Fuel Economy and Emissions from a Two-Valve Direct-Injection S.I. Engine Operating in the Stratified-Combustion Regime

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ABSTRACT

This study is an experimental and computational investigation of the influence of in-cylinder air motion on the combustion, fuel economy, and engine-out emissions of a single-cylinder, 2-valve, spark-ignition direct-injection (SIDI) engine, operating under stratified-charged conditions. The engine required relatively retarded injection timings (in comparison to other charge- or wall-controlled DI engines), high swirl levels, and a spray orientation that is directed towards the intake-valve side and targets the ridge wall of the piston. The stratified combustion was thus found to be a combination of wall and charge control, as indicated by the indirect spray targeting relative to the spark location coupled with the high swirl requirements of the engine, as well as the crucial role of the piston ridge in deflecting the fuel vapor towards the spark gap. The CFD computations, which showed close agreement with the experimental results, suggest that minimizing the amount of fuel deposited on the piston surface is a requirement for low exhaust smoke. It was also concluded from the combined results of the computations and the experiments that the time-averaged air-fuel mixture in the immediate vicinity of the spark gap at the time of ignition should be biased towards a rich mixture to minimize misfires and cyclic combustion variability.

INTRODUCTION

Significant advancements have been made in recent years in the development of combustion systems for spark-ignition direct-injection (SIDI) engines, which have resulted in large fuel economy benefits and good exhaust emissions under stratified conditions and a significant power advantage under homogeneous conditions, compared to equivalent PFI engines. All of these studies were directed on four-valve engines with the majority employing wall-controlled combustion systems [1-9], although, very recently, the jet-controlled combustion system, which is based on the early SIDI engines [10-14], has been revisited [15-17] with good results.

The work described in this paper was performed on a 2-valve SIDI engine, with a wall-controlled combustion system. The application of the multi-cylinder version of this engine is predominantly to light trucks. These engines employ only 2 valves due to their displacement and relatively lower operating speed, compared with the 4 valves commonly used for car applications. However, larger displacement engines may also show significant benefits in fuel economy when the throttling losses associated with part-load operation are eliminated. The present study was performed as part of a wider study that included a variety of piston, combustion chamber and port geometries at different loads and speeds.

At the part-load test point used for screening, results are presented on the influence of in-cylinder swirl motion on the performance and emissions of the engine. The CFD computations, which complement the experimental program, were made to relate the observed performance (fuel economy, combustion stability and misfires) and engine-out emissions characteristics of the SIDI engine to fundamental in-cylinder processes that describe the interactions of the fuel spray, flow field, and combustion-chamber walls. The goal of effectively coupling the CFD computations and the experiments is to improve the fuel economy and emissions of the engine by a thorough understanding of the fundamental mechanisms of mixture formation in this particular engine.

ENGINE AND TEST CONDITIONS

The single-cylinder experimental engine has two valves per cylinder and a pushrod valve train. Its main specifications are shown in Table 1, and its combustion-chamber geometry is presented in Figure 1.

Table 1. Engine Specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
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<tbody>
<tr>
<td>Bore</td>
<td>96 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>92 mm</td>
</tr>
<tr>
<td>Connecting Rod Length</td>
<td>154.9 mm</td>
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<tr>
<td>Piston Pin Offset</td>
<td>0.8 mm</td>
</tr>
<tr>
<td>Displacement</td>
<td>0.67 L</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>11.2:1</td>
</tr>
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</table>

The engine geometry is characterized by a) a split intake port, which allows for high in-cylinder swirl flows at part
load conditions, b) an offset, swirl-type injector, which
allows for variations in fuel-spray orientation, and is
mounted with a small inclination angle with respect to
the horizontal, and c) a piston with a shallow bowl and a
ridge wall for the containment of the fuel spray and of
the fuel-air cloud which is subsequently formed. The injector
was a swirl-type injector with a 70° cone angle, a 20°
offset and a rated fuel-mass flow rate of 15 cc/s at
10 MPa. A single part-load stratified test point was
chosen for optimizing key operational and geometric
parameters such as in-cylinder air swirl, and injector
orientation.

The engine speed is 1300 r/min, a representative part-
load speed for the truck application of the multi-cylinder
version of this engine. All the tests were run at constant
fueling rate of 13.2 mg/cycle, and the MAP was adjusted
to give an overall mixture air-fuel ratio of 45:1. Besides
the routine measurements of engine performance
(torque, fuel economy, exhaust temperatures, etc) and
engine-out emissions (NO, CO and HC), measurements
were also made of the gas pressure history in the
combustion chamber of the engine, and of the exhaust
smoke. The in-cylinder pressure measurements were
utilized to perform heat-release analysis, besides
computing the various net and indicated parameters
(e.g., IMEP, NMEP).

