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Special Issue Article

International Journal of ENGINE RESEARCH

International J of Engine Research
2016, Vol. 17(1) 6–15
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DOI: 10.1177/1468087415604754
jer.sagepub.com
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A review of design considerations for light-duty diesel combustion systems

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Abstract
Practical aspects of light-duty diesel combustion system design are reviewed, with an emphasis on design considerations reported by manufacturers and engine design consultancies. The factors driving the selection of compression ratio, stroke-to-bore ratio, and various aspects of combustion chamber geometry are examined, along with the trends observed in these parameters in recently released engines. The interactions among geometric characteristics, swirl ratio, and the fuel injection nozzle parameters are also reviewed.

Keywords
Engines, diesel, combustion, design

Date received: 24 May 2015; accepted: 13 June 2015

Introduction

Diesel engine combustion system design is an exceedingly complex undertaking, requiring a mastery of multiple fields of engineering, chemistry, and physics. Thousands of man-years of effort have been expended in balancing multiple trade-offs as basic design parameters such as compression ratio, bore-to-stroke ratio (B/S), and combustion chamber geometry are selected. Despite this complexity, engineers have developed a number of guidelines that capture the experience gained from this effort. Our primary objective in this article is to provide a concise summary of these guidelines.

Moreover, there are multiple, often conflicting, drivers that impact engine design,[1] including the following:

- Compliance with emission regulations
- Improvement in fuel economy or reduction in CO_2 emissions
- Requirements for vehicle performance
- Factors impacting consumer acceptance, such as noise and vibration
- Durability, reliability, and maintenance considerations
- Compact, light-weight packaging
- Cost

The relative importance of these drivers, and the specific restrictions they place on the design process, varies with the market segment being targeted. In the limited space available here, we can only focus on a few of these drivers and will be primarily concerned with the first three and how they impact the design decisions.

Finally, our focus here is primarily on light-duty diesel engines suitable for passenger car applications. Although many of the considerations apply directly to heavy-duty truck, marine, and locomotive engines, there are a number of critical differences between light-duty and heavy-duty engines. Heavy-duty engines often operate at high loads, with long injection durations. Under these conditions, much of the energy needed to complete the fuel–air mixing process is provided by the fuel spray, and a useful understanding of the combustion process can be obtained by considering the combustion of a free jet in the absence of combustion chamber walls.[2] In contrast, light-duty engines are optimized for the low-to-medium loads typical of an urban drive cycle. Under these conditions, injection can be mostly complete before combustion commences, and the spray is unable to deliver all the kinetic energy needed to drive the mixing. The energy deficit is largely supplied by mean flow swirl, which must be carefully matched to the fuel injection parameters. Moreover,

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interactions between the sprays and the walls of the combustion chamber (the piston bowl) are much more pronounced, and the combustion performance and emissions are generally found to be very sensitive to the bowl geometry.

Top-level engine parameters

Cylinder displacement

The highest level parameter impacting diesel engine design is the engine displacement. By benchmarking current production and prototype engines, brake mean effective pressure (BMEP) and peak power density targets can be defined which, in concert with the torque and power requirements of the specific vehicle application, define the required displacement. With the further selection of the engine topology (number of cylinders), the individual cylinder displacement is defined.

While high power or torque density is always desirable, the benefit must be weighed against additional expense associated with a stiffer block and head structure, more stringent charging requirements, more effective aftertreatment, and more capable fuel injection equipment. Different manufacturers have adopted distinctive design philosophies based on these trade-offs. Many manufacturers have chosen small displacement engines coupled with high power and torque densities. These down-sized engines spend more time operating at higher loads, where friction and heat transfer losses are a smaller fraction of the total fuel energy released. In contrast, some designers argue that by increasing displacement, lower boost levels, lower injection pressures, and lower swirl ratios can be employed. Consequently, friction and heat transfer losses are reduced and improved fuel economy can be obtained at the same emission levels. The gearing can also be adjusted while maintaining similar vehicle performance levels, leading to a further reduction in fuel consumption.[3] Moreover, due to the non-linear increase in NO_x as torque increases, employing larger displacement engines can help control engine-out NO_x emissions.[4,5] Finally, a larger displacement will generally result in a lower combustion chamber surface area-to-volume ratio, and thus in lower relative heat transfer losses.

