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Effects of engine operating conditions on particle emissions of lean-burn gasoline direct-injection engine

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ABSTRACT

Direct injection of fuel into the cylinder of an engine leads to the problem of particulate matter (PM) emissions. Lean-burn gasoline direct-injection (GDI) engines are known to emit higher levels of ultrafine particles than do conventional engines. The level of PM emissions by lean-burn GDI engines is unlikely to meet the EURO-VI emissions standards. In this study, the effects of combustion strategy and excess air ratio on the PM concentrations and particle size distribution were evaluated for a naturally aspirated lean-burn GDI engine. The engine operating conditions—including the fuel-air mixture and load—were varied in order to analyze the PM formation and the particle size distribution. The PM concentration was found to increase dramatically at an excess air ratio of 1.5, at which ratio lean combustion with a stratified mixture occurred. This was regarded as being the transition region between the premixed flames and the stratified mixture flames. Further, an increase in the excess air ratio to beyond 2.0 caused the PM concentration and particle number to increase again, possibly as a result of the relatively high ambient pressure and lower combustion temperature.

1. Introduction

Since 2012, the European Union has introduced regulations for limiting the levels of CO_2 emitted from vehicles and has proposed that technology for reducing fuel consumption by over 20% so as to reduce CO_2 emission levels to 130 g/km be developed by 2015 [1]. For this reason, the size of the market for hybrid and diesel vehicles is expected to increase fairly rapidly. Meanwhile, the development of gasoline engine technology that is capable of coping with the enforced emission regulations via enhancement of fuel economy to the level of diesel engines is being given serious consideration because manufacturers are faced with the problem of a significant shrinkage of the gasoline vehicle market. Examples of technologies for improving fuel economy in a commercial gasoline engine include downsizing while improving performance, engine lubrication, variable valve timing, and variable valve lift. Successful development of lean-burn gasoline direct-injection (GDI) engine technology is expected to enable a 15% reduction in fuel consumption and CO_2 emissions in comparison to those for conventional gasoline engines with port fuel injection (PFI) systems [2–6].

Low fuel consumption with high power output can be simultaneously achieved by means of the GDI engine technology, since a very lean fuel-air mixture is possible with highly precise controlled fuel injection directly into the cylinder, which leads to highly efficient combustion. The significant reduction in fuel consumption is achieved as a result of the smaller pumping loss during the intake stroke; further, when the engine is operating under low loads, the cooling loss is small because of the lower combustion temperature. Meanwhile, when the engine is operating under high loads, the volumetric efficiency improves by the latent heat of the fuel and the possibility of knocking reduces.

However, direct fuel injection into the cylinder of a GDI engine leads to the formation of particulate matter (PM) in the same way as that in a diesel engine. In particular, it has been reported that the particle number (PN) increases significantly in the case of direct fuel injection into the cylinder of a GDI engine [7,8]. Regulations limiting the PN were planned to be introduced with the implementation of the EURO-VI standard in 2014, possibly with a grace period of 1–2 years. The maximum allowable PN value under these regulations would be 6.0×10^{11} #/km, which is identical to the limit imposed on diesel engines [9]. Therefore, the use of a gasoline particulate filter will most likely be essential in order to meet the PN standard for lean-burn GDI engines.

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Several studies have been conducted for investigating the PM produced by a GDI engine, and some have reported that the degree of fuel-air mixing affects the size of the particles emitted by the engine [8,10–13]. The reason for this influence is that particles are likely to be formed in the fuel-rich mixture region as a result of the poor spray evaporation and wall wetting during the catalyst heating phase. Strategies for reducing the PM emissions, such as retarding the injection or ignition timing, splitting the control of the injection factor, and changing the spray pattern of the injected fuel, have been proposed because it is well known that PM emissions are closely related to the engine operating parameters [10,14,15]. Some previous studies have shown via transmission electron microscopy that mixture preparation plays an important role in the particle size and morphology. However, all of these studies focused only on stoichiometric mixture combustion and not on lean-burn GDI combustion [10,11,16].

Unlike in a PFI or stoichiometric GDI engine, in a lean-burn GDI engine, a diffusion flame is formed by the combustion of the stratified mixture when the combustion reaction occurs. Therefore, it is necessary to investigate the details of the mechanism that leads to the formation of PM in a lean-burn GDI engine. In the present work, the characteristics of PM emissions are studied from the viewpoints of the PM concentration and particle size distribution. The engine load is varied in order to evaluate the effects of the combustion strategy on the particle size distribution in a naturally aspirated lean-burn GDI engine. In addition to this lean operation, the results for stoichiometric operating conditions are compared with those for the lean mixture conditions to gain a deeper insight into the effects of the excess air ratio on the particle size distribution.