COMPUTATIONAL MODELING DETAILS

The corresponding computational fluid dynamics (CFD)
calculations were performed using GMTEC [18], an in-
house code for three-dimensional flow and combustion
calculations. The code solves the momentum, energy
and species conversation equations on an unstructured
grid and includes the capability to include non-conformal
interfaces that can slide relative to each other. This
capability was used to incorporate the valve motion in
the intake grid as well as to include the details of the
spark electrode and strap in the calculations in the
remapped grid, as shown in Figure 1. The conservative
remap is performed after intake valve closure to improve
the in-cylinder resolution for the spray and combustion
calculations. In addition, the ability of GMTEC to
adaptively refine the mesh in the spray region was
utilized to jointly optimize run time and grid density.
 Adequate grid resolution was found to be especially
important for the fuel vapor penetration. The base grid
for the closed-valve calculations consisted of 85,000
cells and at the peak of refinement 125,000 cells were
used.

The spray liquid phase is modeled in the common
particle Lagrangian fashion [19] and is coupled to the
gas phase via source terms. A custom injection routine
was developed to represent this specific swirl-type
injector by comparison with planar laser-induced Mie
scattering images and Phase Doppler Particle Analysis
(PDPA) point measurements. In addition, the TAB
breakup model [20] was used in conjunction with the
collision model of O'Rourke [19], which has been
modified to be independent of the gas-phase mesh. The
wallfilm is modeled using Lagrangian particles [21, 22]
and the spray-wall interaction is calculated using the
model of Stanton and Rutland [23]. An important feature
in the spray modeling is the representation of the fuel as
a multicomponent mixture by means of a continuous
distribution [24,25]. This affects the vaporization and the
predicted amount of wallfilm mass, and it was found that
using isooctane as a substitute for multicomponent
gasoline showed a substantial reduction in wallfilm
mass, in line with experimental observations [26]. The
parameters of the distribution function were chosen to
match the distillation characteristics of Indolene.

![Figure 1. View of computational grid (a) showing intake grid cutaway to highlight valve and (b) showing remapped grid with spark plug cross-sectional plane](image)

RESULTS: EFFECT OF SWIRL RATIO

The results for the influence of cylinder air motion, i.e.,
swirl index (SI), on the fuel economy, combustion
characteristics and engine-out emissions are presented
in Figure 2 to Figure 3. These data were obtained with
an injector orientation of +60° and an EOI = 42°bTDC.
partly caused by engine misfiring, although it produced good fuel consumption and engine-out emissions. With further decrease in swirl, the number of misfires and COV keeps increasing.

Both smoke and HC appear to monotonically decrease with increasing in-cylinder swirl level. In general one may conclude that high swirl levels produce significant improvements in the combustion stability and in both the HC emissions and smoke.

The highest swirl level tested (SI = 4.0) produced the lowest NSFC, the lowest COV, the lowest engine-out HC, and consequently highest combustion efficiency, significantly lower combustion duration, and the lowest smoke. The next-lower swirl ratio examined (SI = 3.3) resulted in combustion stability problems, which were partly caused by engine misfiring, although it produced good fuel consumption and engine-out emissions. With further decrease in swirl, the number of misfires and COV keeps increasing.

Both smoke and HC appear to monotonically decrease with increasing in-cylinder swirl level. In general one may conclude that high swirl levels produce significant improvements in the combustion stability and in both the HC emissions and smoke.

To understand the causes of the deterioration in combustion stability and emissions, it is informative to examine the details of the flow field and mixture preparation. The flow field at the start of injection at 50°bTDC shows some significant differences, as would be expected: for SI=4.0, Figure 4(a), there is a strongly organized swirling motion when the cylinder is viewed from above. This strongly organized motion persists throughout the injection event — there are some local changes due to the fuel penetration, but on the whole a similar swirling motion is still clearly evident at 20°bTDC. For SI=0.7, Figure 4(b), there is no such global organized motion evident, with multiple smaller structures clearly visible.

During injection, the high swirl deflects the fuel spray, esp. the fuel vapor cloud, towards the intake side, as seen at 38°bTDC, with little deflection in the case of SI=0.7. At the later time of 30°bTDC in Figure 4, the interaction of the injection-induced flow and the piston...
ridge again shows a different structure: the lower swirl case rises up the piston ridge and turns back on itself as evidenced in the velocity vectors, in part at least due to the squish flow and the interaction with the combustion chamber geometry. The high swirl case also shows a rising up along the piston ridge; however, the turning back is mitigated by the counter-clockwise swirl motion, which convects the fuel towards the center of the bowl and the spark plug. At 20°bTDC, just before ignition, this comes to light fully and is magnified in Figure 4(c).

