

Chapter 2

Basic Concepts on GDI Systems

Abstract Gasoline Direct Engines offer many advantages as compared to PFI engines, as regard efficiency and specific power. To fully exploit this potential a particular attention must be paid to the in-cylinder formation process of air/fuel mixture. More demanding performance is required to the combustion system, since injectors must provide a fine fuel atomization in considerably short time, achieving a spray pattern able to interact with in-cylinder air motion and piston top surface. This is made possible through the Common Rail technology allowing an injection pressure one order of magnitude higher as compared with that of conventional PFI engines. Fuel economy can be obtained regulating load by mixture leaning, minimizing throttle usage at low loads where pumping losses are more significant, and requiring charge stratification for a stable ignition and combustion. Charge stratification can be pursued based mainly on the sole action of the fuel spray or on its interaction with a specially shaped surface on piston top or with the air bulk motion. Depending on the modality of stratification attainment, different combustion systems can be considered. The injector design has in turn a key role being the final element of fuel metering required to the desired spray pattern, injected fuel mass per injection event, resistance to thermal stress and deposits. Injector housing and orientation with respect to the combustion chamber has to be carefully chosen, exploiting in this regard the indications of computational fluid dynamics (CFD), provided by 3D simulations. Some fundamental scheme is provided for the whole high pressure fuel delivery plant, as employed in current vehicles equipped with GDI spark ignition engines.

2.1 Mixture Preparation: Gasoline Port Fuel Injection Versus Direct Injection

A short comparison between gasoline Port Fuel Injection (PFI) and direct injection (DI) can be made to highlight their differences, advantages, and drawbacks as regard mixture preparation and its consequences on performances, fuel economy, and emission. In PFI engines with an injector for each cylinder, the injection system is quite simple and without demanding requirements. Fuel is fed to the injectors by a common manifold at a pressure usually in the range of 3–5 bar, maintained fairly constant thanks to a pneumatic regulator driven by the absolute pressure in the inlet manifold. Depending on injector features, such a pressure allows fuel atomization with a droplet Sauter mean diameter (SMD) in the range of 120–200 micrometer. The SMD is defined as the diameter of the droplet having the same surface to volume ratio as that of the overall spray. Assuming x as the diameter of a single droplet, and being $f(x)$ the corresponding density probability function, SMD is defined as

$$\text{SMD} = \frac{\int_{x_m}^{x_M} x^3 f(x) dx}{\int_{x_m}^{x_M} x^2 f(x) dx},$$

where x_M and x_m are the maximum and the minimum droplet diameter, respectively. Fuel delivery is performed out of the cylinder, with the scope of obtaining a fairly homogeneous charge, and the spray produced by the injector is directed onto the inner wall of the inlet duct and the back of the inlet valve, forming a fuel puddle, and mixture preparation occurs by fuel evaporation from the puddle and the suspended droplets. Even if the engine is fully warmed up, depending on the amount of injected fuel, it is unlikely that the time interval prior inlet valve opening is enough to complete mixture preparation out of the cylinder. Especially if temperatures of duct wall and inlet valve and available time are not adequate for complete external evaporation (for example at cranking and cold start), a portion of fuel is inducted into the cylinder as liquid rivulet and droplets, while some residual remains on the inner surface of the inlet duct. This means that the fuel quantity inducted in the cylinder is not that injected by the injector, and a delay occurs in fuel delivery. Several consequences follow: imprecise fuel metering; time response worsening in transient maneuvers; incomplete combustion and soot formation due to inlet valve seat wetting and incomplete in-cylinder droplet evaporation. In this regard, cold cranking is a critical condition where low temperature and low turbulence hinder evaporation; it may be necessary to deliver a fuel amount up to 4–5 times the stoichiometric one, with delays in engine start and high level of engine out emission as unburned hydrocarbons (UHC) and carbon monoxide (CO). In gasoline DI (GDI) engines air only is inducted through the inlet valve, and mixture is prepared delivering fuel in the cylinder. As already pointed out, this solution is able to promote charge cooling thanks to heat absorption by fuel evaporation, leading to a better volumetric efficiency and a lower propensity to knock and thus to the possibility of higher compression ratio and higher efficiency. Furthermore, a more precise fuel metering is obtainable, giving the possibility to operate with lean mixture and to

control load by means of mixture strength variation, avoiding the pumping losses due to throttling at low loads, with beneficial effect on efficiency. On the other hand, there is the need to complete mixture preparation in the time interval between inlet valve opening and ignition triggering by spark, which is much lower than the analogous time allowed in PFI systems. For this reason, more demanding requirements have to be satisfied by (GDI) systems. Actually, fuel evaporation in so a short time requires a fine atomization, featured by a droplet SMD not higher than $20\text{ }\mu\text{m}$. This is the reason why specifically designed injectors are needed with an injection pressure one order of magnitude higher than that of PFI systems. This is accomplished in current (GDI) engines using common rail (CR) technology which is able to provide injection pressures well beyond 100 bar. However, despite the finer atomization, the possibility of surfaces wetting (e.g. cylinder wall and piston crown) cannot be avoided; actually sometimes it is wanted in conjunction with some interactions with the in-cylinder air flow to promote charge stratification. The possibility of more precise fuel metering offers to (GDI) engines the advantage of low mixture enrichment at cold cranking, with consequent lower UHC emission and a more rapid starting. Basically, depending on the working condition, air/fuel mixture in (GDI) engines can be homogeneous (that is with an fairly uniform AFR in the combustion chamber volume) or stratified (that is with AFR space gradients, richer in the volume near the spark plug and progressively leaner as distance from spark plug increases), a result attained acting on injection timing. Homogeneous charge can be obtained by an early injection, actuated at the beginning of the air induction phase, so as to allow an adequate evaporation and mixing of fuel in conjunction with the air motion promoted by the inlet ducts geometries. Generally such a condition is promoted at high load operation. Charge stratification can be attained by late injection, actuated during the upward piston stroke of compression, metering the fuel to obtain an overall lean AFR ratio. Charge leaning, if not too high, allows load control without throttling, with beneficial effect on fuel economy. Stratification is motivated by the need to have a richer mixture near the spark plug gap for a stable ignition, otherwise misfiring could occur.