Compression ratio

Equation (1), which is applicable to both constant pressure and constant volume heat release processes, shows that increasing the compression (or expansion) ratio r_c increases the theoretical thermal efficiency η_{th} of the engine

$$\eta_{th} = 1 - \frac{1}{r_c^{\gamma-1}}\left[\frac{\alpha\beta^{\gamma} - 1}{\alpha\gamma(\beta - 1) + \alpha - 1}\right] \tag{1}$$

In equation (1), α represents the pressure ratio associated with the constant volume portion of the heat release, β represents the volume ratio associated with the constant pressure portion of the heat release, and γ is the charge specific-heat ratio.[6] For the following reasons, however, the efficiency advantages of a high compression ratio may not always be achieved:

- It may be necessary to retard the combustion event to remain below the allowable peak cylinder pressure or engine-out NO_x emissions when a high compression ratio is employed.[7,8]
- Higher r_c results in a higher compression pressure and, for an equal amount of heat release, a larger pressure rise due to combustion. The higher cylinder pressures increase ring and journal bearing friction.
- The surface area-to-volume ratio near top dead center (TDC) increases as r_c increases, resulting in higher heat losses to the piston and head surfaces (see Figure 1).
- Air utilization becomes more difficult with higher r_c. It is common to evaluate the potential for full air utilization using the so-called *k*-factor, defined as the ratio of the volume inside the bowl to the total volume at TDC. The reduction in *k*-factor shown in Figure 1 as r_c increases results in poorer air utilization, lower maximum torque/power at high loads, and hence reduced efficiency at the highest loads.

Overall, reasonable variations in compression ratio are only expected to have a limited impact on efficiency.[6,8,9] However, there can be other advantages conferred by a high r_c:

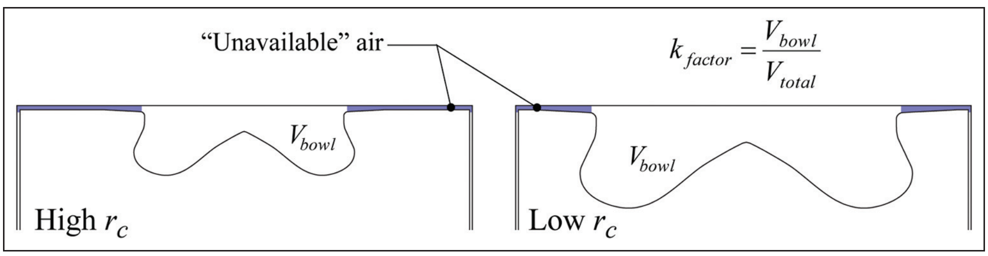

Figure 1. Illustration of the impact of compression ratio on surface area-to-volume ratio and on *k*-factor.

- Higher compression temperatures shorten ignition delay and can help reduce engine-out HC and CO emissions. This is especially beneficial during the first few minutes of operation, before diesel oxidation catalyst (DOC) light-off.
- Shorter ignition delay may also reduce combustion noise and increase combustion stability. Moreover, with low r_c, robust pilot injection strategies must be adopted to recover acceptable noise and stability.[5,10] Larger pilots injected early into low-density gases can over-penetrate, causing oil dilution and cylinder wear.
- Low-speed, mid-load soot emissions can suffer[11] when r_c is low, despite the expected greater premixing associated with longer ignition delay.
- High r_c engines have lower exhaust temperatures and can thus tolerate higher BMEP at high-speeds, when the maximum torque is limited by exhaust gas temperature.
- Higher r_c facilitates cold start.

