can be created near or after the end of the intake stroke while in-cylinder pressure is still relatively low. Moreover, by using an electronically control valve, it is possible to time EGR addition with the ECU (electronic control unit) at will during the intake and compression stroke and thus offering new EGR strategy opportunities. This is the approach proposed in this work.

This paper aims to present a proof of concept for a novel direct injection stratified EGR system, which is experimentally tested to achieve EGR stratification at high load. In the approach, rather than modifying the intake port or valve timing during the intake stroke, EG is directly injected in the cylinder through the cylinder head, allowing the EGR injection to be timed during the intake and compression stroke. Experiments are conducted with a modified single-cylinder engine to demonstrate the system performance on engine combustion and pollutant emissions. The objective is thus to quantify the concept performance with respect to fuel consumption, emissions and combustion duration, as well as IMEP.

2. Experimental Set-up: Engine

The experimental set-up, illustrated in Fig. 1, was based on an air-cooled, 2 valves, single-cylinder engine having the specifications presented in Table 1. The engine was first modified with the addition of an in-house single-port fuel injection system (see Fig. 1) based on a Denso four-hole injector coupled with a low-pressure fuel pump, fuel filter, and pressure regulator that held the injection pressure at 3.1 bar. The injector was controlled by an Open ECU M670B. A single injection strategy with the injection starting at the beginning of the intake stroke was used, similarly to Gold et al. [18]. A wideband lambda sensor (Horiba Mexa 730) was located in the exhaust pipe, as shown in Fig. 1,

Table 1
Engine specifications.

Parameter	Description
Engine	Honda GX390
Displacement volume	389 cm^3
Stroke	88 mm
Bore	64 mm
Compression ratio	8
Engine speed	2100 and 2500 RPM

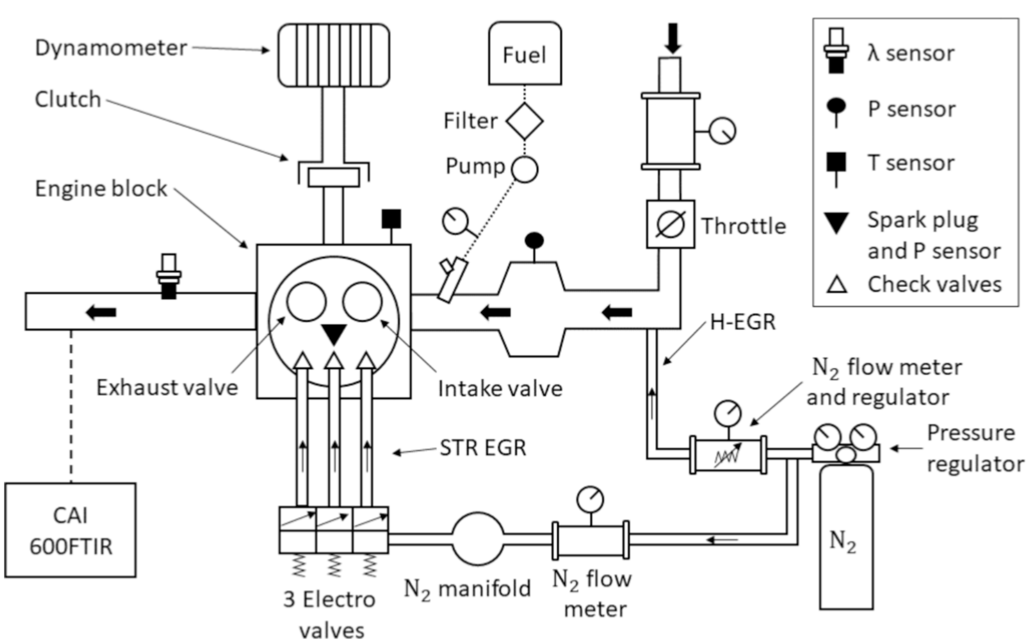

Fig. 1. Diagram of the experimental set-up illustrating both homogeneous and stratified EGR configurations.

and was utilized to maintain a stoichiometric mixture using a closed loop approach with the electronic control unit (ECU).

The engine was coupled to a dynamometer (Land & Sea eddy-current absorber) using a friction clutch. This configuration limited the range of the engine speed that could be covered during the experiment, and for that reason, only the minimum and maximum engine speed were used (see Table 1). During engine testing, the dynamometer controller allows to maintain a constant RPM while also measuring the engine torque. A pressure transducer (Kistler 6113b) embedded in a spark plug, was connected to a charge amplifier (Kistler model 5010A10) and allowed measuring the in-cylinder pressure (Fig. 1), which was pegged and set equal to the manifold pressure at the bottom dead center (BDC) at the end of the intake stroke, similarly to Ausserer et al. [19]. A 360-degree encoder ensured a proper timing between the in-cylinder pressure measurement and the piston position.

Finally, to allow the precise monitoring of the gasoline mass injected, experiments were conducted using the injection system alone in combination to a mass scale to monitor 5000 consecutive fuel injections. This approach was used for 5 injection durations and 4 different injection pressure differentials (injection pressure minus ambient pressure), ΔP, representative of the engine operating conditions. The measurements were then used to estimate the fuel consumption. The average mass injected as a function of the injection duration for 4 different ΔP is presented in Fig. 2, and shows the measurements to be highly repeatable as the uncertainty (methodology presents in section 2.3) is found to be 0.2 %.

2.1. Direct injection EGR system

With a conventional EGR system, the exhaust gas is routed from the exhaust system to the intake manifold using an electronic controlled valve. The EGR valve allows controlling the quantity of exhaust gas recirculated as a function of engine speed and load. This study proposes injecting the EG directly in the cylinder through the cylinder head. To this end, three solenoid valves (Festo MHA4, with a nominal flow rate of 6.67L/s and having a 3.5 ms opening and closing delay) supplied the cylinder head, as can be seen in Fig. 1, and were controlled by the ECU, allowing adjustments to the EGR injection timing and duration. Due to temperature constraints, the solenoid valves were connected, by pipes, to check valves mounted directly on the cylinder head. This configuration also prevents the high-pressure gas of the cylinder from returning into the EGR system. It should be noted that the check valve positions were constrained by the space available in the cylinder head, as well as by the ease of installation, and therefore, the studied design was not optimized in this first configuration. The design could therefore be considered as a first proof of concept, while the EGR inlet position and orientation in the cylinder head will be improved by ongoing CFD calculations. To simplify the set-up, high-pressure nitrogen (N_2) cylinders were used in lieu of EGR gas, consistent with another EGR study [14]. It is noted that using N_2 as an EGR gas substitute has been validated by others in the past as an appropriate diluent gas for such studies [20] in spark-ignition engines, and that therefore the conclusions reached here are expected to translate to the different exhaust gas. Moreover, Randolph et al. [20] found that at high loads, the influence of using N_2 as EGR was particularly negligible on calculated mass fraction burned (MFB) and knock performance when compared to traditional EGR. They also reported a slight increase of engine maximum efficiency (0.9 point higher) with N_2 while the combustion stability behavior followed a similar trend to conventional EGR but with a higher tolerance to dilution, by a few percent, when using N_2. Because high-pressure N_2 is used in this work, the experiments most directly relate to cold EGR given that the N_2 was at room temperature. The nitrogen pressure was set to 800 kPa for all experiments, and a mass flow meter (Brooks MF53S) was used to measure the amount of N_2 flowing into three solenoid valves during stratified EGR (see Fig. 1). To reach high EGR rates with the proof of concept, three valves were necessary due to their limited flow rate. Fig. 1 also shows that a mass flow meter controller (Azbil MQV0200), fed at 200 kPa, quantified and regulated the amount of N_2 flowing into the intake manifold when a homogeneous EGR was in use. Fig. 3 shows the modified cylinder head with the spark plug hole between the intake and exhaust ports on the right and left, respectively, and the three (3) EGR inlet ports.