2.2 GDI Combustion Systems

The attainment of a desired stratified charge have to take into account the reciprocal position of spark gap and injector. Generally, a central position of the spark plug in the combustion chamber is preferred: this choice is motivated by the need to reduce the probability of knocking combustion, occurring when the unburned mixture furthest from spark gap reaches auto-ignition before the arrival of the flame front. A central position of the spark plug allows a symmetrical propagation of the flame front initiated by the spark, performing a shorter path to extend combustion to the whole unburned mixture before auto-ignition occurs. Twin spark engines are equipped with a second spark plug in peripheral position, giving a further support to enhance knock prevention (see also Fig. 2.2).

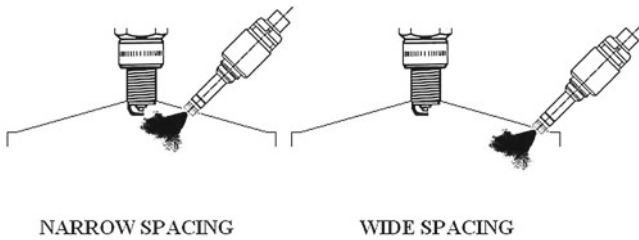
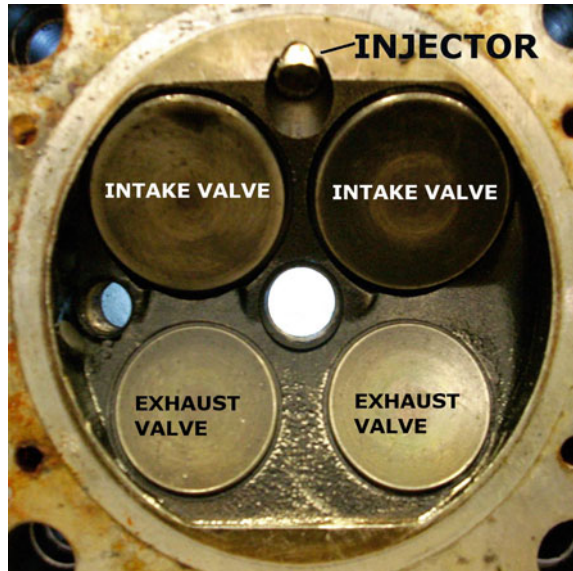


Fig. 2.1 Scheme of the different locations of injector

Fig. 2.2 Example of injector location between intake valves, left and central holes are for the housing of spark plugs (Twin-Spark engine)



Injector can be located in proximity of spark gap (narrow spacing, see Fig. 2.1); this choice is more favorable for stratification attainment since this result can be pursued more easily without involving the motion of the air charge bulk. The main disadvantage of this solution is the higher probability of spark plug fouling caused by electrodes fuel wetting; furthermore, fuel injector location in central position or with a small offset has to be carefully chosen considering the higher thermal stresses due to exhaust valves vicinity. As alternative to the previous solution, fuel injector can be located away from the spark gap (wide spacing, see Fig. 2.1); in this case the sprayed fuel plume has to perform a longer path to reach the gap. This results in a longer time interval for stratification attainment and in the requirement of proper interaction with the air charge bulk and the moving surface of the piston. Generally, in a four valves engine head the position is chosen between the inlet valves, where injector tip overheating is avoided and a more effective interaction is possible between the fuel spray and the air flow incoming from inlet ducts (Fig. 2.2).

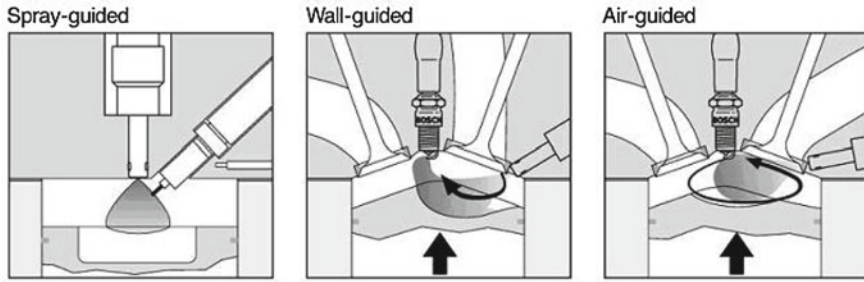


Fig. 2.3 Scheme of the different combustion systems (taken from [1])

In this regard, the correct matching between injector features and in-cylinder air motion is fundamental to attain the desired stratification, with the AFR required for ignition in the volume around the spark plug gap, and at the time of spark generation. To comply with these demanding requirements based on experiments only would be an expensive and time-consuming task; therefore, computational fluid dynamics (CFD) simulations, both 1D and 3D, are a valuable aid to search more quickly the optimal parameters for the design of combustion chamber, inlet and exhaust ducts, and injector. Of course, this does not eliminate the need of final experimental validation of the results produced by simulation, usually referred to specific operating conditions. This interaction between simulation and experiments is useful to setup the best-compromise solutions for the whole operating range of an engine, certainly allowing a significant reduction of development costs and time to market.