In the case of the low swirl, the cutout on the piston ridge below the spark boss causes an earlier effect of the squish flow on the fuel cloud at that location, D, keeping it lower in the bowl and below the spark plug. Towards the intake side much of the fuel cloud rises up against the head, see E. In the high-swirl case, SI=4.0, the effect of the cutout and local squish flow can be seen in the double bulge of the iso-surface, see A and B. The bulge around the spark plug, B, is caused by the continued convection of the fuel cloud from the intake side towards the spark gap by the swirl: since it had risen up higher on the intake side, its lateral convection now feeds the spark gap at the appropriate height, close to the head. The bulge A is the result of the initial spreading of the fuel cloud as it interacted with the piston floor and ridge, and the difference in height between these two is the effect of the local squish flow through the piston cutout. A further effect of the swirl can be seen in the reduced penetration of the fuel cloud towards the head on the intake side, comparing C and E. The swirl thus initially deflects more of the fuel towards the intake side, and then progressively feeds this towards the spark plug, while a greater general dispersion is present for the low swirl case.

The cause of the large increase in COV and misfires with reduction in swirl index can best be understood with reference to Figure 5, showing the cross-sectional planes at 20°bTDC. In the light of the discussion of Figure 4, it is not surprising to see larger spatial variability of the mixture around the spark gap for SI=0.7, and the reduced mixture strength itself. The fuel cloud, in the plane of the spark plug, does not rise fully against the head and the spark gap as it does for the SI=4.0. This is attributable to the mechanism just described above. Given that there will inevitably be cyclic variation in the fuel distribution, it is very probable that this reduced mixture strength and larger spatial variability in the vicinity of the spark gap would be the cause of the increased COV and misfires noted for the lower swirl case(s) [27]. It is also clear that for this engine configuration, there is a significant and complex interaction of the injection-induced flow, the swirl flow, the squish flow and the combustion chamber geometry.

The initial spray deflection can be seen in Figure 6 by means of the wall film location: it is evident that the lower swirl case has more film towards the center of the piston floor and closer to the center of the piston ridge, the former indicating an earlier impingement of the liquid spray on the piston floor. This is born out in the quantitative prediction of the film mass in Figure 6(c), which indicates a higher amount for the lower swirl case. This higher wall film mass contributes to the increased smoke emissions, and possibly the higher HC emissions, evidenced in the experimental measurements [28-30].

CONCLUSIONS

For the best compromise between fuel consumption, combustion stability, engine-out hydrocarbon emissions and smoke, the experimental 2-valve, SI engine required high swirl levels (swirl ratio ~ 4), and a spray orientation that is directed towards the intake-valve side and targets the ridge wall of the piston. The stratified combustion was thus found to be a combination of wall and charge control, as indicated by the indirect spray targeting relative to the spark location coupled with the high swirl requirements of the engine, as well as the crucial role of the piston ridge in deflecting the fuel vapor towards the spark gap. The COV of IMEP and misfires decreased with increasing in-cylinder swirl motion, indicating increased combustion stability. These agree with the CFD computations, which showed that decreasing the air motion may result in lean air-fuel ratios in the neighborhood of the spark plug at the time of ignition, which, in turn, may cause misfires. Also, for the high-swirl valve positioning the in-cylinder air motion is strongly organized, whereas, for the low-swirl valve positioning, the flow structure has noticeable smaller flow structures, which suggest increased cycle-to-cycle variability. To minimize misfires and cyclic combustion variability, the time-averaged air-fuel mixture in the immediate vicinity of the spark gap at the time of ignition appears to require a definite bias towards a rich mixture. Exhaust smoke, and to a lesser degree engine-out hydrocarbons (HC), were found to be sensitive magnitude of the in-cylinder air motion. Both decreased with increasing swirl motion. CFD computations suggest that the above observed decrease in smoke and HC may be attributed to the resulting decrease in piston wetting (amount of liquid fuel on the surface of the piston) and decrease in the fuel vapor concentration near the piston floor, with increasing swirl ratio.

ACKNOWLEDGEMENTS

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NOMENCLATURE

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<th>Description</th>
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<tr>
<td>COV</td>
<td>Coefficient of variation</td>
</tr>
<tr>
<td>EOI</td>
<td>End of injection</td>
</tr>
<tr>
<td>SI</td>
<td>Swirl index (also swirl ratio)</td>
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<tr>
<td>$\phi$</td>
<td>Equivalence ratio, defined as $\phi = (A/F)_{stoichiometric}/(A/F)$</td>
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REFERENCES


Figure 4. Comparison of mixture preparation by means of an isovolume containing fuel-air mixture of $\phi \geq 1.4$; (a) SI=4.0, EOI=42, (b) SI=0.7, EOI=42. The spark electrode and strap are shown but the chamber head is not shown for visual clarity. The plane containing the velocity vectors is at the height of the injector. The vector length for (b) has been increased to show the flow details more clearly. (c) Shows enlarged comparison at 20°bTDC.
Figure 5. Cross-sectional planes at 20°bTDC for 3 stations (11m towards intake side, in plane of spark plug, and 18mm towards exhaust side) colored by equivalence ratio with velocity vectors overlaid; (a) SI=4.0, (b) SI=0.7.

Figure 6. Comparison of wallfilm location at peak wallfilm mass timings for (a) SI=4.0, (b) SI=0.7; and (c) temporal variation of total wallfilm mass.