The Effect of Piston Bowl Offset on the Compression - Induced Air Motion in Direct Injection Diesel Engine Combustion Chambers

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ABSTRACT

The paper describes a theoretical investigation into the effects of an offset of the piston-bowl combustion chamber from the cylinder axis on the compression-induced flow in a diesel engine. This is achieved by calculating the 3-dimensional flow field during the compression period and early stages of expansion. For a 2-dimensional (axisymmetric) computations are also made for the same configuration and conditions. The results of the study reveal that bowl offset produces a higher decay rate of swirl momentum; precession of the swirl structure within the cylinder; a variation of squish velocity around the bowl and consequently a different flow structure within the bowl at TDC.

INTRODUCTION

During the last few years, computational fluid dynamics (or "Multidimensional Modelling", as it is often referred to) has become an increasingly powerful tool for investigating flows in engine cylinders and combustion chambers. The majority of flow calculations reported in the literature have been based upon 2-dimensional (axisymmetric) representations of the in-cylinder geometry and it is only recently that extensions to 3-dimensions have been made. Nevertheless, it is likely that 2-dimensional calculations will continue to be used extensively for routine studies as they have the advantage over their 3-dimensional counterparts that they are much less demanding of computer resources and easier to understand. The question therefore arises as to the conditions that must be fulfilled for the assumption of 2-dimensionality to be satisfactory.

Clearly, there are many instances for which 3-dimensional representations are necessary, such as the in-cylinder flow generated during intake. There are, however, a number of cases where 3-dimensional effects may be negligible. One of the more important of these is the compression-induced flow in the piston-bowl combustion chamber of direct injection diesel engines and certain designs of lean-burn and stratified charge gasoline engines. In the absence of fuel injection and combustion, there are two main aspects which may cause the flow field to be 3-dimensional. Firstly, the residual flow field at inlet valve closing is usually strongly 3-dimensional, as evidenced in both measurements (1) and calculations (2). In particular, swirling flows with tilted and precessing swirl centres have been observed. Secondly, geometric features of the combustion chamber or its positioning, such as valve recesses in the piston or non-coincidence of the cylinder and combustion chamber axes, will invariably lead to 3-dimensional effects. The importance of these is, as yet, unknown and will undoubtedly form the subject of many future investigations.

The present contribution focusses on one of the above parameters, namely the effect of offset of the piston bowl from the cylinder axis. This situation prevails in the vast majority of small D.I. diesel engines and it is desirable to maximise the inlet and exhaust valve sizes which dictates that the injector is offset from the cylinder axis and, in an attempt to maintain symmetry when using multi-hole nozzles, it is also necessary to displace the piston bowl from the cylinder axis. Typically, this is half the injector offset and may be up to 15 ° of the cylinder bore. It is therefore of interest to know what effect this offset has on the flow and whether axisymmetric representations are adequate for such configurations.

The paper first reviews briefly the salient findings of previous experimental and theoretical studies of flows in piston bowls, as these are of relevance to the work reported here. Subsequently, the main assumptions and results of the present study are described. This is concerned with the calculation of the flow in a D.I. diesel engine combustion chamber, the axis of which is offset 12 ° from that of the cylinder.

* The offset is defined as the distance between the cylinder and bowl axes divided by the cylinder diameter.
REVIEW OF PREVIOUS WORK

Many experimental and theoretical studies have investigated the flows in D.I. diesel engine combustion chambers. Some of the more important findings are summarized below:

- Squish velocities (radial inflow into the piston bowl near TDC of compression) at the edge of the bowl increase with increased squish area and reduced TDC bumping clearance (3,4).

- Swirl reduces the penetration of the squish jet towards the bowl axis due to the interaction of the radial momentum and centrifugal force (5,6). This becomes more pronounced with higher swirl levels. However, calculations indicate that the peak mass flux of the squish inflow at the edge of the bowl is unaffected by swirl (6).