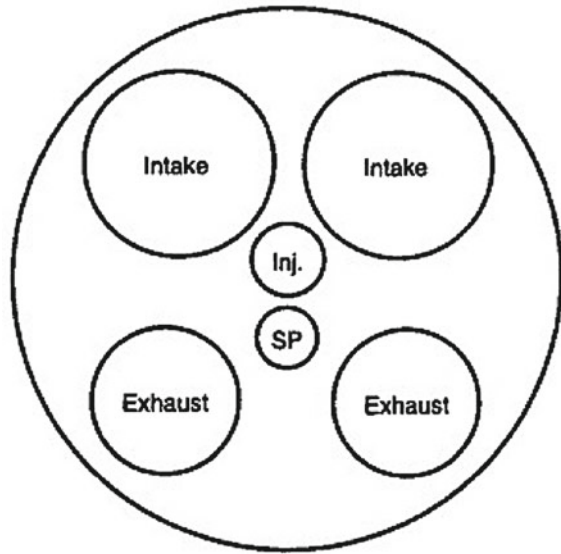
The combustion systems can be distinguished depending on the way used to obtain charge stratification. Thus, the following classification is possible (Fig. 2.3):

- Spray guided systems, where the stratification process relies mainly on the features of the spray and its dynamics, as regard fuel atomization, droplet distribution, and overall geometry;
- Wall guided systems, where the stratification process is based mostly on the interaction between the fuel spray and the surface of a proper cavity on the top of piston;
- Air guided systems, where stratification is obtained mainly by means of the interaction between the fuel spray and the motion of the air charge inducted in the cylinder.

Actually, a real engine can be classified according to the features of one of the previous system, but this does not exclude the possibility that some of the features of the other ones can be recognized.

In the following, a brief description of the main features of these system will be provided.

Fig. 2.4 Example of injector and spark plug location in a 4 valve spray guided GDI spark ignition engine (taken from [2])



2.2.1 Spray Guided Systems

As previously outlined, in spray guided systems charge stratification results mainly from the spray dynamics, with minor contribution from the interaction with the bulk air charge motion and the surface of the piston cavity. To this aim, injector has to be located in a position close to the center of the cylinder head and near the spark plug (Fig. 2.4), giving the advantage of a symmetrical distribution of fuel in the combustion chamber with a better utilization of air charge.

There are several drawbacks of this system; one of them is the higher tendency to spark plug fouling and soot formation, as well as the sensitivity to spray geometry variations, as consequence of off-design operation of injectors resulting, for example, from deposits or CR pressure deviations with respect to the value required for the desired performance. These spray variations can lead to higher Indicated Mean Effective Pressure (IMEP) coefficient of variation and misfiring. In relation to the central position of the injector, some problems arise for unwanted fuel impingement and consequent wetting of piston head, occurring with late injection when piston approaches top dead center in the upward stroke. A way to mitigate this drawback is a combustion chamber design providing a bowl in the piston top (Fig. 2.5) or a dome in the cylinder head (Fig. 2.6), so as to increase the path of the fuel plume toward the piston surface.

Another issue in spray guided systems, again related to spark plug proximity, is represented by housing difficulties in the cylinder head, forcing designers to reduce the diameter of inlet valves, with consequent penalization of air charge intake. The problems related to the sensitivity to spray variation can be relieved by the adoption of air assisted injectors; in this case, with the injector aligned with cylinder axis, fuel distribution is obtained by pneumatic atomization. This solution reduces

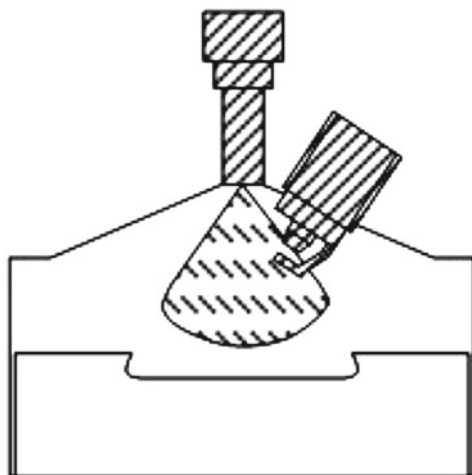


Fig. 2.5 Example of injector and spark plug location in a 4 valve spray guided GDI spark ignition engine (taken from [3])