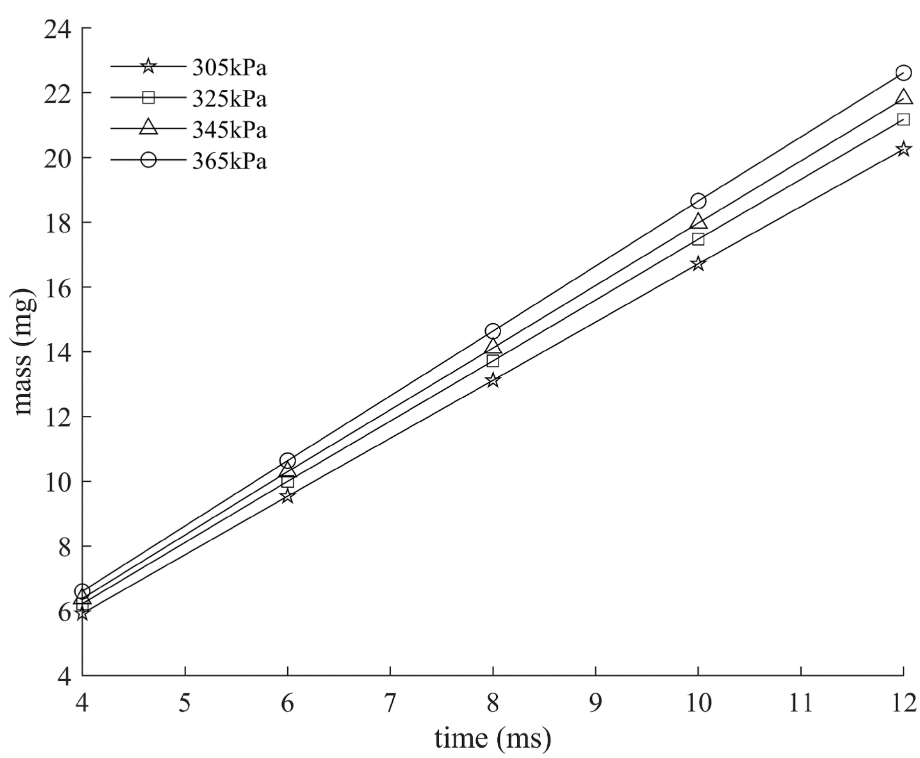

Fig. 2. Gasoline mass injected for different injection durations and pressure differentials.

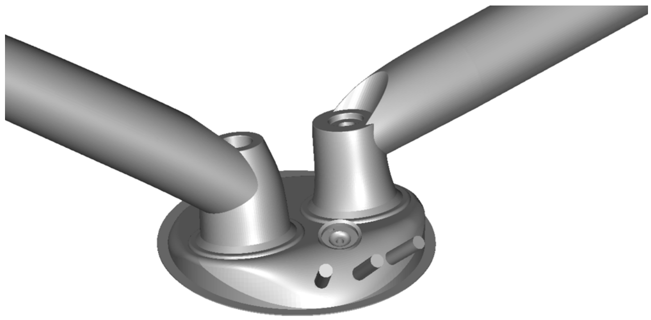

Fig. 3. Modified cylinder head with three direct injection EGR inlets located below the spark plug hole.

2.2. Engine test procedure

The experimental procedure was laid out as follows, ensuring consistency between each test. The engine was started and was given enough time to warm up based on a K-type thermocouple measurement located against the side of the engine block between two cooling fins. Once the engine temperature reached 413 K, the experiments began. It should be noted that the 413 K engine steady-state temperature varied by less than 20 K on a day-to-day basis. Once the steady-state temperature was reached, the engine was brought to its operating point with a brake mean effective pressure (BMEP) of 388 kPa at 2100 RPM or 2500 RPM. Under these conditions, the engine throttle is near its wide-open position, with an intake air pressure of 82 kPa measured in the absence of EGR, and values increasing to reach WOT under maximum homogeneous EGR. It should be noted that with the proposed stratified EGR, a lower intake air pressure was measured than with the homogeneous approach. In all cases, a stoichiometric gasoline-air mixture was maintained using the closed-loop control function of the OpenECU, while the spark timing did not vary significantly between the different configurations and unless otherwise stated, the spark timing was set to 37 degrees before the top dead center (TDC). The engine settings allow a stable operation such that a constant RPM was held within a maximum ± 18 RPM. At a given operating point, a data acquisition system recorded engine-related data such as the lambda sensor, temperature and intake manifold pressure over a period of 2 min at 100 Hz while RPM and torque were simultaneously acquired at 200 Hz with the dynamometer software. Dedicated software from the manufacturer was also used for the acquisition of the exhaust gas analyzer output and N_2 mass flow meters during that same period. Finally, a Labview acquisition system was activated to record 250 consecutive in-cylinder pressure signals synchronized with encoder pulses.

Four different EGR operation modes were tested using the following nomenclature. In a homogeneous EGR (H-EGR hereafter), N_2 is injected continuously in the intake manifold, similarly to the conventional approach, at a fixed mass flow rate. This configuration allows a comparison with the new proposed system which allows controlling the EGR addition independently to the intake valve opening and closing, which is not possible from other concepts. This is a great advantage as EGR addition during the intake stroke decreases the amount of air entering the cylinder which hinders the maximum power an engine can develop in the presence of dilution. To show this greater flexibility with the proposed proof of concept, three different timings are evaluated as depicted in Fig. 4 in relation to the intake and compression stroke as well as spark timing (S.T.). It is noted that no optimization of EGR injection timing was pursued as the objective is to show the possibilities of the proposed approach. Thus, one strategy sets the EGR injection at 35 degrees before bottom dead center (BDC), thus during the second half of the intake stroke. Another timing was chosen when the piston reached BDC and for which the intake valve is nearly close. Finally, one last timing is chosen at the end of the compression stroke, acknowledging that less EGR would flow due to the high in-cylinder pressure. However, it allows illustrating the capability of such an approach, which effectively modifies the cycle diagram potentially increasing the effective compression ratio, without incurring a higher risk of auto-ignition. Thus, the nomenclature for the three stratified EGR timings evaluated are as follows: STR215 as the injection begins 215° before TDC (thus, 35° before BDC intake stroke); STR180, with an injection beginning at 180° before TDC (thus, at BDC of the intake stroke); and STR50 which set the injection at 50° before TDC, so near the end of the compression stroke. For STR215 and STR180, two EGR levels were tested, while with STR50, a single EGR level was studied to explore a very late EGR injection strategy acknowledging that in-cylinder pressure becomes a constraint and limit the amount of EGR flowing due to the limitations of the solenoid valves and the EGR pressure itself. In all cases, the throttle was adjusted to maintain the engine torque constant at a given engine speed.

