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E25 stratified torch ignition engine emissions and combustion analysis

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ABSTRACT

Vehicular emissions significantly increase atmospheric air pollution and greenhouse gases (GHG). This fact associated with fast global vehicle fleet growth calls for prompt scientific community technological solutions in order to promote a significant reduction in vehicle fuel consumption and emissions, especially of fossil fuels to comply with future legislation. To meet this goal, a prototype stratified torch ignition (STI) engine was built from a commercial existing baseline engine. In this system, combustion starts in a pre-combustion chamber, where the pressure increase pushes the combustion jet flames through calibrated nozzles to be precisely targeted into the main chamber. These combustion jet flames are endowed with high thermal and kinetic energy, being able to generate a stable lean combustion process. The high kinetic and thermal energy of the combustion jet flame results from the load stratification. This is carried out through direct fuel injection in the pre-combustion chamber by means of a prototype gasoline direct injector (GDI) developed for a very low fuel flow rate. In this work the engine out-emissions of CO, NOx, HC and CO_2 of the STI engine are presented and a detailed analysis supported by the combustion parameters is conducted. The results obtained in this work show a significant decrease in the specific emissions of CO, NOx and CO_2 of the STI engine in comparison with the baseline engine. On the other hand, HC specific emission increased due to wall wetting from the fuel hitting in the pre-combustion chamber wall.

1. Introduction

Currently, more than 1 billion vehicles operate throughout the world; this estimate is expected to increase to 1.3 billion vehicles in 2030 and will exceed 2 billion vehicles by mid-2050 [1–3]. These vehicles generate approximately 70% of global carbon monoxide (CO) and 19% of carbon dioxide (CO_2) emission [2,4]. The increasing number of vehicles around the world increases the demand for fuel and affects the global climate, the stability of eco-systems, the health of the population, as well as the global economy [5–7]. The growing issues of climate environment changes and energy are driving the scientific community to further development of the engine technology road map.

Aiming to reduce specific fuel consumption and emission levels without losing performance, the scientific community has been adopting several different technologies and engine control strategies. At the present time, downsized engines associated with turbocharging, exhaust gas recirculation (EGR), gasoline direct injection (GDI) and the use of bio-fuels, both pure and blended in fossil fuels, are considered to be the most effective and inexpensive ways of reducing fuel consumption and emissions in the years to come. Homogeneous charge compression ignition (HCCI) technology has the potential to meet low NOx levels and soot formation, while still achieving high thermal efficiency. Apart from these favorable attributes of HCCI combustion, the complexity of the control of auto-ignition is still a major obstacle for the implementation of HCCI technology in practical engines, which depends mostly on the heat release from chemical kinetics [8–10].

The reduction in the engine displacement, the so-called "downsizing" combined with the boosting system, makes possible the recovery of power output and it has been extensively used in current engine development. However, it is important to point out that the reduction in engine displacement can be carried out up to a certain level, which is established by the combustion efficiency losses due to the limits imposed by fuel properties such as knock occurrence. Knock-limited combustion is a major challenge for highly-boosted downsized engines [11]. The reduction of the geometric compression ratio by at least 1.5 as compared to naturally aspired engines has been the most typical method of suppressing knock in boosted engines, which produces a penalty in efficiency at part load operation [12]. At the present time, cooled exhaust