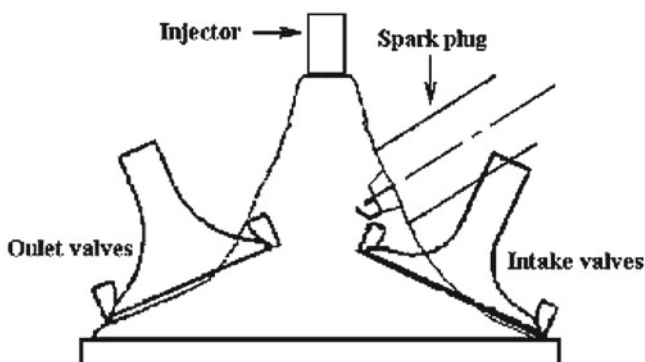


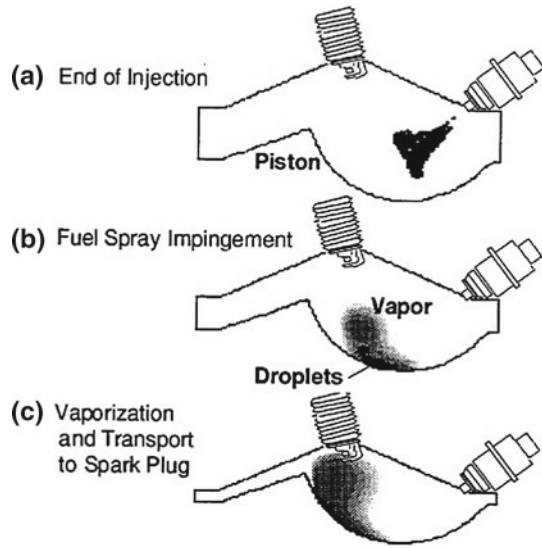
Fig. 2.6 Example of injector and spark plug location in a 4 valve spray guided GDI spark ignition engine (taken from [4])

the possibility of piston surface wetting, requires a lower fuel pressure difference, allowing less demanding features for the fuel pressurization system, but involves other complications for the presence of an air compressor and a storage plenum.

2.2.2 Wall Guided Systems

A stable charge stratification can be pursued maintaining a short distance between injector tip and spark gap, as well as a short time interval from injection start

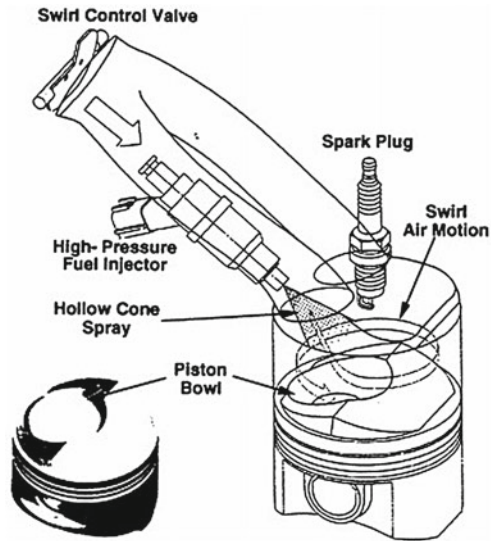
Fig. 2.7 Scheme of the operating phases of a wall guided combustion system (taken from [5]).



and spark ignition; but the related short time available for mixture preparation has negative consequences as soot and UHC production. At some cost on stratified charge stability, the process of mixture preparation can be improved allowing a greater time interval between injection and ignition events, and increasing the distance between the injector tip and the spark plug electrode. Wall guided systems adopt this approach locating the spark plug in central position while the injector is housed in side position (Fig. 2.7). At the same time, piston top is specially shaped or is provided with a bowl or cavity facing the injector: the injection event promotes an air flow, directed toward the piston cavity, inducing the spray droplets so as to avoid significant impingement and wetting, and guiding them during their evaporation along the cavity surface to reach the spark gap with the AFR conditions suitable for ignition.

The quality of the resulting charge stratification depends on the design of the whole system, and in this regard CFD simulation are an essential tool to find the optimal solution for piston top or piston bowl geometry, injector position, and spray features. Most of the current wall guided systems locate the fuel injector on the side of intake valves; this choice has the benefit of maintaining injector tip temperature at safe level, and a better mixing the incoming air flow into the fuel spray, improving droplet evaporation, at early injection. If an unintended cylinder wall wetting occurs at early injection, an increase in UHC emission follows because of incomplete evaporation and mixing with air and of adsorption and subsequent desorption of the fuel that, after being trapped in the piston top land and inter-ring crevices, is dissolved in oil with consequent dilution and loss of lubricant properties. Such a condition occurs for example, at high load and speed when CR mean pressure is increased to meter a higher fuel amount, and possible pressure pulsations in the CR systems make the injector operate with spray penetration exceeding the design limits. This underline

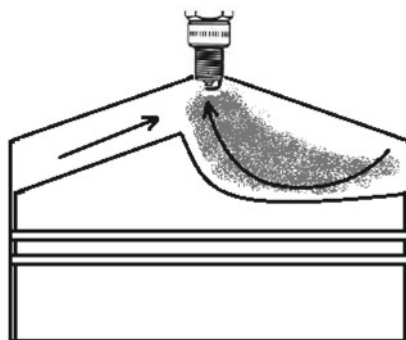
Fig. 2.8 Example of swirl air motion coupled with a bowl in the piston (taken from [6])



the importance of a control able to regulate CR pressure in all the possible engine operating conditions. The design of piston bowl is crucial to attain the desired charge stratification, however, in all cases a compromise solution has to be adopted. For example, the choice of a deep bowl allows to extend the operating range where engine is able to work with stratified mixture, but this is counterbalanced by several drawbacks. First, with respect to a flat piston an actual homogeneous charge at wide open throttle is more difficult to be obtained with a deep bowl shaped piston, resulting in combustion worsening and reduction of full load torque. Second, the more complex piston shape, apart from higher manufacturing difficulties and costs, leads to higher heat losses from burned gas affecting negatively engine efficiency. In-cylinder air motion has a significant role in the guiding of the fuel cloud produced by the injector along the piston cavity walls toward the spark gap, as well as for the evaporation of the fuel droplets impinging on cavity surfaces. Basically, in-cylinder air motion can be of swirl type, featured by a rotation around an axis parallel to that of cylinder, or of tumble type, where the air flows rotate around an axis perpendicular to that of cylinder.