2. Experimental procedures

2.1. Setup

The test engine selected for this study was a 3-L six-cylinder naturally aspirated GDI engine designed for use in a passenger vehicle. The test engine was equipped with a spray-guided direct-injection system that is capable of realizing lean combustion through the formation of an ignitable stratified lean mixture in the vicinity of the spark plug near the centrally mounted injector. The outward-opening piezo injectors are installed in the engine head; it has been reported that their design is capable of coping with changes in the spray shape and injection rate caused by coking. A stable, hollow, cone-shaped spray formed by the introduction of fuel through the injector results in relatively smaller penetration than that in the case of a normal multi-hole injector and also causes the formation of a recirculation zone owing to presence of vortices near the electrode of the spark plug [17,18].

Table 1 lists the key specifications of the test engine, and Fig. 1 shows a schematic of the experimental setup of the engine. The engine speed and load were controlled with an eddy current (EC) dynamometer (ESF-H300, Fuchino Manufacturing Co., Ltd.). The gasoline was pressurized by a high-pressure plunger pump with a fuel control valve, before being fed to the fuel rail, where the fuel pressure was maintained at 20 MPa. The fuel injection, spark advance timing, and amount of fuel injected were controlled using a universal engine control unit.

Table 1
Specifications of test engine.

Parameter	Specification
Bore (mm)	85
Stroke (mm)	88
Compression ratio	12.0
Displacement (cc)	2996
Max. power	200 kW/6700 rpm
Max. torque	320 Nm/2750 rpm

The flow rate of the gasoline was measured using a coriolis flow meter (CMF010M, Micro Motion, Inc.). Engine control variables such as the engine speed and the intake throttle position were monitored, and the excess air ratio for each cylinder was measured using six wideband lambda meters (LA4 and LSU 4.2, ETAS Co.). A gas sampling probe was installed in the exhaust pipe and connected to an exhaust gas analyzer (MEXA 7100DEGR, Horiba). A data acquisition system (GL820, GRAPHTEC Co.) was used to enable acquisition of numerous channels of data; this system also provides real-time measurement displays. A scanning mobility particle sizer, consisting of a condensation particle counter (5.404, GRIMM) and a dynamic mobility analyzer, was used to measure the PN and particle size distribution of the PM emissions. A PM measurement device (DustTrak 8533, TSI) was used to simultaneously measure the PM concentrations with a time resolution of 1 s. This device has a flow rate of 1.7 L/min, and it measures the mass concentrations of the total suspended particles (PM_{total}), PM_{10}, $PM_{2.5}$, and PM_1 within a range of 0.001–150 mg/m^3.

A piezoelectric pressure transducer (6052A, Kistler) was used to measure the in-cylinder pressure. The indicated mean effective pressure (IMEP), coefficient of variation of IMEP (COV_{IMEP}), and heat release rate were calculated with a combustion analyzer (DEWE800CA2, Dewetron) by using ensemble-averaged cylinder pressure data acquired over 200 engine cycles.

2.2. Experimental methods

The test conditions were set to 2000 rpm and the brake mean effective pressure (BMEP) was set to 0.2 and 0.4 MPa in order to reflect the typical operating conditions and to ensure two representative operating points for the evaluation of the specific fuel consumption in a commercial gasoline engine of a passenger vehicle. Under the selected test conditions and at a certain engine speed, the PM emissions at various loads were compared. The data obtained from the experiments under the selected test conditions are sufficiently meaningful and worthwhile because the maximum excess air ratios are generally determined from the loads as being the values at the wide-open throttle condition, irrespective of the engine speed. After the engine had warmed up sufficiently, the cooling water control system was used to maintain the cooling water temperature at 82.5 ± 2.5 °C. The operating conditions are summarized in Table 2. The excess air ratio was controlled by manipulation of the throttle valve of the engine while controlling the amount of air, and the ratio was varied from stoichiometric to the lean limit. The throttle valve was manipulated to control the excess air ratio in increments of 0.5 and 0.2 for the BMEP 0.2 and 0.4 MPa load conditions, respectively. However, the lean limit was determined as being achieved at the wide-open throttle condition and the excess air ratio at the lean limit could not did not coincide with one of the values attained using the intended increments. In the present study, the lean limits corresponded to excess air ratios of 3.6 and 2.1 for the BMEP 0.2 and 0.4 MPa load conditions, respectively, and the results at these lean limits were analyzed and compared. To realize stable combustion with an overall ultra-lean mixture, a late injection strategy was employed wherein the injection timing was extremely close to the top dead center (TDC). Two-stage injection was employed at the operating point of BMEP 0.4 MPa (hereafter simply "BMEP 0.4 MPa operating point"), stable combustion could not be realized even with a late injection strategy. When the two-stage injection was employed at the BMEP 0.4 MPa operating point, the ratio of fuel injection for the first