On the other hand, there are potential advantages to lowering r_c:

- Lower r_c allows greater specific power at a given peak firing pressure—this is one of the main reasons for the current trend of decreasing compression ratios in diesel engines.
- Lower compression and peak combustion temperatures lower NO_x formation. Ignition delay is also increased, mitigating soot formation through increased fuel–air premixing.
- Higher exhaust gas temperatures allow more energy extraction by the turbocharger. Higher boost levels can thus be achieved at low speed, improving the low-speed torque characteristics.[11,12]
- DOCs operate more efficiently with higher exhaust gas temperatures, and post-DOC HC and CO levels can be reduced[13] despite higher engine-out emissions. Faster catalyst light-off is also expected.
- Lower r_c reduces the rate of charge cooling due to expansion, providing more time for oxidation of soot and other products of partial combustion.[8,14]
- Lower r_c may allow the removal of piston top valve pockets, potentially improving the k-factor.
- At low load, low r_c engines may be able to meet soot emission targets with lower injection pressures, resulting in improved brake-specific fuel consumption (BSFC).[11]

In the end, selection of compression ratio is dominated by considerations related to power density, emission characteristics, and startability rather than efficiency and fuel consumption.

B/S

Diesel engines are almost always "under-square" or "long stroke"—with B/S typically ~0.9. Although B/S will clearly impact engine packaging, it also significantly affects efficiency and combustion system design. In many respects, the relatively low B/S of typical diesel engines is surprising, as a large B/S can have many advantages:

- Friction is the most frequently cited factor that is influenced by B/S. Piston ring "boundary" friction[6] is thought to scale with S/B^2, which for a fixed engine displacement scales as $(B/S)^{-4/3}$. Likewise, viscous piston friction can be shown to scale with $(B/S)^{-2/3}$. Overall, it is reasonable to anticipate that piston assembly friction scales approximately inversely with B/S, and short-stroke engines will exhibit lower friction.
- Piston speed is reduced with large B/S, thereby allowing greater engine rotational speeds and potential for higher peak power.
- With large B/S, a larger area is available for the inlet valves, allowing higher intake air mass-flow rates and thus higher engine power or torque density.
- A larger bore also allows a wider bowl, which may be advantageous at high engine speeds[12] and reduces the risk of liquid fuel wetting the bowl wall. This is mainly a problem during cold conditions when vaporization is impeded.[15] Large B/S may also be well-suited to low r_c designs due to the larger fuel jet penetration that occurs when the ambient density is low.

There are also disadvantages to having a large B/S:

- As shown in Figure 2, with a smaller B/S, the k-factor is increased, suggesting more efficient air utilization.
- Turbulent velocity fluctuations scale with mean piston speed, which at fixed displacement scales as $(B/S)^{-2/3}$. Thus, with larger B/S, turbulent fluctuations are reduced, and turbulent mixing times are expected to increase—leading to slower combustion rates. This factor can be quite significant in spark ignition (SI) engines.[16,17] Although the impact of B/S on the duration of diesel mixing-controlled combustion has not been explored explicitly, engine speed has been shown to exert a meaningful influence on combustion rates in diesel engines.[18]
- Figure 2 also shows that with smaller bores the bowl aspect ratio (depth-to-diameter ratio) more closely approaches 1, and surface area-to-volume ratio is reduced near TDC—thereby reducing heat loss. Total heat transfer losses may actually increase, however, because near bottom dead center (BDC) the surface area-to-volume ratio is larger.[19] The expected reduction in near-TDC heat loss will be tempered by somewhat higher convective heat transfer coefficients at smaller B/S, due to higher velocity fluctuations.

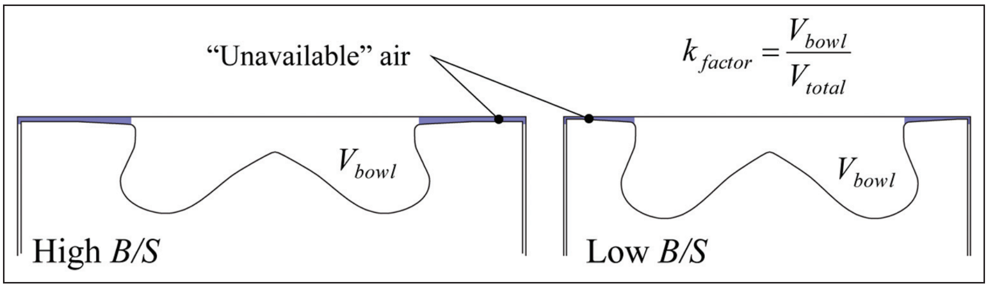

Figure 2. Illustration of the impact of bore-to-stroke ratio (at fixed r_c) on surface area-to-volume ratio and on the *k*-factor.