- There is experimental evidence (1,7,8) that squish velocities may be considerably lower (20% and 40% of expected values in references (7) and (8), respectively) than those calculated using either simple theory or multidimensional calculations. It has been suggested that heat transfer and/or blowby may be responsible for the discrepancies. However, multidimensional calculations - which include heat transfer, indicate that the squish velocity reduction is only a few percent. As to the effect of blowby, unreported investigations at AVL into the simultaneous effects of blowby and dead volume above the top ring land showed that, for an engine in good condition (total blowby = 2% of swept volume/cycle at low engine speed), the squish velocity was only slightly reduced.

- The answer may lie in the uncertainties associated with the experimental data (this is a difficult experiment as the LDV probe volume length is, typically, the dimension of the clearance gap and must be positioned "on the wall" and at the edge of the bowl) or in determining the TDC bumping clearance accurately under running conditions. In real engines, the problem is often compounded by valve recesses. For the present, these discrepancies remain unresolved.

- When the bowl is in the centre of the cylinder and in the absence of swirl, squish invariably induces a single toroidal vortex in the bowl (5,9,10). In the presence of swirl, the in-bowl flow becomes more complicated with the appearance of two or more toroidal vortices (5,6,9,10,11). In diesel engines operating with near-optimum swirl, the dominant vortex is usually in the opposite rotational sense to that found under non-swirling conditions. It has also been observed in experiments using grease smeared on the piston surface, to indicate the direction of air movement, that an "equator" (which signifies the existence of a stagnation point) is formed on the side wall of the bowl (11,12).

- Local turbulence and swirl maxima are produced in the bowl entrance from the squish inflow (1,6,9,10,13). The position of these maxima is dependent upon the squish penetration which, in turn, depends upon the swirl level. Local maxima may also be observed near the edge and bottom of the bowl after TDC due to the redistribution of swirl momentum by the induced toroidal vortex (1,14).

After TDC, the swirl turbulence decays rapidly in the upper part of the bowl (8,14). This is not true for highly re-entrant bowls which may show a second, post-TDC maximum at the top of the bowl due to the "spin-up" effect as the swirl momentum forced into the bowl before TDC is transported back into the cylinder via the bowl entrance.

Reverse squish produces a local turbulence maximum between the edge of the bowl and the cylinder head. This is more pronounced with straight rather than rounded bowl edges. One important consequence of this is increased heat transfer in this region (15).

- In general, the swirl does not rotate about the bowl axis when either the inlet valve or the bowl are offset from the cylinder axis (1,8,11,16). The position of the swirl centre varies with bowl depth, crank-angle and engine speed. It has been observed that rotation of the swirl centre is in the opposite direction to that of the swirl during compression (8). Even in axisymmetric model engines, the position of the swirl centre may vary from cycle to cycle (17) and therefore make an erroneous contribution to ensemble-averaged turbulent velocities.

- In bowls which are offset from the cylinder axis, stronger squish is produced on the side of the bowl opposite the direction of offset (14).

- There is indirect evidence from engine performance data that the in-bowl swirl may be reduced when the bowl axis is offset from that of the cylinder. A 30% reduction in swirl was estimated for a 10% bowl offset in reference (18).

- Estimates of the swirl ratio (equivalent solid body swirl speed divided by engine speed) and swirl momentum at inlet valve closing (IVC) and at TDC have been made by a number of researchers from swirl velocity measurements in the cylinder and piston bowl. For comparison, these are given in table 1 below.
<table>
<thead>
<tr>
<th>Reference</th>
<th>( \text{SR}_{\text{TVC}} )</th>
<th>( \text{SR}_{\text{TDC}} )</th>
<th>( \text{SR}_{\text{IVC}} )</th>
<th>( \Omega_{\text{TVC}} )</th>
<th>( \Omega_{\text{TDC}} )</th>
<th>( \Omega_{\text{IVC}} )</th>
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<tbody>
<tr>
<td>(1)* Tangential Port</td>
<td>3.0</td>
<td>5.7</td>
<td>1.9</td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>(1)* Shallow ramp Helical port</td>
<td>3.5</td>
<td>7.6</td>
<td>2.2</td>
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<tr>
<td>(1)* Steep ramp Helical port</td>
<td>3.5</td>
<td>7.9</td>
<td>2.25</td>
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<tr>
<td>(17) Axisymmetric model engine with idealised re-entrant bowl</td>
<td>2.5</td>
<td>5.0</td>
<td>2</td>
<td>1</td>
<td>.6</td>
<td>9.6</td>
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<tr>
<td>(17) As above</td>
<td>6.9</td>
<td>11.0</td>
<td>1.6</td>
<td>2.4</td>
<td>1.27</td>
<td>0.53</td>
</tr>
<tr>
<td>(8) Centrally-located cylindrical bowl</td>
<td>6.6</td>
<td>10.9</td>
<td>1.65</td>
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Notes: 
- SR = Swirl Ratio
- \( \Omega \) = Angular Momentum
- * = Values at 1440 rev/min