The EGR rate, in percent, was based on Eq. (1) [6], where the nitrogen mass (m_{N2}) was obtained from a N_2 flow meter, while the air mass (m_{air}) was computed using Eq. (2) based on the definition of λ [21] using the lambda sensor measurement (λ), using a stoichiometric A/F of 14.7, and the mass of gasoline injected (m_f). Herein, a low and a high EGR level implies an average EGR rate of 14 % and 20 % of N_2, respectively, while the exact EGR rate value is used when displaying the results.

$$EGRrate\% = \frac{m_{N2}}{m_{air} + m_{N2}} \times 100 \tag{1}$$

$$m_{air} = m_f \bullet \lambda \bullet 14.7 \tag{2}$$

For STR50, the injection occurred during the in-cylinder pressure rise due to compression. This prevented part of the EGR from getting inside the cylinder and to be trapped in the injection pipe located between the check valve and the solenoid valve. As such, the mass injected in the cylinder for the STR50 configuration was estimated using Eq. (3), which is based on the mass flow rate of a solenoid valve [22] following ISO6358-1989 standard. In that equation, C is the sonic conductance, ρ_{N2} is the nitrogen density at 293,15 K and 100 kPa, P_{N2} is the nitrogen pressure feeding the solenoid valves, P_{cyl} is the in-cylinder pressure and dt is the time interval by crank angle degree. With STR50, it was possible to reach an estimated EGR rate of 6 % while the calculated EGR rate value is used in the figures.

$$m_{N2} = C \bullet \rho_{N2} \bullet P_{N2} \bullet dt \;\; \text{when}\, P_{cyl}/P_{N2} < 0.528 \tag{3}$$

$$m_{N2} = C \bullet \rho_{N2} \bullet P_{N2} \left(\sqrt{1 - \left(\frac{\frac{P_{cyl(\theta)}}{P_{N2}} - 0.528}{0.472} \right)^2} \right) \bullet dt \;\; \text{when}\, P_{cyl}/P_{N2} \geq 0.528$$

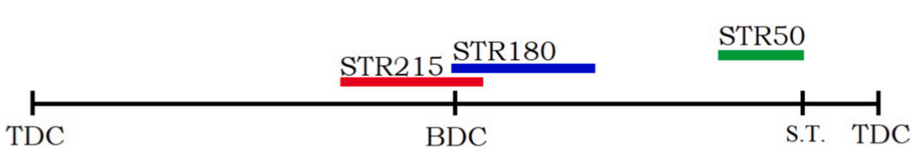

Fig. 4. Electrical commands of the stratified EGR strategies and spark timing (S.T.) in relation to engine strokes (not to scale).

2.3. Data post-treatment

The in-cylinder pressure measurements allowed quantifying the combustion process. Hence, the data was used to examine the repeatability of the combustion process as determined by the Coefficient of Variability of the Indicated Mean Effective Pressure (COV of IMEP). Eq. (5) [23] was used to this end, with σ being the standard deviation and $\overline{IMEP}$ being the average IMEP based on 250 consecutive cycles.

$$COV\ of\ IMEP = \frac{\sigma_{IMEP}}{\overline{IMEP}} \tag{5}$$

The in-cylinder pressure was also used to determine the mass fraction burned (MFB) through a net heat release analysis in order to quantify the early kernel growth expressed by 0–10 % of MFB (MFB0-10) and the fully developed turbulent combustion defined by the interval between 10 % and 90 % of MFB (MFB10-90). The net heat release was determined, neglecting heat loss to the walls, based on Eq. (6) [21], using the instantaneous in-cylinder pressure (P) and in-cylinder volume (V) and their respective variations, dP and dV. Finally, γ is the heat capacity ratio and was taken as equal to 1.3, based on the work of Yeliana et al. [24].

$$\frac{dQ_{net}}{d\theta} = \frac{\gamma}{\gamma-1}P\frac{dV}{d\theta} + \frac{1}{\gamma-1}V\frac{dP}{d\theta} \tag{6}$$

Pollutant species concentrations of NO, carbon monoxide (CO) and unburnt hydrocarbons (THC) were measured using Fast Transform InfraRed spectroscopy (California Analytical Instruments, model FTIR600) and its associated software. The accuracy of the analyzer was 2 % for NO, CO and THC, based on the manufacturer's manual. From the FTIR molar concentrations (x_i), the emission index (EI) and specific emissions (SE) were computed using Eq. (7) & (8) [21]. In these equations, M_i is the molar weight of the species i; α is the number of carbon atoms in a molecule of gasoline, taken herein as C_8H_{18}, and M_f is its molar weight, while $\dot{m}_f$ and $\dot{W}$ are the fuel mass flow rate and engine power, respectively.

$$EI_i \approx \frac{x_i \times M_i}{[(x_{CO2} + x_{CO} + x_{HC})/\alpha] \times M_f} \tag{7}$$

$$SE_i \approx \frac{\dot{m}_f \times EI_i}{\dot{W}} \tag{8}$$

Finally, the uncertainty analysis methodology of Telli et al. [25] has been used and is based on Type A (U_A) and Type B (U_B) uncertainty [26], expressed by Eq. (9) and (10) respectively [25]. Type A uncertainty is derived from measurements while Type B uncertainty is based on the manufacturer's specifications when provided. In Eq. (9), s is the global variance, N is the number of observations, $\overline{x}_N$ is the mean value of each test, while $\overline{x}$ is the mean of all tests. In Eq. (10), B is the manufacturer reported specification and k is normally taken as equal to 2 or 3 [26] depending of the level of confidence (95 % or 99 %) and assuming a normal distribution. Herein a value of 2 is used.

$$U_A^2 = s^2 = \frac{1}{N(N-1)}\sum_{N=1}^{i}(\overline{x}_N - \overline{x}) \tag{9}$$

$$U_B = \frac{B}{\sqrt{k}} \tag{10}$$

The combined uncertainty (U_C) is given by Eq. (11) [25], while for engine parameters that are calculated based on measurement, such as specific fuel consumption, the combined uncertainty is computed using the derivative method shown in Eq. (12). f is a function of at least two directly measured variables. Table 2 presents the calculated combined uncertainty of the results while Table 3 shows the instrument specifications.

Table 2
Combined uncertainty of the experiments.