A very broad summary assessment of the impact of B/S on combustion system design is that a small B/S is favored for higher engine efficiency, while larger B/S favors power density.[20]

Connecting rod-to-crank radius ratio (L/R)

The ratio of the length of the connecting rod L to the crankshaft radius R ($R = S/2$) influences the kinematics of the piston motion. Although this parameter is typically selected based on friction, balancing, packaging, and manufacturing considerations, L/R can also impact the engine efficiency. With large L/R, the piston motion near TDC is slower, allowing more time for near-constant-volume heat release as well as more time for heat transfer. Depending on the relative magnitudes of the rate of heat release and the rate of heat transfer losses, engine efficiency can either increase or decrease.[21]

Combustion chamber geometry

The combustion chamber geometry plays a crucial role in generating the gas motion that supports the combustion process. It thus influences both emissions and efficiency, though generally soot emissions are most strongly impacted by geometry.[22] Due to the complexity of the interactions between chamber geometry, in-cylinder flow, and the fuels sprays, it is difficult to state general design rules that apply across the full range of light-duty combustion systems, ranging from open-bowl designs with nearly quiescent flow to re-entrant bowl designs that rely on strong flow swirl. Nevertheless, we attempt to summarize the important considerations in this section.

Axisymmetry

Axisymmetry of the combustion chamber promotes efficient air utilization and can benefit part-load emissions as well as full-load performance.[12] Along with improved breathing leading to higher power density, axisymmetry was a major factor driving the adoption of four-valve designs employing a central, vertical injector, and piston bowls centered on the cylinder axis.[23] Although non-axisymmetric bowls (e.g. square) have been released in production engines,[24] such designs have not been widely adopted.

Valve pockets in the piston top adversely affect combustion chamber symmetry. Consequently, many new engine designs feature vertical valves and no valve pockets in the piston top.[14,25–29]

Piston bowl diameter

Computational studies indicate that bowl diameter is the dominant geometric parameter impacting combustion system performance.[30] Like many geometric parameters, there is no single value that proves optimal at all engine operating conditions—although full-load operation is usually found to benefit from a wide bowl design and narrower bowls are typically preferred at lower load.[10,31,32] Typically, the maximum bowl diameter is greater than ~60% of the bore and roughly three to four times the bowl depth. Re-entrant combustion chambers are tending to become wider and shallower with each new generation.[29,33,34]

There are multiple ways in which a wider bowl diameter impacts the combustion process, most of them positive:

- Higher power density designs require higher fueling rates, which translates into larger nozzle orifices and longer liquid penetration lengths, especially at cold ambient conditions. Wider bowls provide a longer free spray length to reduce the risk of wall wetting[35] and over-penetration of fuel vapor. Higher injection pressures also support increased power density, and wider bowls are also thought to more closely match combustion chamber shape to higher injection pressures.[36–38]
- Wider bowls complement low compression ratio designs, due to the increased spray penetration associated with lower ambient density.[13]
- Wider bowls improve the *k*-factor (improve air utilization) and reduce the heat load on the piston due to a more favorable surface-to-volume ratio.[34]
- Large diameter bowls are more tolerant to advanced injection timing and help prevent oil dilution.[13]

- Wider bowls tend to increase the tolerance of the combustion system to variations in spray targeting.[12]

On the negative side, wider bowls may adversely impact the soot–NO_x trade-off at part-load[12] and low-load HC and CO emissions may increase in low compression ratio engines.[10]

Piston bowl re-entrancy

Most light-duty combustion systems feature re-entrant bowls. Re-entrancy promotes the amplification of the swirl velocity as the charge is compressed into the bowl and impacts the strength of the squish flow, thereby increasing turbulence levels and mixing rates within the bowl. This allows greater fuel injection timing retardation before fuel consumption or soot emissions suffer excessively,[9,39–41] as well as greater exhaust gas recirculation (EGR) tolerance. Moreover, re-entrant bowl shapes preserve the kinetic energy of the fuel sprays after they impact the bowl wall, re-directing them toward the cylinder center and preventing stagnation of rich mixture near the bottom of the bowl.[5] Re-entrancy further retains the swirling flow within the bowl during expansion, impeding the spread of burning fluid into the squish volume.[42] Even large-bore, quiescent combustion systems are reported to benefit from re-entrancy when highly pre-mixed operation is desired.[43]