Table 1 Comparison of IVC and TDC Swirl Levels from Various Experimental Studies

PURPOSE AND DESCRIPTION OF THE STUDY

The purpose of this study is to elucidate the 3-dimensional flow effects that may arise when the piston bowl is offset from the axis of the cylinder. For this investigation, multidimensional flow calculations have been performed for both 2-dimensional (axisymmetric) and 3-dimensional (offset bowl) representations to provide a comparison. Details pertaining to the calculations are given below, whilst further details of the methodology can be found in reference (19).

Engine Geometry and Computational Mesh

The combustion bowl shape, which is typical of those used in current high speed D.I. engines, and computational mesh arrangement for the 2-dimensional case are shown in figure 1. Figures 2 and 3 show the circumferential-radial plane mesh and the complete mesh used for the offset bowl case. The 2- and 3-dimensional meshes contain 533 and 15990 active cells (i.e., cells inside the solution domain and excluding those necessary for storing boundary conditions) respectively. Various "I" and "K" planes are indicated in figures 1 and 2 as these have been selected in which results are presented. Geometric data relevant to
the engine geometry are given below:

Bore x stroke = 85 mm x 94 mm
TDC clearance = 0.7 mm
Bowl entrance diameter = 32 mm
Bowl offset = 0 and 10.2 mm (12 %)
Compression ratio = 22
Speed = 1700 rev/min

Initial and Boundary Conditions

Calculations were started at inlet valve closing (225° = 45° ABDC) and terminated at 390° (30° ATDC). The initial pressure and temperature were determined from a prior gas-dynamics calculation whilst the turbulent velocity and length scale were assigned values of 5 m/s (= .95 x mean piston speed) and 1.0 mm respectively and assumed spatially uniform.

The swirl level and its axial variation within the cylinder, shown in figure 4, were taken from the helical port intake calculation of reference (2) at IVC. Unfortunately, that study was for a larger engine (120 mm x 120 mm) and the swirl level was lower than would be expected from the smaller engine configuration used here (IVC swirl ratio = 1.9 in reference (2)) compared with an estimate of 2.35 for this engine using a method similar to that described in reference (20). However, as the study is comparative in nature, this discrepancy is not important.

In the absence of a prior intake calculation with the combined valve and bowl in their correct positions, two further uncertainties arise in setting up the initial swirl structure. The first relates to the importance of an initially off-axis and tilted orientation of the swirl centre in the cylinder. Here, it has been assumed that the swirls in both the cylinder and the bowl rotate about their respective axes. The second concerns the axial swirl variation in the bowl. This is, however, unlikely to be of great importance as the bowl contains only 1 % of the total swirl momentum at IVC. It has been assumed that the in-bowl swirl is of solid body structure with a swirl ratio equal to that at the bottom of the cylinder.