Measurements	Uncertainty
THC emissions	0.8 g/kW-h
NO emissions	0.5 g/kW-h
CO emissions	2 g/kW-h
SFC	0.06 g/kW-h
0–90 MFB	0.6 deg
Stratified-EGR	0.17 %
H-EGR	1.15 %

Table 3
Instrument specifications.

Instrument	Uncertainty
Gas analyzer	2 %
Encoder	1 deg.
Lambda sensor	0.7 %
N_2 flow meter H-EGR	1 % of full scale
N_2 flow meter S-EGR	0.21 lpm

$$U_C^2 = U_A^2 + U_B^2 \tag{11}$$

$$U_C^2 = \sum_{j=1}^{k}\left(\frac{\partial f}{\partial x_j}\right)^2 U_C^2(x_j) \tag{12}$$

3. Results and discussion

This section contrasts the results of STR215 and STR180 with those of H-EGR, at similar EGR rates. Since STR50 had lower EGR rates, its in-cylinder pressure analysis is done separately.

3.1. Cycle-to-cycle variations

The trend of the cycle-to-cycle variations as a function of EGR rates was found to be impacted by the EGR timing. This section describes this effect.

3.1.1. Early EGR injection

The analysis first looks at the engine stability based on the COV of IMEP, which shows that the engine, while stable with respect to RPM, can be considered as having an inherently high COV of IMEP, in part due to the experimental set-up that required a friction clutch to interface the engine to the dyno. Fig. 5 shows the COV of IMEP for H-EGR, STR215 and STR180 as a function of the EGR rate, and for the two engine speeds studied herein. It is observed that in the absence of EGR, a COV of 8.8 % and 9.5 % is found at 2100 and 2500 RPM, respectively. For both engine speeds, the addition of a homogeneous EGR increased the COV of IMEP. It is noted that the experimental set-up can be considered as being equivalent to using a cold EGR, which inherently increase the COV of IMEP [1]. Moreover, the highest EGR rate tested with H-EGR can be considered to be close to the EGR limit of the engine. Fig. 5 also illustrates that a stratified EGR can render the engine less sensitive to EGR than H-EGR, as a decrease of COV of IMEP is observed, depending on the engine speed and amount of EG injected. For example, at a 14 % EGR level, STR180 resulted in a lower COV of IMEP than H-EGR at 2100 RPM, while an earlier injection (STR215) was less advantageous, although a retarded spark timing has been used to obtain the STR215 data shown in Fig. 5 (left). On the other hand, at 2500 RPM, STR215 offered lower values of COV of IMEP than H-EGR for both EGR rates. Finally, at 2500 RPM, STR180 offered a similar trend and values as H-EGR, with an increase of COV of IMEP with increasing EGR rates, even though at the highest EGR level, STR180 was less stable.

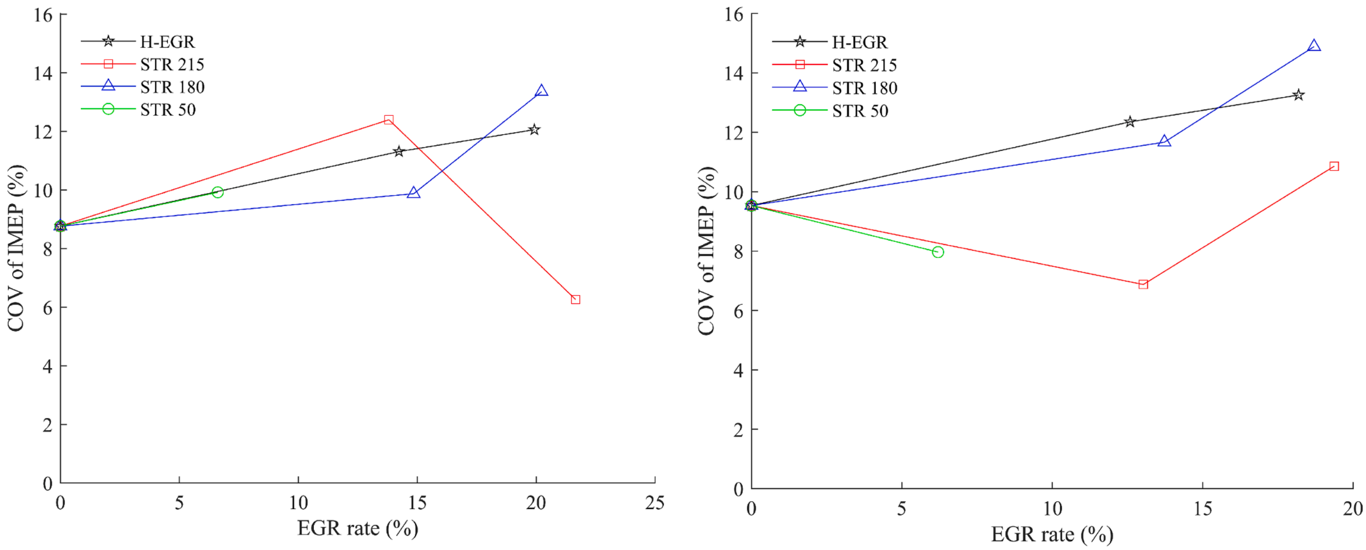

Fig. 5. Impact of EGR strategy on the COV of IMEP as a function of EGR rate at 2100 RPM (left) and 2500 RPM (right).

3.1.2. Late EGR injection

As stated previously, a test was conducted to see how a very late EGR injection would impact the engine performance, knowing that with an in-cylinder pressure higher than during the intake stroke, the amount of EGR injected would be smaller due to the maximum operating pressure of the nitrogen valves. Therefore, with STR50, N_2 addition was considered exploratory so as to see how such a late EG injection would impact the combustion process. With STR50, the end of the injection was reached when the in-cylinder pressure reached 8 bars (which is close to the spark timing). Using such a late injection leads the configuration of the experimental set-up to trap a residual amount of N_2 in the pipe located between the electronic valve and the cylinder head, where a check valve was added to prevent gas from returning from the cylinder. However, the configuration causes the residual nitrogen in the pipe to leak into the cylinder once the in-cylinder pressure fell below approximately 5 or 6 bar. From in-cylinder pressure measurements, this late N_2 addition was nearly synchronized with the exhaust valve opening. So, the main impact of such a late nitrogen addition would be to enhance the dilution. It was estimated that less than 5 % of the total in-cylinder mass was injected during that late period and thus making the total EGR level (including late dilution) similar to the lowest EGR cases studied herein.

Therefore, because of the lower EGR rate achieved with STR50, a comparison was made without EGR and with the 14 % H-EGR case. The results are shown in Table 4. It is observed that at 2500 RPM, STR50 offered a lower COV of IMEP than in the absence of EGR, suggesting that a late injection might increase turbulence, which benefits the combustion process, which will be verified in a later section, and lowers the COV of IMEP.

3.2. Mass fraction burned analysis

The effect of EGR timing was found to be significant on the combustion duration and is thus analyzed in what follows.