Swirl air motion is chosen when piston top design provides a bowl or an open chamber (Fig. 2.8). Reverse tumble motion, where the air flow rotates from the bottom to the top, is generally combined with a piston cavity shaped to receive the fuel plume during the upward piston stroke, flowing air and redirecting the resulting mixture toward the spark plug gap housed in central position (Fig. 2.9). The piston top side opposite to the cavity can be shaped in turn to promote a squish flow at the end of compression stroke, enhancing the tumble motion after ignition with effect on flame front propagation. Injector location and axis inclination is another crucial parameter to be considered, requiring a careful analysis of the whole injection process and its

Fig. 2.9 Example of tumble air motion enhancement by a shaped cavity in the piston



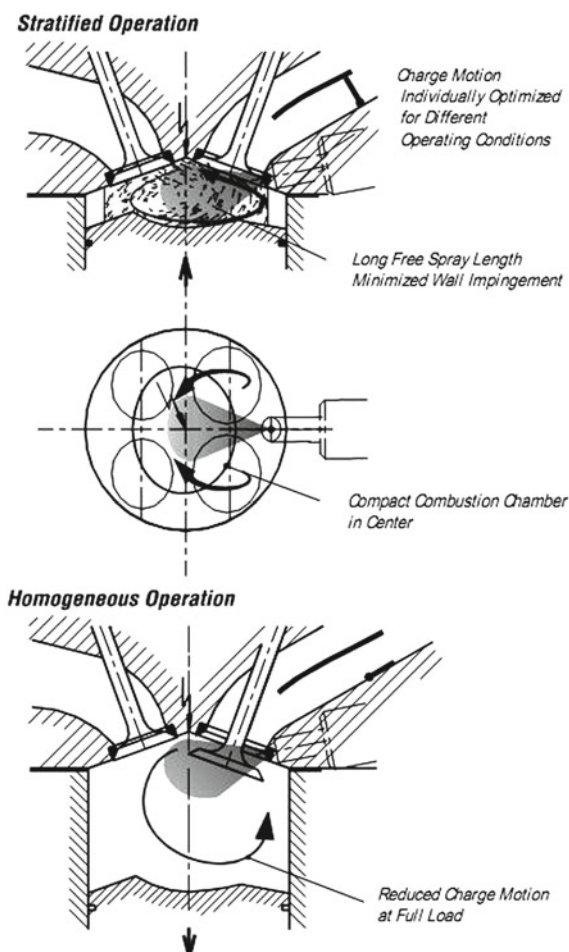
subsequent interaction with the air flow and the geometrical features of the combustion chamber. In this regard, CFD simulation is a fundamental and indispensable aid. Provided an injector housing on the intake valve side, a low inclination of its injection axis can often lead to spray impingement on the opposite cylinder wall; further on, the outer region of the spray can impact against the intake valves during early injection. Predictable consequences are fuel deposit at piston top land crevices and valve seats, leading to UHC emission increase and soot production. As countermeasures there can be chosen an offset spray or a higher injector axis inclination; the latter solution has the collateral benefit of a good interaction between the spray and the piston cavity independently on piston position, with the possibility of stratified charge operation in a wider speed range. However, as the injector axis inclination is increased towards the vertical position, a significant worsening of mixture homogeneity can occur with penalization of full load torque and specific consumption. Special attention has to be paid to house injector tip neither projected in the combustion chamber, with consequent risk of intake valve wetting and deposits formation, nor in a recess though small, which could collect fuel droplets at spray periphery again with wetting and carbon deposit formation, but flush with the surface of combustion chamber.

2.2.3 Air Guided Systems

In air guided systems charge stratification is pursued based on the mutual action between the injected spray and the bulk air charge motion. Spark plug is still in central position while the injector is located in side position; air swirl or tumble motion is generated by means of the orientation and geometry of the inlet ports and ducts. Sometimes the desired motion can be enhanced by proper devices, like baffles, in the inlet ducts (Fig. 2.10).

In systems with two inlet valves, if allowed by the valve train mechanism, swirl could be induced by deactivation of one of them. Since charge stratification depends mainly on air motion in the cylinder and not on spray impingement, there is the

Fig. 2.10 Scheme of air guided system based on tumble motion. Notice the tumble control valve housed in the inlet duct (taken from [7])



advantage of avoiding UHC emissions related to fuel wetting of piston surfaces. This is counterbalanced by a higher sensitivity of combustion to changes of the air flow field, requiring a careful design of the combustion system considering the interaction between combustion chamber shape, inlet and exhaust geometry, and injector features. Despite in principle air guided systems do not require a special shape on piston top, actually some modification are made to ensure the attainment of the desired air motion, and this causes some difficulties in maintaining homogeneous charge at wide open throttle. As consequence, these systems too suffer of penalization of full load performance in terms of torque and specific fuel consumption.