RESULTS

Axisymmetric Calculations

Before discussing details of the flow structure, it is worth examining the variation of swirl momentum during the calculation period. Figure 5 shows the distribution of swirl momentum in the cylinder and bowl and the total momentum, these being normalised by the total swirl momentum at IVC. Here, it can be seen that 22 % of the total momentum is lost up to TDC and, at TDC, the bowl contains 54 % of the IVC momentum. This gives an equivalent solid body swirl ratio in the bowl of 5.3. As the predominant mechanism of swirl decay is surface friction, the momentum decay rate is directly related to the surface area to volume ratio and the swirl speed, which is the reason for the
increased decay rate near TDC. It should be noted that the rate of momentum reduction in the bowl after TDC is lower than the rate of increase before TDC. This feature is only observed in re-entrant bowls and is caused by the "storage" of swirl momentum in the lower outer regions of these bowls.

In the 2-dimensional case, the most interesting features of the flow occur around TDC and discussion is therefore restricted to this period. Figure 6 shows the velocity field at 340° (20° BTDC), the swirl field being depicted as contours. This shows a strong squish flow into the bowl and a swirl maximum at approximately 2/3 bowl radius. By 355° (5° BTDC), shown in figure 7, it can be seen that a double vortex structure has developed within the bowl with the smaller but stronger squish-induced vortex immediately adjacent to the edge. The squish penetrates to 1/2 bowl radius where the local swirl velocity maximum is also found. The double vortex structure has developed further by 360° (TDC), shown in figure 8, especially near the axis whilst the swirl structure is similar to that at 355°. After TDC (figure 9) the vortex structure in the bowl entrance is quickly destroyed by the outflow. The swirl, however, now shows maxima near the central pip and on the flank, these both being caused by the inward redistribution of swirl momentum from the bottom of the bowl.

Figure 6 Velocity Field at 340° for the Axisymmetric Configuration

Figure 7 Velocity Field at 355° for the Axisymmetric Configuration

Figure 8 Velocity Field at 360° for the Axisymmetric Configuration
Offset Bowl Calculations

Figure 10 shows the variation of normalised swirl momentum in the cylinder and the bowl during the compression and early expansion periods. As with the initial conditions, these are defined with rotation about the cylinder and bowl axes respectively. In comparison with figure 5, it is immediately apparent that the momentum reduction is much higher than for the axisymmetric case, with only 27% of the total momentum remaining at TDC. As the cylinder contains nearly all the momentum for most of the compression stroke, the reasons must be sought here.

Figure 11 shows the flow in 3 diametral planes of the cylinder at 270°. The axial position of each plane is indicated on the inset sketch whilst the cylinder centre is marked by a solid circle. The bowl is offset to the left side of the cylinder axis, as in Figure 2. At the bottom of the cylinder, it can be seen that the centre of the swirl has moved towards the bowl axis and has been simultaneously displaced in a direction normal to the line joining the cylinder and bowl centres. In the mid-plane of the cylinder, the swirl-centre has moved in a direction opposite to that of the offset whilst in the upper part of the cylinder, it has shifted in a direction perpendicular to the geometric symmetry line.

To try and understand why the position of the swirl centre moves through the compression stroke, its location was obtained from the graphical output at each crank-angle. Figure 12 shows the locus of the swirl-centre in planes immediately above the piston and in the mid-plane of the cylinder (it was not possible to determine unambiguously the swirl centre at the top of the cylinder through the whole compression stroke). This shows that the swirl centre moves in a direction opposite to the main swirl and that its axis of rotation is tilted with respect to the cylinder axis. Figure 11 also suggests that it is curved. Clearly, the reason why the swirl centre moves at all is because of the offset position of the bowl, thus, the "driving force" for movement of the swirl centre originates at the bottom of the cylinder and is likely to be directed towards the bowl. Intuitively, it would be expected that the swirl centre in the upper planes would follow the movement in the lower planes, but with a phase lag, due to friction. However, it is significant that the swirl centre in the mid-plane moves initially in the opposite direction, that is, away from the bowl. This, together with
the shape of the locus up to 300° (figure 12), suggests that the swirl structure is undergoing gyroscopic precession rather than being simply "dragged" towards the bowl, as shown schematically in figure 13. Nearer to TDC, the centres become nearly coincident, which is compatible with the increased effects of turbulent "frictional" momentum transfer between adjacent layers due to higher axial gradients in swirl velocity caused by the reducing distance between the cylinder head and piston.