Despite these advantages, there appears to be a trend toward lower levels of re-entrancy as existing engines are modified and updated.[35,36] Although a part of this trend is related to higher mechanical and thermal loads in today's down-sized diesel engines, the degree of re-entrancy is also selected based on combustion performance. Lower re-entrancy is reported to promote robustness to variability in spray angle and injector protrusion.[12,44] On the other hand, the higher rim temperatures of more re-entrant bowls can shorten ignition delay[41]—potentially reducing combustion noise and mitigating cold-start emissions.

Piston bowl lip shape

Figure 3 defines the features of the outer bowl geometry. A small radius at the upper lip of the piston is thought to be beneficial for soot emissions as well as for increasing the combustion system EGR tolerance[25]—likely due to turbulence generation by the reverse squish flow during expansion. The smallest radius that can be practically incorporated, however, is often limited by piston durability issues. The lower lip shape, including both the angle with which the spray impacts the bowl wall and the radius of curvature near the lip, affects the distribution of fuel within the chamber and the strength of the flow structures within the bowl.[44] Accordingly, it is not surprising that the lower lip shape impacts both smoke and fuel consumption.[9] Finally, a large outer bowl radius, resulting in a relatively small lip height, has been found to improve the soot/NO_x trade-off characteristics at both part- and full-load.[45]

"Stepped-lip" bowl geometries

Recent designs released for both light- and heavy-duty applications feature "stepped" or chamfered bowl lips, as shown in Figure 4. These steps are also sometimes referred to as "soot-in-oil" rims.[46] One objective of stepped-lip bowls is to split the fuel spray, directing a portion of it upward toward the head and the remainder downward, into the bowl. By re-directing the radial momentum of the upper portion toward the head, the spray penetration into the squish volume is impeded—resulting in less soot generated near the cylinder walls where it could find its way into the engine oil.

Another objective of stepped-lip designs is to improve air utilization by using multiple injections—thereby reducing soot emissions.[47] By targeting the upper portion of the bowl with an initial injection and the lower portion with a second injection, mixing of the second injection with O_2-depleted charge is avoided and both soot and CO emissions can be reduced.[48] Enhanced air utilization is also reputed to improve the EGR tolerance of the combustion system, and reports of increased robustness of stepped-lip designs to parameter variations, including swirl and spray angle,[49] are similarly likely to be related to improved air utilization.

Additional benefits expected from stepped-lip bowls include reduced heat loss to the cylinder liner, better cold-start performance obtained by positioning the glow-plug at the head of the sprays[49] and reduced heat loss to the piston surfaces due to an improved surface area-to-volume ratio.

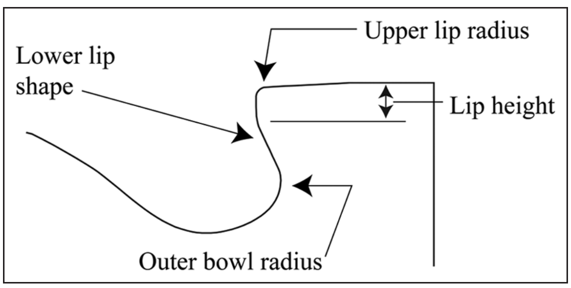

Figure 3. Features defining the bowl lip geometry.

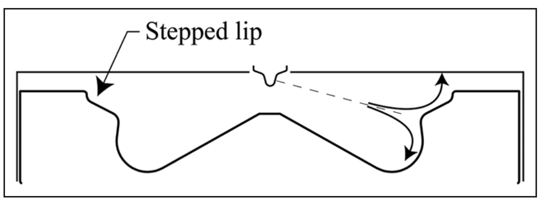

Figure 4. "Typical" stepped-lip bowl geometry.