2.3 Requirements of a GDI System

The first experiences of gasoline direct injection used devices basically derived from compression ignition engines injection systems. Unfortunately, a direct conversion of diesel engine systems to direct injection spark ignition engines is not possible because of the basic differences in the combustion process. While in diesel engines combustion is spontaneously initiated by compression in several zones of the chamber, in spark ignition engines combustion has to be triggered in a precise point corresponding to spark gap. This is a stringent constraint if unthrottled engine regulation is pursued by charge stratification, requiring both a spatial and temporal optimization of the interaction between air and injected fuel. Another demanding requirement is the capability of performing both early and late injection with proper atomization: early injection, aimed to an homogeneous charge, needs a more dispersed fuel spray, while late injection, aimed to stratification, requires a more narrow fuel spray. In addition, in-cylinder pressure conditions for early and late injection are quite different, and this results in different requirements for CR fuel pressure. As consequence, research and development in this field have been addressed to obtain specific features on GDI systems. The injector is the most important element of the whole system, having a key role in the mixture preparation process. Apart from the ability to give different spray features depending on engine operating conditions, it must guarantee sealing during combustion; resistance to temperature and leakage; resistance to deposit formation. An important parameter of injector design is the sac volume, which is the fuel volume downstream of the tip sealing element, residual of the previous injection. Being not subjected to the CR high pressure, it is ejected with poor atomization and hampers the correct atomization of the subsequent fuel volume; therefore, it has to be reduced as much as possible. The design of the injector pintle, as well as of its actuation mechanism and control, has to be aimed to minimize bounce phenomena, especially at closing event, which can lead to uncontrolled and unintended secondary injections producing larger droplets responsible of higher UHC and particulate emission. One of the basic parameters of GDI injector is the static flow capacity, corresponding to the volume flow rate resulting at pintle fully open and rated CR pressure condition. Based on this flow capacity, fuel is metered in the same way as in PFI systems, that is by injection duration which can range from 1 to 6 milliseconds. This is accomplished using a square wave voltage signal, defined as pulsewidth (PW) signal, generated by an ECU, and routed to an injector driver module. This module turns the PW signal into a current signal which is properly shaped to attain a fast opening and the desired hold-on condition. At a fixed CR pressure, a linear relationship exists between the duration of the input PW signal and the fuel amount metered per injection event, representing the injector characteristic. This linear function occurs over a range between a minimum and a maximum PW value, depending on injector design, and constitutes the operating range of the injector. By setting a different CR pressure, a different linear characteristic can be obtained; the higher the pressure the higher the characteristic slope as it will be shown later on (see Fig. 3.12a in Sect. 3.3).

By using a CR pressure regulator properly controlled by the EMS, it is possible therefore to have at disposal a wider dynamic range of the injector; for example, by setting a lower CR pressure it is possible to meter small fuel quantities with short injection duration using still a linear dependence between injected fuel mass and injection time. CR pressure level is set according to the desired features of the fuel spray, as penetration length, droplet SMD and distribution, cone angle, and so on. Fuel mass, ranging from some units to several tens of milligrams, is metered depending on the injection duration and on the difference between CR pressure and in-cylinder pressure. Any variation of this difference, which cannot be evaluated and taken into account in absence of an in-cylinder pressure measurement, leads unavoidably to metering errors. Further on, an unwanted change in CR pressure results in a transient change of the working characteristic of the injector, which in turn induces metering errors and alteration of the spray features from the desired ones. In this regard CR pressure pulsations, due to cyclic and sequential opening and closing of injectors as well as to high pressure pump operation, can contribute to combustion irregularities. The injection process is related to the fuel pressure in the injector, which in turn depends on CR pressure. The choice of its level is made combining the requirements of mixture formation, design features of the injector, parasitic load of high pressure pump, overall noise of the injection system. Often a compromise solution has to be searched. For example, a higher injection pressure on one hand can increase fuel atomization, on the other hand can lead to excessive spray penetration with unintended impingement on cylinder wall surface. On the contrary, for some type of injector, a higher injection pressure produces smaller droplets which spread and slow down too rapidly with reduction of spray penetration. CR pressure has a crucial role in cranking operation mode too; since the high pressure pump needs several crankshaft revolution to develop the steady rated pressure, during this transient condition the lower injection pressure of the tank fuel pump is available to meter the required fuel amount. This result in worsening of fuel atomization with an increase of droplet Sauter mean diameter up to $100\text{ }\mu\text{m}$; injection duration increases as well and at cold start condition, when a higher fuel mass is required, it can exceed the maximum time allowable in relation to the intake phase. In order to comply with the requirement of a quick engine starting and low cold start emission, especially as regards UHC, injector must be designed to take count of these unfavorable conditions. As already mentioned, the injector has to provide a fuel atomization with a SMD not higher than $20\text{ }\mu\text{m}$. Related to gasoline physical properties, this leads to the necessity of providing the CR with pressure levels up to 120 to 130 bar depending on the in-cylinder backpressure. Besides the effect on fuel atomization, the high pressure level, as compared to PFI systems, is motivated by the requirement of metering the needed fuel mass in a very short time interval at high speed and load. However, accurate fuel metering with a high injection rate is important too for stratified charge at low load: in those operating conditions the injector may work outside of the range of the linear relationship between injection duration and injected fuel mass, that is beneath the minimum value. In this case, when injection pulse duration is shorter than the minimum of the linear range, a PW lookup table is provided by a specific calibration. Another crucial injector parameter is the resistance to the formation