The tilt angle to the cylinder axis was estimated to be 10° and 17° at 250° and 300° crank-angle respectively. As the calculated swirl momentum is defined with rotation about the cylinder axis, tilting of the axis of rotation of the swirl will, in itself, bring about an apparent reduction in momentum. However, as this varies with the cosine of the tilt angle, the reduction amounts to only a few percent.

It was found from examination of the momentum loss in each diametral plane of the cylinder that, although the momentum was lower in every plane compared with the 2-dimensional case, there was a greater loss towards the bottom of the cylinder. As frictional decay was found to be nearly the same for the 2- and 3-dimensional cases, other mechanisms were sought for the
momentum loss. The first possibility is that the swirl momentum transferred from the cylinder into the bowl is higher for the offset bowl configuration. Taking an idealised situation, where the air in the cylinder immediately above the bowl is displaced into the bowl, and ignoring the movement of the swirl centre, the swirl momentum associated with the displaced air is estimated to be 80% higher for the offset bowl because of the higher swirl velocities relative to the cylinder axis. However, this loss of swirl momentum from the cylinder does not register as a gain for the bowl, as rotation of the air transferred into the bowl is defined about the cylinder axis.

The second possibility arises because of the effects of squish inflow into the bowl. Figure 14 shows the decomposition of a squish component, taken to be directed along a radial line to the bowl centre, into radial and swirl components with respect to the cylinder axis. It can be seen that in sector "A" of the cylinder, this produces a negative contribution to the swirl, whilst in sector "B" a positive contribution results. Clearly, for an engine operating without swirl, the flow would be symmetrical about C-C and the net swirl would remain zero. It is suggested that, when swirl is present, the effects in each sector do not cancel and that the negative contribution in sector "A" outweighs the positive contribution in sector "B". As will be seen, this hypothesis is certainly supported nearer TDC when squish becomes more important but it is also surmised that this occurs throughout the compression stroke.

Figure 15 shows the flow field in planes K = 16 & K = 1 and K = 24 & K = 8 at 320° (40° BTDC). There is, apparently, strong squish into the bowl from planes K = 16 and K = 24 compared with their opposite planes. However, as seen in the mid-diametral plane of the cylinder (figure 16) the "squish" contains a substantial component from the swirl. It can also be
At 340° (20° BTDC) the swirl centre is nearly coincident with the bowl axis (see figure 12) and the squish flow into the bowl is more symmetrical than before, as seen in figure 18 for planes \( K = 16 \) & \( K = 1 \) and \( K = 24 \) & \( K = 8 \). This is the closest that the flow structure in the bowl approaches 2-dimensionality and is between the period when the swirl has been nearly centred on the bowl axis and before squish reaches its maximum. Within the cylinder, however, there is a dramatic reduction in
swirl in sector "A" (see figure 14) due to the squish, as shown in figure 19. Typically, the swirl component in sector "B" is four times that in sector "A".

The peak squish velocity is reached at 352° (8° BTDC) and the effect of bowl offset is to produce a variation of squish velocity around the bowl. Figure 20 shows the variation around the bowl of the axially-averaged squish velocity at three different crank-angles. This quantity was obtained by equating the mean mass flow rate to the sum of the individual mesh cell mass flow rates through each sector and is shown schematically in the inset sketch. In figure 20, the magnitude of the squish velocity can be obtained from the product of the scaling factor and the distance along a radial line between the bowl outline and the respective curve to the reference length. At 300° and 330° the profile is strongly biased by the swirl component, but, when the squish reaches a maximum immediately before TDC, the profile is symmetrical about the 0°-180° plane. It can also be observed that the squish component at the 180° position is 50% greater than at the 0° position. This occurs because of the greater distance between the bowl edge and the cylinder wall at the 180° location.