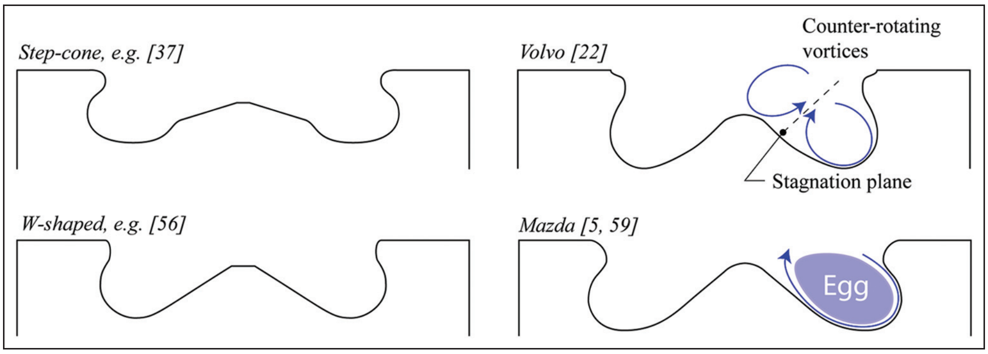

Figure 5. Examples of central pip shapes found in production engines.

Top clearance or “squish” height

The top clearance between the uppermost surface of the piston and the head surface is typically set to 0.6–0.8 mm. To maximize air utilization (a high *k*-factor), this height should be kept as small as possible. Indeed, both soot emissions[50] and low-load HC/CO emissions[51] have been found to benefit from a small squish height. However, too small a squish height can place severe demands on manufacturing tolerances, even when multiple head gasket thicknesses are available to compensate. Fuel consumption has also been shown to be minimized with a squish height between 0.6 and 0.8 mm, possibly due to increased heat losses when too small a squish height is employed.[50]

Bowl pip geometry

The bowl “pip” refers to the central protuberance in the bowl floor. The bowl center is a region where flow velocities and mixing rates are low, making optimal mixture formation difficult.[9,52] By filling this area with metal, at least three benefits are obtained:

- The fuel jets that have been deflected inward by the bowl walls will be re-directed upward and stagnation of fuel-rich mixtures in the central region of the bowl is avoided.
- Increased charge mass (volume) at the bowl periphery can be made available for mixing with the heads of the fuel sprays. This simple consideration is likely at least partly responsible for the observations of increased mixing rates in bowl designs with central pips,[9] resulting in less deterioration of fuel consumption and smoke as injection is retarded.
- The pip is thought to increase turbulence levels within the bowl,[53,54] which will also enhance mixing rates.

Typical bowl pip shapes are illustrated in Figure 5. Although the pip and lower bowl shape have a clear, load-dependent influence on emissions and performance, no profile has been found that gives universally better behavior at all loads or for all performance metrics.[55] The simple W-shaped or conical pip design has been featured in numerous engines[56] and has a high expected reliability.[12] The so-called “step-cone” pips displace a greater amount of charge from the bowl center, thereby favoring larger bowl diameters. Step-cone pips promote better full-load air utilization and tolerate more advanced part-load injection timings without excessive oil dilution.[13] However, too high a pip may interfere with air entrainment into the spray.[57]

The more complex shapes on the right-hand side of Figure 5 were designed to influence the bulk flow structures within the bowl late in the combustion process. In the Volvo design,[22] the designers sought to create a system of two counter-rotating toroidal vortices that serve to transport fuel and oxidant to a common interface (stagnation plane), where local turbulence generation rates are high. Enhancing the late-cycle mixing via this dual-vortex system led to reduced soot emissions and reduced combustion duration, thus leading to improved fuel economy. This concept has also been employed in off-highway engines,[58] enabling them to meet off-highway particulate emission regulations without aftertreatment.

In the Mazda design, the focus was on reduction in NO_x.[5,59] In this concept, the outer, “egg-shaped” vortex is thought to transport hot combustion products to the cylinder center where they mix rapidly with cooler surrounding charge, thereby quenching thermal NO_x production. The thermal efficiency is also increased due to a shortening of the combustion duration.[59] Toyota has also recognized the importance of the outer vortex on promoting soot oxidation.[60]

Matching geometry, flow, and fuel injection parameters

The geometry of the combustion chamber, the swirling flow, and the fuel sprays all mutually interact, and it is impossible to state general design rules for matching these characteristics under all relevant operating conditions. Consequently, we again endeavor to summarize only the most important considerations.