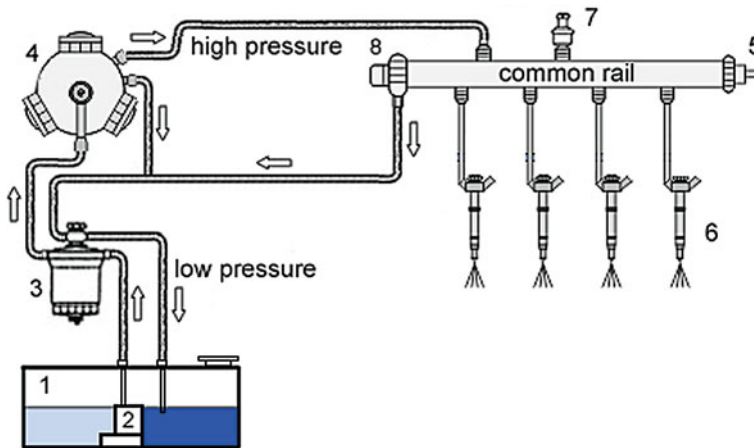


Fig. 2.11 Common rail injection system description: 1 fuel tank; 2 low pressure pump; 3 fuel filter; 4 high pressure pump; 5 common manifold; 6 electro-injector; 7 pressure sensor; 8 pressure regulation electro-valve

of deposits as related to drippage, that is the tendency to the accumulation of fuel resulting from defective closure or fuel droplets remained near the tip.

2.4 Description of the Common Rail Injection System

The main objective of a common rail system is to supply the electro-injectors with high pressure fuel independently by the amount to be injected. Therefore, the main benefit is that to decouple the regulation of the pump by the functioning of the injectors unlike the traditional injection systems where the mechanical pump generates a pressure that depends on the amount of fuel to spray.

A schematization of a common rail plant for spark ignition engines is shown in Fig. 2.11. The injection system is mainly composed by two separated sections: a low pressure circuit, consisting of a fuel tank, a fuel feed pump with a downstream filter, a low pressure pipe, and high pressure circuit formed by a high pressure pump, a high pressure line with a pressure sensor, a pressure regulator valve, a flow stopper, and the injectors. The low pressure electro-pump (2) forces the fuel from the tank (1) toward the high pressure mechanical pump (4) crossing the filter (3), which cleans the fuel from impurity. The second pump compresses the fuel and sends it into the common manifold (5) (named common-rail) equipped with the electro-injectors (6). The manifold is designed in order to hold low the pressure oscillations in the fuel due to the pump and the intermittent working of the injectors. Finally, the pressure in the manifold is regulated through the sensor (7) and the electro-valve (8) that flows the excess of fuel back into the tank.