As previously mentioned, the squish penetration distance into the bowl is determined by the "centrifugal barrier" presented by the swirl and, as the swirl velocities are lower in sector "A", the flow within the bowl is highly asymmetric with air "spilling over" from sector "A" (planes K = 8 and K = 12) to sector "B" (planes K = 24 and K = 27), as seen in figure 21.
Within the cylinder, the swirl is now dramatically reduced with large regions in sector "A" exhibiting a negative swirl component, shown in figure 22 for the mid-plane section.

By TDC, the flow within the bowl has developed strong asymmetries as shown in the series of planes in figure 23. As can be seen, the centre of the toroidal vortex varies from near the top of the bowl and adjacent to the lip in planes $K = 1$ & $K = 5$ to deep inside the bowl in planes $K = 16$ & $K = 20$. In the cylinder, figure 24 indicates that the swirl in sector "A" has been totally destroyed even above the bowl. Within the bowl itself, there is a distinctly asymmetric swirl structure close to the bowl entrance (figure 25 - plane $I = 14$) with negligible swirl velocities near the surface of the piston in the sector $K = 20$ to $K = 30$.

After TDC, the flow contains a number of features not seen in the axisymmetric calculation. Figure 26 shows that, by 380°, the reverse squish has regenerated the swirl in sector "A", where it had previously disappeared, and destroyed the remaining swirl in sector "B". Although not shown in figure 20, the reverse-squish velocity variation around the bowl is similar to the inflow at 335° but, of course, in the opposite direction. Impingement of the reverse squish on the cylinder wall around plane $K = 16$ has also created a narrow recirculation zone around the periphery of the cylinder, however, the mesh density is insufficient to resolve this properly. What is particularly interesting is the difference in outflow from the bowl.
Figure 25 Velocity Field in the Bowl (Plane I = 14) at 360°

Figure 26 Velocity Field in the Mid-Diametral Plane (I = 5) of the Cylinder at 380°

Figure 27 Velocity Field in Various Axial-Radial Planes at 380°

between planes $K = 1$ and $K = 16$, as shown in the series of plots in figure 27. Although not seen in figure 27, there is still a strongly swirling flow within the bowl.
SUMMARY AND CONCLUSIONS

This investigation into the effects of offset of the piston bowl axis from that of the cylinder has revealed a number of interesting phenomena which do not occur in axisymmetric configurations. These are summarized below:

1. Swirl momentum is lost or destroyed at a much higher rate. No single mechanism could be found to explain this and the total loss is probably the cumulative effect of a number of factors, including:
   a) Problems in the definition of swirl momentum due to tilting of the axis of rotation.
   b) Convective transport of higher swirl momentum (relative to the axisymmetric case) from the cylinder into the bowl.
   c) Destruction of swirl momentum by the squish.

The last of these appears to be the dominant mechanism near TDC.

2. The centre of rotation of the swirl varies with both crank-angle and depth in the cylinder. During compression, the swirl centre rotates in an opposite direction to the swirl and starts to follow a reverse path during expansion. The locus of the swirl centre in different planes suggests that the entire swirl structure is precessing. As the piston approaches TDC the swirl centre loci become nearly coincident and close to the bowl axis.

3. At the time when the squish flow becomes significant, immediately before TDC, the mean squish velocity around the bowl is symmetrical about the geometric symmetry plane (K = 1 & K = 16, which is "C-C" in figure 14). However, there is significant squish velocity variation around the bowl, with 50% higher values on the side opposite the direction of offset.

4. In spite of the symmetry of the mean squish velocity profile, the penetration of squish inside the bowl is not symmetrical and the air "spills over" from certain sectors of the bowl to the opposite side, producing an asymmetric flow in the bowl. This, coupled with the reduced in-bowl swirl produces a different TDC flow structure to that found in the axisymmetric configuration.

5. After TDC, the flow structure within the bowl caused by the reverse-squish outflow varies significantly around the bowl. The reverse squish also produces a narrow recirculation zone around the periphery of the cylinder. Squill is regenerated in the sector in which it was destroyed during compression ("A" in figure 14) and destroyed in the other sector ("B").

REFERENCES