Swirl level

Flow swirl is employed in light-duty engines with the objective of increasing and sustaining adequate mixing rates throughout the combustion process. To accomplish this, swirl is used to promote the development of large-scale flow structures that disperse the burning fuel throughout the bowl and to generate small-scale turbulence that completes molecular level fuel–air mixing on sufficiently short time scales. The increased mixing rates associated with flow swirl generally result in both lower soot emissions and improved fuel consumption.[25]

A number of metrics are used to quantify swirl.[61] However, the most common is the swirl ratio $R_s = \omega_s/2\pi N$. The swirl ratio represents the ratio of the characteristic angular velocity ω_s of the in-cylinder flow to the angular velocity of the crankshaft $2\pi N$. Light-duty engines typically operate with either a fixed swirl ratio of between roughly 2 and 2.5, or have a swirl control valve that restricts one port and allows the flow swirl to vary in the range from approximately 1 to 4. The lower swirl is achieved when both ports are fully open and is appropriate for high engine speeds and loads. At lower speeds and loads, when flow restrictions are less important, the higher swirl levels are used to promote mixing and soot oxidation.[10,38]

Flow swirl also has drawbacks:

- Ports designed to generate swirl typically have greater flow losses and the ensuing reduction in volumetric efficiency results in a loss in power density. Recent engine designs make greater use of chamfers in the cylinder head, which help increase the swirl generated at low valve lifts and reduce interference between the flows from the two intake ports.[14,37,38,62,63] Consequently, flow loss considerations are becoming less important.
- Flow swirl can adversely affect fuel economy despite more rapid combustion due to increased heat losses. Increased heat losses also deteriorate cold-start performance.
- Excessive swirl can increase entrainment and hence reduce the penetration of the fuel jets, thereby impeding air utilization. Although near-nozzle entrainment is not expected to be impacted by swirl due to the low local tangential velocities and high injection velocities,[64] high swirl levels have been shown to impede penetration at part load[65] and can also significantly deflect the fuel sprays even at high load.[66]
- High swirl at low loads and high EGR rates can increase HC and CO emissions.[10]

Despite these drawbacks, the benefits of swirl are such that, to the authors' knowledge, no modern automotive-scale direct injection (DI) engines have been introduced that do not employ some level of flow swirl.

The optimal swirl level depends on the number of fuel injector nozzle holes and is usually lower for a greater number of holes.[12,14,67,68] With a high enough hole count, the full-load soot is not overly sensitive to swirl and it may be possible to use a single swirl level optimized under part-load conditions without significantly deteriorating full-load behavior.[14] Bowl diameter also impacts the optimum swirl level. As the swirling flow is compressed into the piston bowl, conservation of angular momentum amplifies its rotational speed, and combustion systems have been thought to optimize at amplified "bowl" swirl ratios near 5. Because the swirl amplification is less for larger bowls, they can be found to require higher swirl levels.[35] Note, however, that in new low compression ratio designs, where part-load soot emissions are mitigated primarily through increased fuel–air premixing rather than increased late-cycle oxidation, high bowl swirl may not be necessary. In this case, reduced swirl levels and large bowl diameters can be jointly employed to reduce heat losses and thereby improve both cold-start behavior and fuel consumption.[8]

Fuel injection parameters

Selecting the fuel injection equipment characteristics to meet the desired power density and to match the piston bowl geometry and the swirl level is a critical aspect of diesel combustion system design. As a first step, the fuel injector flow capacity and the maximum injection pressure must be selected to provide sufficient fuel to meet the rated power requirements. The full-load fuel delivery must take place within a 30°–35° window (about 1 ms in duration at rated speed). The lowest flow nozzle that meets the full-load requirements is usually selected, since lower flow nozzles generally result in an improved soot/NO_x trade-off.[12] The fuel nozzle flow capacity is determined by the number of nozzle holes, their diameter, and the nozzle hole discharge coefficient—which is ~0.85 for modern, conical nozzles with rounded nozzle hole inlets. Thus, with smaller nozzle holes, a larger number of holes are required in order to meet rated power targets. Rapid injector needle opening and closing characteristics can also significantly impact the amount of fuel delivered for a fixed nozzle flow capacity and overall injection duration. Here, hydraulically actuated injector designs with control chambers close to the nozzle tip and designs which are directly driven by piezoelectric actuators can be helpful.