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Nomenclature

GHG	greenhouse gases
STI	stratified torch ignition
GDI	gasoline direct injection
CO	carbon monoxide
NO_X	nitrogen oxide
HC	hydrocarbon component
CO_2	carbon dioxide
O_2	oxygen gas
HCCI	homogeneous charge compression ignition
CEGR	cooled exhaust gas recirculation
EGR	exhaust gas recirculation
SI	spark ignition
PFI	port fuel injection
TWCC	three-way catalytic converter
VCR	volumetric compression ratio
BSFC	brake specific fuel consumption
ISFC	indicated specific fuel consumption
K	polytrophic coefficient
CVCC	compound vortex controlled combustion
TJI	turbulent jet ignition
DOHC	double overhead command
VVT	variable valve timing
B-TDC	before top dead center
A-BDC	after bottom dead center
B-BDC	before bottom dead center
A-TDC	after top dead center
DI	direct injection
MCO	CO specific emission
$\dot{m}_{air}$	air mass flow
$\dot{m}_{fuel}$	fuel mass flow
BSXX	brake specific XX (specie) emission
MMXX	XX (specie) molar mass
MMex	exhaust molar mass
[XX]	XX specie concentration
EMS	engine management system
λ	lambda factor (inverse of equivalence ratio)
BL	baseline
S	stratified
BMEP	brake mean effective pressure
IMEP	indicated mean effective pressure
MBF	mass burnt fraction
T	in-cylinder temperature
t	time reference
θ	crank angle reference
m	in-cylinder mass
R	in-cylinder gas constant
V	combustion chamber current volume
P	in-cylinder pressure
HRR	heat release rate
SOC	start of combustion
EOC	end of combustion
Q_a	apparent heat release
η_{comb}	combustion efficiency
LHV	low heat value
$m_{fuel/cycle}$	fuel mass per cycle
H_R	enthalpy of reactants
H_P	enthalpy of products
n_i	molar fraction of specie i
Δh_f	specific energy of formation
$x_{[X]}$	mass fraction of specie X
$h_{[X]}$	specific enthalpy of specie X
T_o	reference temperature
ABNT	associação brasileira de normas técnicas
INMETRO	instituto nacional de metrologia, qualidade e tecnologia
f	function of calculated quantity
U_e	expanded uncertainty
U_m	measured uncertainty
k	confidence factor (*t-student*)
σ	standard deviation
n	number of samples
$A_{Q_{m_i}}$	measured quantity accuracy
Q_c	calculated quantity
Q_m	measured quantity
HLCS	heat loss to cooling system

gas recirculation (CEGR) seems to be the most effective way of suppressing knock in boosted engines at high loads and simultaneously improve fuel efficiency [11,13–16].

EGR was first adopted in Diesel engines. Currently, due to stringent emission regulations, it has also been used in spark ignition (SI) engines [17]. According to Fontana et al., EGR is primarily used in SI engines to reduce throttling losses at part load operation, thus reducing fuel consumption, and secondly, to reduce NOx emission levels [18]. Applying EGR in SI engines demands further opening of the engine throttle in order to maintain the same mean effective pressure. This reduces pumping losses, and consequently fuel consumption, compared to the condition where no EGR is used. Nevertheless, it is important to highlight that port fuel injection (PFI) in spark ignited engines works at stoichiometric air–fuel ratio in order to meet the emission requirements by means of a three-way catalytic converter (TWCC). When a lean mixture is used, the O_2 concentration increases in the exhaust, which will cause a corresponding higher NOx emissions due to the bad efficiency of the TWCC. Since current research GDI engines operate with lean or even ultra-lean mixtures, an in-cylinder technique must be used to control NOx emissions, since TWCC cannot be used for this purpose under lean mixture operation. EGR is an effective measure and nowadays represents the principal method to reduce NOx emissions of GDI engine [17].

The amount of CO_2 released due to the combustion of a bio-fuel such as ethanol is compensated by the regrowth of the plants used, which will absorb a similar amount of CO_2 during photosynthesis. For this reason, ethanol can be considered CO_2 free, this being the major reason for advocating its use [19,20]. The higher octane number, latent heat of vaporization and also faster laminar speed in comparison to gasoline makes ethanol a suitable fuel to be used in boosted engines [20]. A SI engine fueled with ethanol is less prone to knock and therefore allows the use of higher volumetric compression ratio (VCR), which is one of the key factors in efficiency improvement [20–22]. The use of an oxygenated fuel leads to a more complete combustion process, favoring a significant reduction of total hydrocarbon (HC) and carbon monoxide (CO). In addition, as the oxygen molecule reduces the locally fuel-rich region, favoring, in this way, the decrease of soot formation in direct injection engines [23–33]. On the other hand, the lower energy content of ethanol in comparison with gasoline represents a significant drawback, increasing brake specific fuel consumption (BSFC) and CO_2 specific emission. Currently, because of economic reasons, the use of pure ethanol is not cost-effective in Brazil, making its application limited. Because of the environmental advantages in regard to CO_2 emission achieved by the use of ethanol, it has been widely blended with regular commercial gasoline [34,35].