Table 4
Comparison of COV of IMEP as a function of engine speed.

Engine speed	EGR strategy	COV of IMEP
2100 RPM	0 % EGR	8.8 %
	14 % H-EGR	12.1 %
	STR50 with 6 %	9.9 %
2500 RPM	0 % EGR	9.5 %
	14 % H-EGR	13.3 %
	STR50 with 6 %	8.0 %

3.2.1. Early EGR injection

An MFB analysis is used to further quantify the impact of stratified EGR injection. The stratified EGR injector layout tested here was intended to bias them away from the spark plug, but at the same time, it also increased in-cylinder turbulence. This sped up the combustion process, which resulted in faster flame propagation. Fig. 6 shows the mass fraction burned duration (0-10MFB and 10-90MFB) for H-EGR, STR215 and STR180. As stated previously, STR215 at 2100 RPM required a retarded spark timing. Fig. 6 (left) shows the results at 2100 RPM, where the 0-10MFB is nearly constant irrespective of the EGR strategy employed. However, once the fully turbulent combustion phase (10-90MFB) is reached, the stratified EGR is always faster than H-EGR, suggesting that the increase in turbulence and/or the stratification of EGR within the cylinder may directly contribute to more rapid flame speeds, as expected.

The above conclusion is based on the following: 1) It is known that a homogeneous cooled EGR increases the 10-90MFB duration in spark-ignition engines [1,4]; 2) Data (not shown, for brevity) shows that STR180 and STR215 decreased 50-90MFB at both engine speeds when the highest EGR rate was used, but that the 10-50MFB remained relatively constant and similar to that of H-EGR. This result suggests an N_2 stratification located near the center of the cylinder (in concordance with the orientation of the N_2 direct injection inlet ports); 3) At 2100 RPM, STR215 and STR180 with the highest EGR levels offered shorter 50-90MFBs than in the absence of EGR, suggesting not only the stratification of nitrogen, but also a possible increase of turbulence (particularly with STR180) due to the direct N_2 injection. Ongoing CFD calculations shall help clarify the phenomena at play for the strategies studied herein in an upcoming paper.

The mass fraction burned results above provide some insight into the impacts of using a stratified EGR with the proposed approach. It is assumed that the injection of nitrogen directly in the cylinder through three different ports will increase the turbulence and thus foster the combustion process. Based on EGR mass flow rate measurements, the average velocity of the N_2 flowing in the cylinder was estimated considering the size area of the three ports. It is expected that the injected N_2 could reach an average velocity of around 50–55 m/s, depending on the EGR rate. Such a velocity can thus generate an increase of turbulence during the intake and compression strokes, resulting in a faster combustion process. To further quantify this effect, the maximum pressure rise rate during combustion $\left(\frac{dP}{d\theta}\right)_{max}$ was calculated. This effort was motivated by the results of Zhang et al. [27], who, when

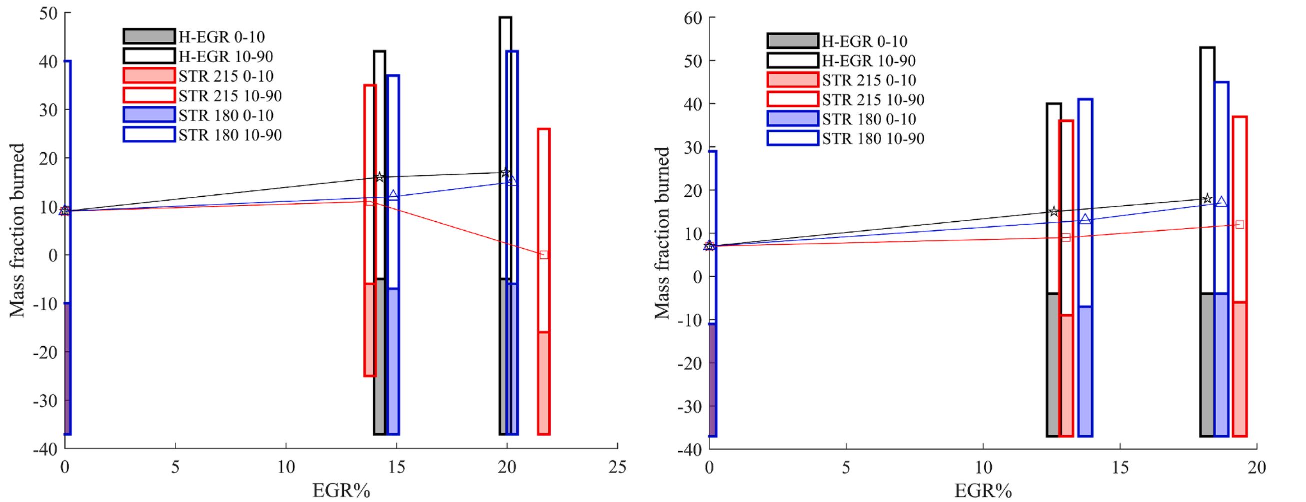

Fig. 6. Impact of EGR approaches on combustion duration at 2100 RPM (left) and 2500 RPM (right). Symbols identify the location of the 50 % MFB.

studying the effect of the tumble ratio in presence of homogeneous EGR, reported that turbulence increase resulted in higher $\left(\frac{dP}{d\theta}\right)_{max}$. Fig. 6 shows the maximum pressure rise rate in the cylinder for both engine speeds. It is observed that $\left(\frac{dP}{d\theta}\right)_{max}$ is generally greater with a stratified approach and that STR215 is always providing a higher value, for a given level of EGR, than for an H-EGR by as much as 200 % at 2100 RPM with 20 % EGR. It is noted that under stratified EGR, the manifold pressure had to be decreased, by slightly closing the throttle, to maintain the engine load constant. Overall, Fig. 7 supports the hypothesis of the turbulence increase induced by stratified EGR.

3.2.2. Late EGR injection

The late EGR injection strategy, STR50, is compared with the results obtained in the absence of EGR in Table 5. It can be observed that at 2100 RPM, STR50 decreased the early flame kernel development duration, but that this positive impact was not observed at 2500 RPM, when the 0-10MFB increased, probably due to the presence of EGR in the vicinity of the spark plug. However, at 2500 RPM, the benefit of using STR50 is seen on the 10-90MFB, which is faster than the case without EGR, suggesting that at higher engine speeds, the turbulence increase induced by the injection of EGR is felt later in the cycle and/or that EGR distribution is different than in the case at 2100 RPM. Therefore, by looking at the 50-90MFB, it appears that STR50 at 2500 RPM is mostly felt during the latter part of the combustion process, suggesting an EG stratification located in the center of the cylinder and/or a possible increase of turbulence at the periphery of the cylinder due to air entrainment following EG injection. Therefore, the impact of STR50 being felt late in the cycle, no impact on $\left(\frac{dP}{d\theta}\right)_{max}$ is observed when compared to results in the absence of EGR.

Table 5
Comparison of mass fraction burned duration between STR50 and without EGR.