In addition to the required flow capacity, the nozzle hole diameter and number of holes also influence how the fuel jets interact with both the flow and the piston. As noted above, the optimal number of holes is coupled to the swirl ratio. Although a greater number of holes generally optimizes at a lower swirl level, the optimum achieved may be higher for too many or too few holes, or have less tolerance to swirl variation.[12,14,67] The optimal number of holes also depends on the specific engine operating condition, compression ratio,[8,10] and the metric used to define the optimum. Typically, the higher swirl needed when too few holes are used will adversely affect part-load fuel consumption, while soot emissions and fuel consumption at higher loads deteriorate with too many nozzle holes. Historically, the

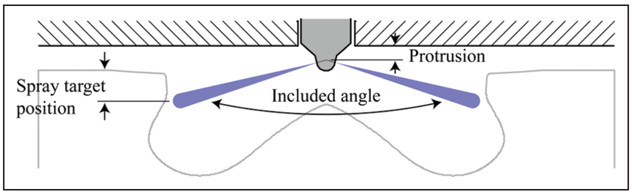

Figure 6. Definitions of variables impacting the interaction of the sprays with the piston.

number of nozzle holes employed has steadily increased; seven to eight holes are now typical for new engine designs.

The spray interaction with the piston is affected by the spray target position, which is determined by the injector nozzle tip protrusion and the included spray angle or umbrella angle—see Figure 6. Optimal spray targeting, like the optimal nozzle hole size and number, will depend on numerous other combustion system parameters (bowl geometry, swirl ratio, etc.) as well as the engine operating condition. However, a few generalizations can be made:

- At higher loads, spray targeting chiefly impacts soot emissions. As described earlier, the shape of the bowl lip in conjunction with targeting location must create an appropriate fuel split between the squish volume and the bowl and avoid stagnation of fuel-rich mixture near the bowl walls or floor by producing sufficiently strong flow structures. When the load is increased, the optimal targeting that satisfies these conditions moves lower in the bowl.[22]
- Targeting the spray lower in the bowl reduces light-load HC and CO emissions,[13,22,51] indicating that over-lean mixture formed from fuel injected directly into the squish volume is an important source of these emissions.
- Spray targeting has been found to impact secondary liquid atomization by the piston, and appropriate targeting has potential for enhancing cold-start behavior.[69]

Investigations of spray targeting have generally only varied the included angle of the spray, and the impact of nozzle tip protrusion is not typically addressed directly. The spray included angle has been identified as the dominant nozzle geometry–related factor impacting NO_x, soot, and fuel consumption[30] in heavy-duty engines. In light-duty engines, however, the included angle has been found to be of little importance at mid-to-high loads, provided the same spray targeting position is employed by simultaneously adjusting the tip protrusion.[22] Generally, with typical re-entrant bowls and widely varying spray targeting position, wide included angles are found to give improved part-load soot emissions.[13,70] Finally, note that increased protrusion will tend to increase the nozzle tip temperature, which is known to increase deposit formations within the nozzle.[71]

Summary

The various considerations dictating the selection of combustion system design parameters relevant to light-duty diesel engines have been reviewed. It is the hope of the authors that by collecting and condensing this information we have provided a resource that new engineers can use to quickly gain an understanding of the various trade-offs involved and to identify those parameters that might bear careful scrutiny as they work to improve specific aspects of engine performance. This article is a highly condensed version of a more complete work, to which we refer the reader interested in linking these more practical aspects of combustion system design to the fundamental physics governing the combustion process.[72]

Declaration of conflicting interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding

This work was performed at the Department of Energy Sciences of Lund University in Lund, Sweden and at the Combustion Research Facility, Sandia National Laboratories in Livermore, California with support provided by the Swedish Energy Agency, the United States Department of Energy (Office of Vehicle Technologies) and by General Motors Corporation (agreement # FI083070326). Sandia is a multi-program laboratory operated by Sandia Corporation, a Lockheed Martin Company, for the United States Department of Energy s National Nuclear Security Administration under contract DE-AC04-94AL85000.

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