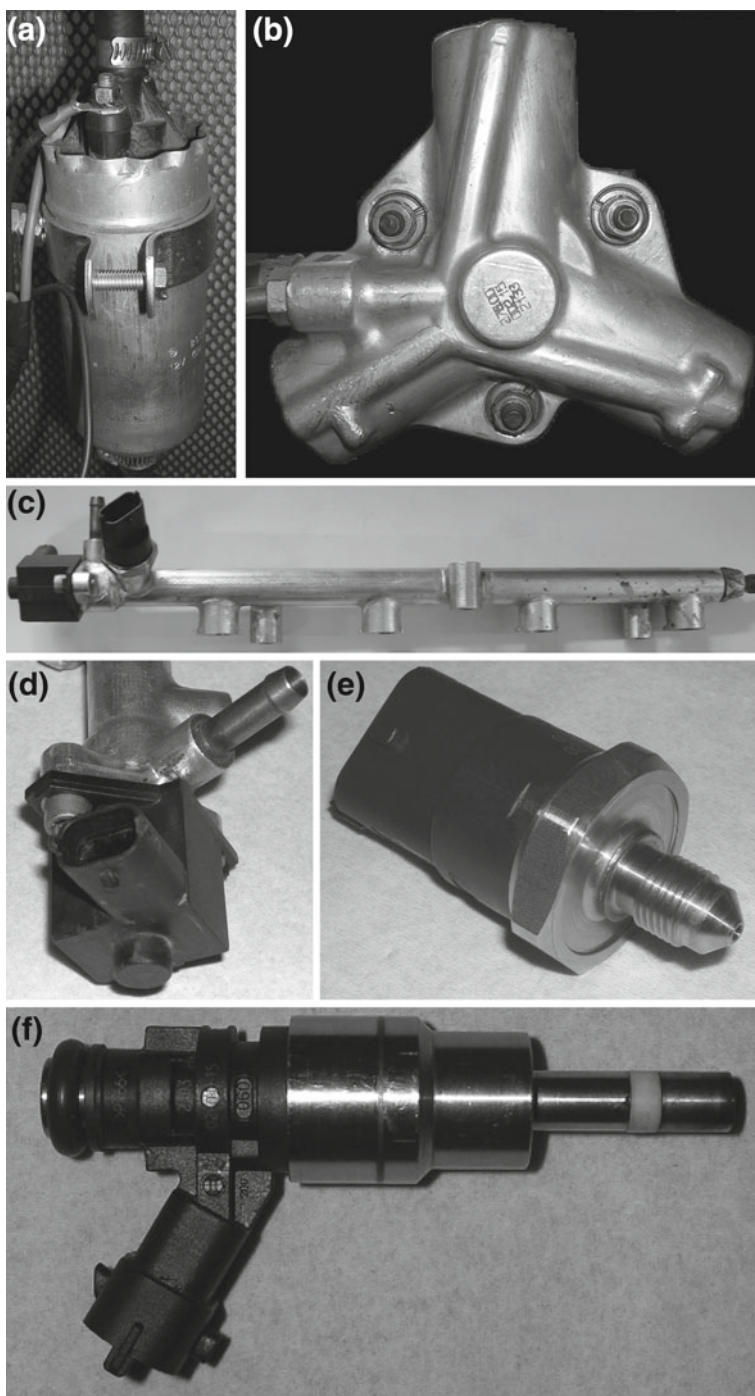


Fig. 2.12 Pictures of components common rail injection system mounted on a 2.0-liters GDI engine: **a** low pressure electrical pump; **b** high pressure mechanical pump; **c** common fuel rail; **d** electrovalve; **e** high pressure sensor; **f** electro-injector for direct injection of fuel

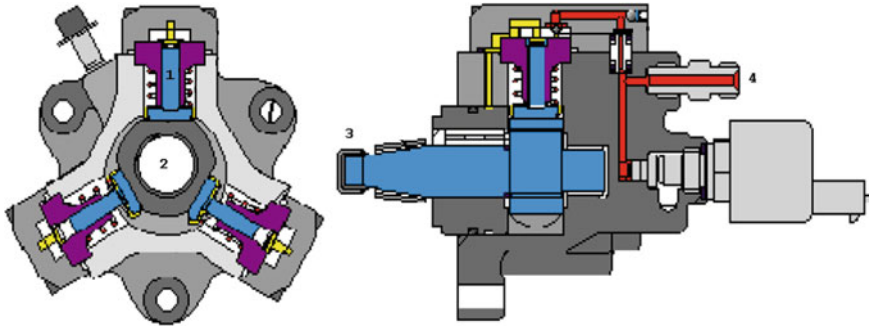


Fig. 2.13 High pressure pump scheme (taken from [8]): 1 piston; 2 triangular cam; 3 shaft; 4 exit hole

An overall of the components constituting a common rail injection system used to feed a 2.0-liters GDI engine is shown in Fig. 2.12. In detail, we have depicted: the electrical fuel pump in Fig. 2.12a; the high pressure mechanical pump in Fig. 2.12b; the fuel rail line in Fig. 2.12c; the control electro-valve in Fig. 2.12d; the pressure sensor in Fig. 2.12e; the solenoid injector in Fig. 2.12f.

2.4.1 High Pressure Mechanical Pump

Figure 2.13 shows a drawing of the high pressure mechanical pump. This pump is formed by three small pistons arranged in radial position (radial-jet) at an angular distance of 120° . The pump is motored by the engine through the camshaft without the need of phasing since the start and duration of injection are imposed by the electronic control unit, which directly controls the opening of the injectors. The alternating movement of the three small pistons is assured by a triangular cam connected to the pump's shaft and each pumping group is characterized by an intake and exhaust valve. The combined action of the three pumping groups allows to reach high pressure values, up to 100–120 (bar), maintaining low level of residual pressure into the external manifolds.

References

1. C. Preussner, C. Dring, S. Fehler, S. Kampmann, Gdi: Interaction between mixture preparation, combustion system and injector performance. SAE Technical Paper (no. 980498) (1998). [10.4271/980498](#)
2. W. Anderson, J. Yang, D.D. Brehob, J.K. Vallance, R.M. Whiteaker, Understanding the thermodynamics of direct injection spark ignition (disi) combustion systems: An analytical and experimental investigation. SAE Technical Paper (no. 962018) (1996). [10.4271/962018](#)

3. M. Kawamoto, T. Honda, H. Katashiba, M. Sumida, N. Fukutomi, K. Kawajiri, A study of center and side injection in spray guided disi concept. SAE Technical Paper (no. 2005-01-0106) (2005).[10.4271/2005-01-0106](#)
4. T. Georjon, E. Bourguignon, T.D.B. Delhay, P. Voisard, Characteristics of mixture formation and combustion in a spray-guided concept gasoline direct injection engine. SAE Technical Paper (no. 2000-01-0534) (2000).[10.4271/2000-01-0534](#)
5. Y. Iwamoto, K. Noma, O. Nakayama, T. Yamauchi, H. Ando, Development of gasoline direct injection engine. SAE Technical Paper (no. 970541) (1997).[10.4271/970541](#)
6. Y. Takagi, T. Itoh, S. Muranaka, A. Iiyama, Y. Iwakiri, T. Urushihara, K. Naitoh, Simultaneous attainment of low fuel consumption high output power and low exhaust emissions in direct injection si engines. SAE Technical Paper (no. 980149) (1998).[10.4271/980149](#)
7. J. Geiger, M. Grigo, O. Lang, P. Wolters, P. Hupperich, Direct injection gasoline engines—combustion and design. SAE Technical Paper (no. 1999-01-0170) (1999).[10.4271/1999-01-0170](#)
8. K. Ahlin, Modelling of pressure waves in the common rail diesel injection system. Ph.D. thesis, University of Linköping, Sweden, 2000

Common Rail System for GDI Engines

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