Engine speed	EGR strategy	0-10MFB (deg.)	10-50MFB (deg.)	10-90MFB (deg.)	50-90MFB (deg.)
2100 RPM	0 %	26	18	40	22
	STR50 with 6 %	23	18	40	22
2500 RPM	0 %	27	19	50	31
	STR50 with 6 %	31	18	42	24

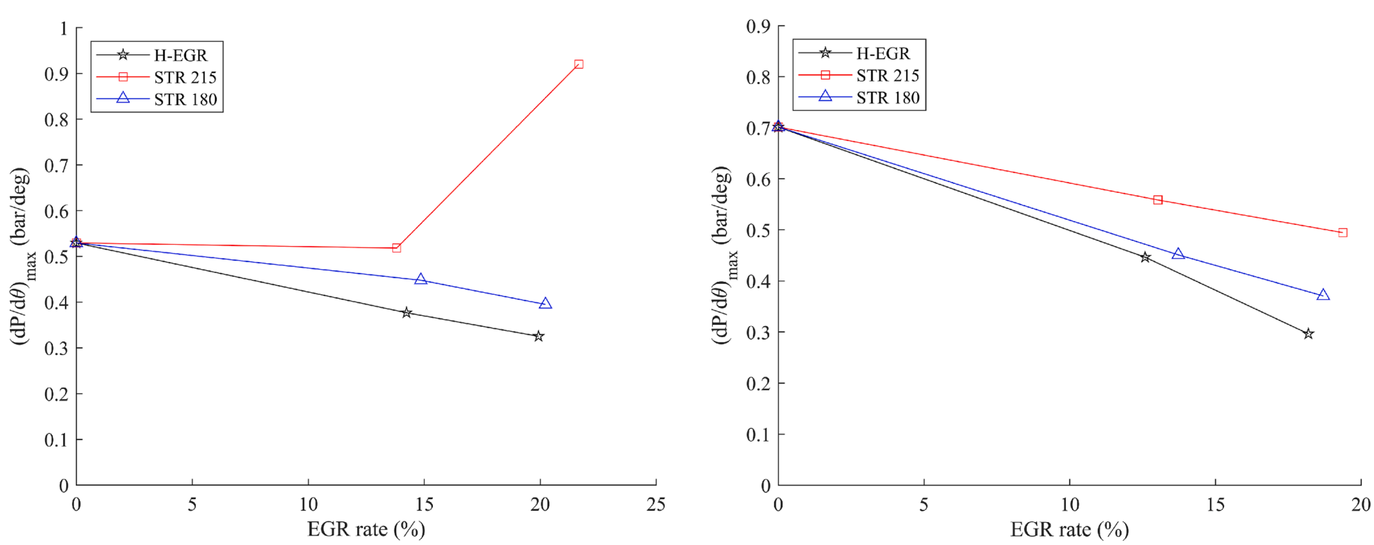

Fig. 7. Impact of EGR strategy on the in-cylinder maximum pressure rise $\left(\frac{dP}{d\theta}\right)_{max}$ at 2100 RPM (left) and 2500 RPM (right).

3.3. Specific fuel consumption

The impact of using a stratified EGR on the specific fuel consumption (SFC) is presented in Fig. 8, where the values are of the same orders as the ones reported by Zhang et al. [27] with a 4-cylinder engine at a similar load. The general behavior observed is a decrease of the SFC at the lowest EGR level, followed by its increase at the highest EGR level at 2100 RPM, while at 2500 RPM, the SFC is increasing with the EGR level. At the higher engine speed, as with the highest EGR level at 2100 RPM, the throttle is nearly fully open, and therefore, the addition of N_2 with a homogeneous EGR decreases the volumetric efficiency. When compared to H-EGR, the advantage of using a stratified EGR on the SFC is observed at both engine speeds, with the highest EGR rate offering a 12.5 % and 6.6 % SFC decrease at 2100 RPM and 2500 RPM, respectively, with STR215. Overall, it is observed at both engine speeds that retarding the stratified EGR injection timing slightly increases the SFC, but that on the whole, the values are lower or similar to the one obtained with H-EGR. This general trend is due, in part, to a slight decrease of volumetric efficiency experienced as the stratified EGR injection timing is retarded, because the throttle opening has to be decreased to lower the intake manifold pressure and thus keep the engine torque constant as compared to H-EGR. Finally, the results with STR50 shows a decrease of SFC at 2100 RPM but an increase at 2500 RPM. As noted in the previous section, at 2100 RPM, STR50 offered a slightly faster 0-10MFB than without EGR, which might explain the decrease of SFC in combination to the constant spark timing used here. On the other hand, at 2500 RPM the spark timing needed to be retarded to stabilize the engine and resulted in a slower initial combustion. This has the consequence of significantly retarding the location of 50 % MFB, which is detrimental to the SFC of STR50. This combustion location is an important parameter influencing SFC [28] in spark-ignition engines.

3.4. Pollutant emissions

Brake-specific nitrogen oxide (BSNO) emissions are presented in Fig. 9. The H-EGR strategy offered trends and values results similar to those of Zhang et al. [27], with a multiple-cylinder engine, and to Gu et al. [29], at a BMEP very close to the one used here. The results show that H-EGR offered a greater reduction of NO at both engine speeds when a 14 % EGR rate was used, with the exception of STR50 at 2100 RPM, which provided a similar BSNO, but with a lower EGR. At a higher EGR rate of 20 %, STR180 matched H-EGR at both engine speeds, while STR215 also did the same at 2500 RPM, albeit at a slightly higher EGR level. It is observed that a very late EGR strategy (STR50) at 2500 RPM does not offer any benefit with respect to NO, possibly because the added N_2 might be confined to a smaller zone in the spark plug vicinity (based on MFB above) than at 2100 RPM, due to a shorter duration between the N_2 injection and spark timing. This configuration thus has no significant impact on the combustion temperature as the combustion process here was faster than in all the different EGR configurations, and the different MFBs duration were close to the reference point without EGR.

At 2500 RPM with STR50, in combination with the highest EGR rate, it was observed that the strategy did not offer a decrease in NO, probably due to a lack of mixing between the added nitrogen and the fuel–air mixture. This hypothesis is supported by the THC emissions reported in Fig. 10, where the STR50 configuration offered the lowest amount of THC at 2500 RPM in concordance with the results of Fig. 8 when considering the trends reported in [28] with a spark-ignition engine. It is assumed there was insufficient time to mix N_2 to the air–fuel mixture, the late N_2 addition might have increased the in-cylinder turbulence, a fact that is known to be beneficial for homogenizing the air–fuel mixture [30], resulting in a decrease of THC. A similar observation was made for STR215 at 2100 RPM, where higher NO levels are associated with the lowest THC, and this EGR strategy also has the fastest MFB10-90 at that engine speed. The general increase of THC observed at 2100 RPM or with the higher EGR level at 2500 RPM is in concordance with the general effect of using cold EGR reported by others [1,31].

Finally, Fig. 11 indicates that carbon monoxide (CO) emissions are similar at 2100 RPM, irrespective of the EGR strategy employed, while it should be noted that the homogenous strategy offered results similar to Gu et al. [29] in values. As CO is mainly dependent on the air–fuel ratio, the results suggest that the air–fuel mixture did not change significantly at 2100 RPM. On the contrary, at 2500 RPM, an increase of CO is observed with stratified EGR strategies, which could be due to the EGR/mixture interface, and thus the stratification of the EGR, which impacts the air–fuel mixture locally. Ongoing CFD results might help explain the observed trends.

4. Discussion

As noted previously, all stratified EGR systems proposed prior to this study involved modification to the intake manifold by using independent runner for each intake valve [16] or consisted in feeding EGR in the intake runner at different moments [15,32] during the intake stroke as to achieve an EGR gradient. Moreover, tests were conducted at low load [16] or at a very low engine speed (800 RPM) [15,32]. By using the intake manifold to generate EGR stratification still limits engine power as less air enters the cylinder. The direct EGR injection technique proposed herein circumvents this problem by adding EGR through the cylinder head.

The proof of concept herein has been evaluated at engine speeds

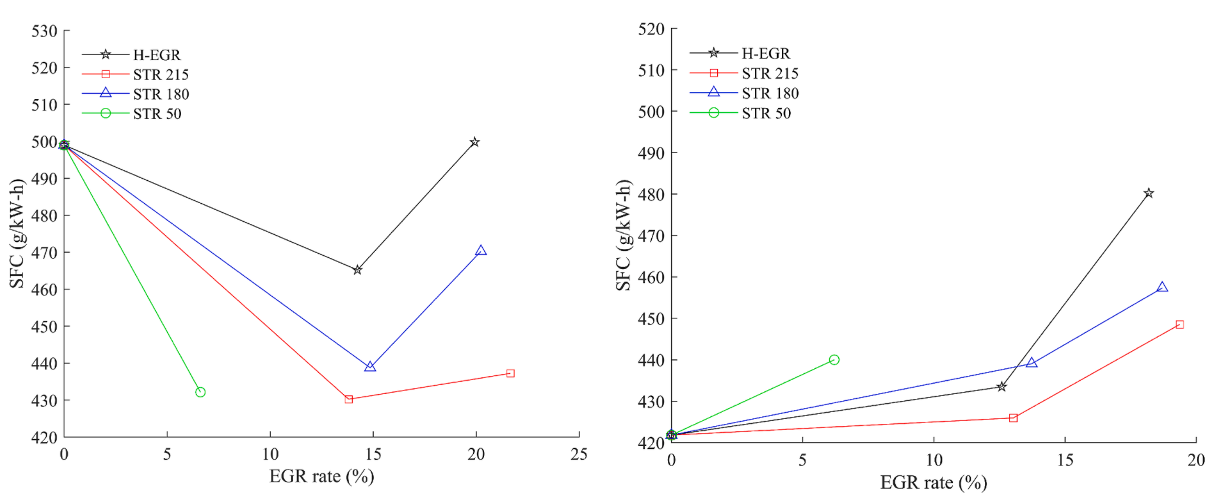

Fig. 8. Influence of EGR strategy on the specific fuel consumption. Left: at 2100 RPM. Right: at 2500 RPM.

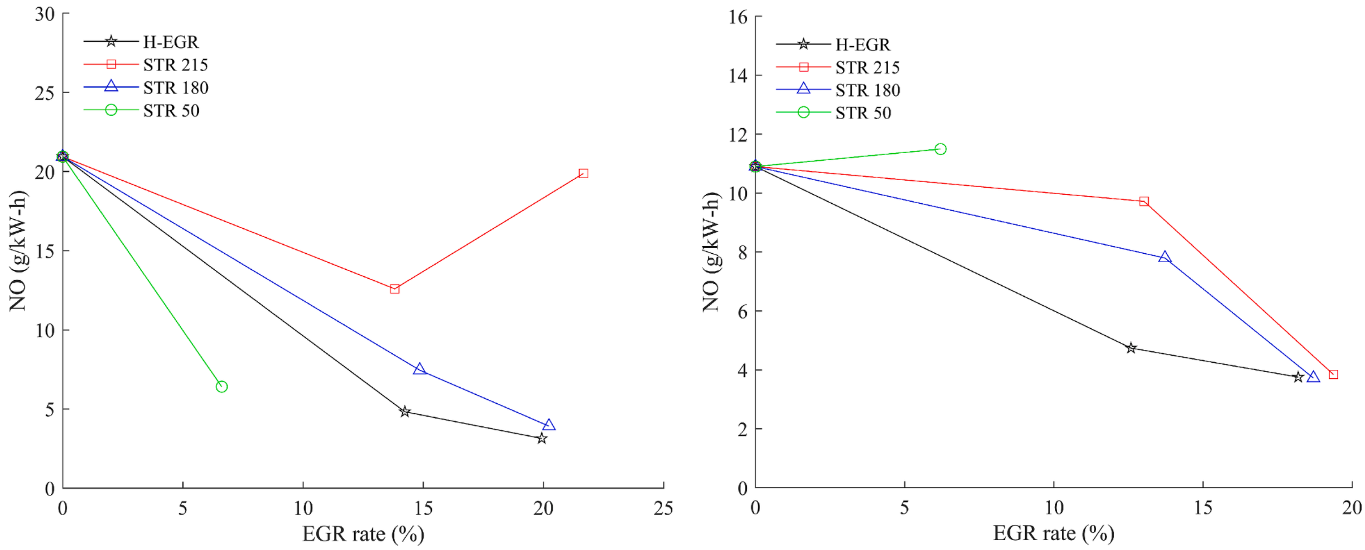

Fig. 9. Impact of EGR strategy on NO emissions. Left: at 2100 RPM. Right: at 2500 RPM.

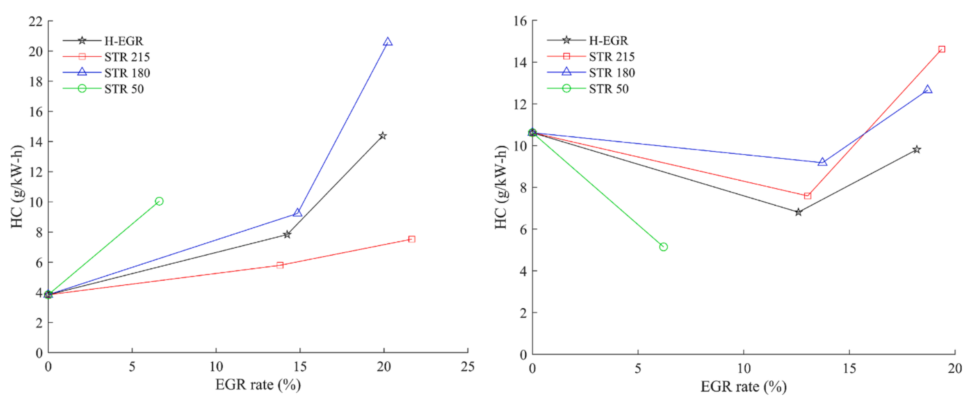

Fig. 10. Impact of EGR strategy on THC emissions. Left: at 2100 RPM. Right: at 2500 RPM.

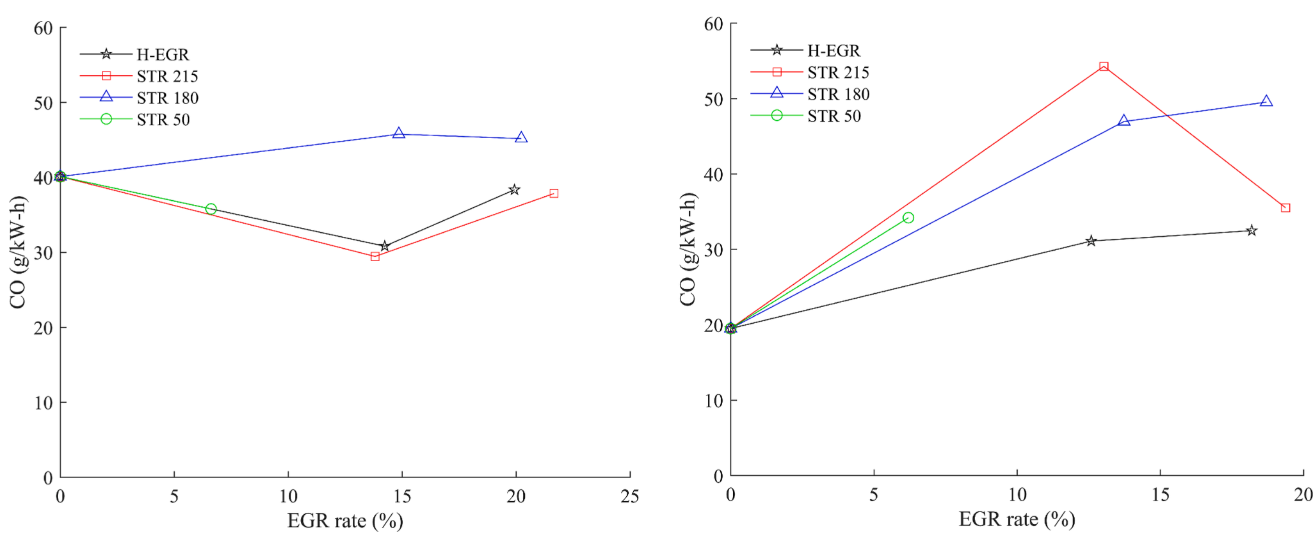

Fig. 11. Impact of EGR strategy on the CO emissions. Left: at 2100 RPM. Right: at 2500 RPM.

above the other concepts and with loads approaching WOT. The main results at high EGR rate are shown in Table 6 because the impact is more significant and that high dilution is also the aim of every stratified EGR concept. Table 6 quantify the impact of the EGR injection timing with respect to the homogeneous EGR at both engine speeds. The results show that combustion duration and SFC are decreased in presence of stratified EGR when compared to homogeneous EGR. It is noted that STR215 offered decrease of 12,5% and 6.5 % in SFC at 2100 RPM and 2500 RPM

Table 6
Impact of the strategy on main engine performance with respect to homogeneous EGR at 20% EGR rate.

Strategy	RPM	Combustion duration (0-90MFB)	SFC	NO emissions	THC emissions
STR215	2100	−26.7 %	−12.5 %	+1370 %	−48 %
	2500	−17.8 %	−6.5 %	0 %	+49 %
STR180	2100	−8.1 %	−6%	−2.6 %	+43 %
	2500	−8.9 %	−4.7 %	+2.7 %	+2.7 %

respectively, at a high EGR rate. STR180 allowed reaching NO emission nearly as low as H-EGR at 2100 RPM. By considering SFC and NO emission simultaneously, STR215 offers the best performance at 2500 RPM (−6.5 % in SFC and no change in NO) while STR180 performs better at 2100 RPM with a decrease of 6 % and 2.6 % in SFC and NO emission, respectively. The results confirm that EGR timing would be a function of engine speed.

Finally, comparisons with the few stratified EGR concepts that have been proposed with a spark-ignition engine is pursued, even if engine speed and load are lower than the tests herein. For example, decrease of SFC is observed at both RPM herein, while decreasing and increasing fuel consumption has been reported with stratified EGR [15] and [16] respectively. Concerning NO emissions, the higher NO observed above with respect to the homogeneous EGR are in concordance with results reported by Sarikoc et al. [16] with some of their EGR-stratification strategies while Fig. 9 showed that with a proper injection strategy NO emission could be as low as H-EGR at high EGR rate. Finally, with respect to THC, Sarikoc et al. [16] observed increase of that pollutant when compared to H-EGR which is similar to the behavior observed with STR215 and STR180 at 2500 RPM. However, STR215 allowed to decrease that pollutant at 2100 RPM when compared to H-EGR. From this comparison, the proof of concept showed that flexibility by EGR injection timing might be key to optimize engine performance under stratified EGR at high engine load.

5. Conclusion

This work presented a stratified EGR proof of concept based on the direct injection of EGR through the cylinder head. To that end, the cylinder head of a single-cylinder engine was modified to include 3 solenoid valves activated by the engine ECU so as to enable control on the EGR injection timing and duration. The solenoid valves were fed by pressurized nitrogen. The study objective was to quantify the performance of direct injection EGR at high engine load when compared to homogeneous EGR. Thus, 3 different stratified injection timings were chosen to minimize the impact of EGR addition on air mass flow entering the cylinder. The investigated EGR injection timing took place 1) near the end of the intake stroke; 2) early in the compression stroke and; 3) in the second half of the compression stroke. Experiments were conducted with two EGR rates, at two engine speeds at high loads (near wide-open throttle). The main findings are summarized as follows:

- **Combustion duration:** Using direct injection EGR decreases combustion duration significantly (between 8.9 % to 26.7 %) when compared to a traditional EGR. This trend suggests that an increase of turbulence might also come with the EGR injection.
- **Fuel economy:** At both engine speeds, decrease of SFC between 4.7 % to 12.5 % has been observed with respect to homogeneous EGR;
- **Pollutant emissions**: At 20 % EGR rate, it was possible with each strategy to reach NO emission level comparable to homogeneous EGR;
- **EGR injection strategy**: The results show that it is possible to decrease simultaneously SFC and NO using a stratified EGR strategy. At high EGR rate, STR215 offered the best performance at 2500 RPM while at 2100 RPM, STR180 performed best.

The results of the first direct injection EGR system shows that it is possible to achieve stratified EGR at near WOT and that this approach is not detrimental to engine power. Finally, EGR injection strategy is a function of EGR rate and engine speed.

Declaration of competing interest

The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: Patrice Seers and Patrick Lemieux has patent pending to Ecole de technologie supérieure. If there are other authors, they declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Data availability

Data will be made available on request.

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