MIXTURE FORMATION AND COMBUSTION IN CI AND SI ENGINES

7.0 Mixture Formation in Diesel Engines

Diesel engines can be operated both in the two-stroke and four-stroke process. Diesel engines that run at high and average-speeds are utilized, for example, in passenger cars, commercial vehicles, industrial, maritime and stationary applications. These engines are customarily equipped with one or two inlet valves and one or two outlet valves. The bowl piston forms part of the combustion chamber. In most applications, in each cylinder, a single, centrally positioned injector with a multi-hole nozzle is used. An example of the spray in such combustion chamber is shown in **Figure 1**. Low-speed two-stroke diesel engines are mostly used to power ships and in stationary power generation. Modern two-stroke diesel engines have **uniflow scavenging**, intake ports and a centrally arranged outlet valve. In these applications, usually two to four injectors with multi-hole nozzles positioned on the periphery of the combustion chamber are employed. The fuel is introduced to the combustion chamber in a tangential direction.

In addition to the mixing energy induced by the injection system, mixture formation is strongly dependent on the interaction of the fuel spray with the in-cylinder flow. **Figure 2** gives a schematic representation of the two main in-cylinder flow structures important for mixture formation in a DI-Diesel engine: swirl and squish flow. The swirl flow is a rotating flow around the cylinder axis. It is generated by the geometry of the intake ports and in four-stroke engines additionally by the geometry of the valve seats. The squish flow is generated when the piston approaches top dead center, displacing the air in the squish area. Both the squish and the swirl flow can improve the mixing between fuel and ambient air. Other directed flow structures generated by the gas exchange, as for example tumble flow, are usually decayed during compression other aligned flows, such as tumble flows, usually break down during compression.

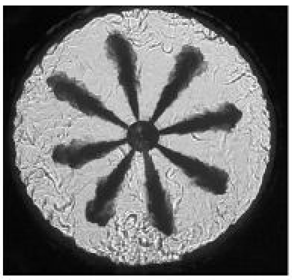

Figure 1: The spray of a direct-injecting four-stroke diesel engine

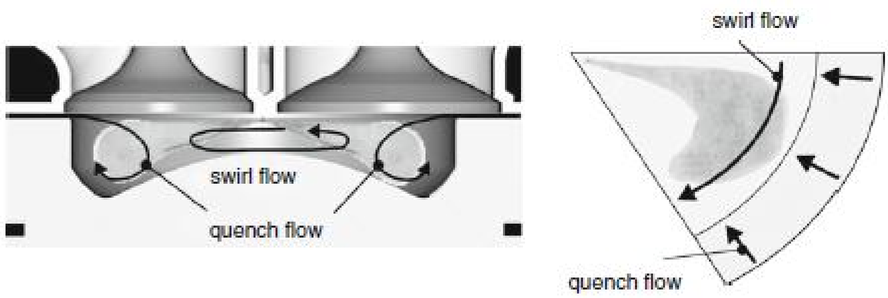

Figure 2: Macroscopic flow structures in the combustion chamber

Especially **smaller passenger car** and some **light-duty truck** engines use **deep ω-piston bowls** with relatively small diameters and high swirl ratios. The **swirl ratio** is strongly dependent on the **engine speed** and an optimization of the swirling motion for different points of operation for example by adjustable guiding vanes in the inlet port is necessary. The injection system has to be adjusted to the swirling motion. Combustion processes with high swirl ratios usually use fewer nozzle holes in order to avoid an interaction of adjacent fuel jets. Modern large four-stroke engines employed in trucks and other heavy-duty power generating and marine engines, in most cases, use combustion systems with low swirl which has a shallow piston bowl compared to passenger car engines. The advantage of a **low-swirl** combustion process is the **improved volumetric efficiency**, since the generation of directed flow structures always increases gas exchange losses. Here, the mixing energy is supplied mainly by the injection system. In contrast, **large two-stroke engines** have a **strong swirling motion** due to the **scavenging process**. However, usually, very shallow piston bowls with a diameter equal to the engine bore are used because of the position of the injection system. As a result, two-stroke engines have a weak squish flow.

7.1 Phenomenology of Mixture Formation

The injection nozzle represents the link between the injection system and the combustion chamber. The fuel leaves the nozzle at high speed through small holes with diameters in the order of 0.12 mm for passenger car engines up to about 1.5 mm in the case of very large two-stroke diesel engines. **Figure 3** shows a qualitative sketch of the fuel spray exiting the injection nozzle. The spray generated during injection can be roughly subdivided into two regions, one with **a dense spray near the nozzle exit** and **a thin spray region further down the flow**. The first decomposition of the cohesive fuel spray into **ligaments and droplets is called primary spray breakup**. In modern high-pressure injection systems, cavitation and turbulence are the most important mechanisms of primary spray breakup. In the injection nozzle, the liquid fuel is accelerated into the nozzle holes in the transition from the nozzle blind-hole.

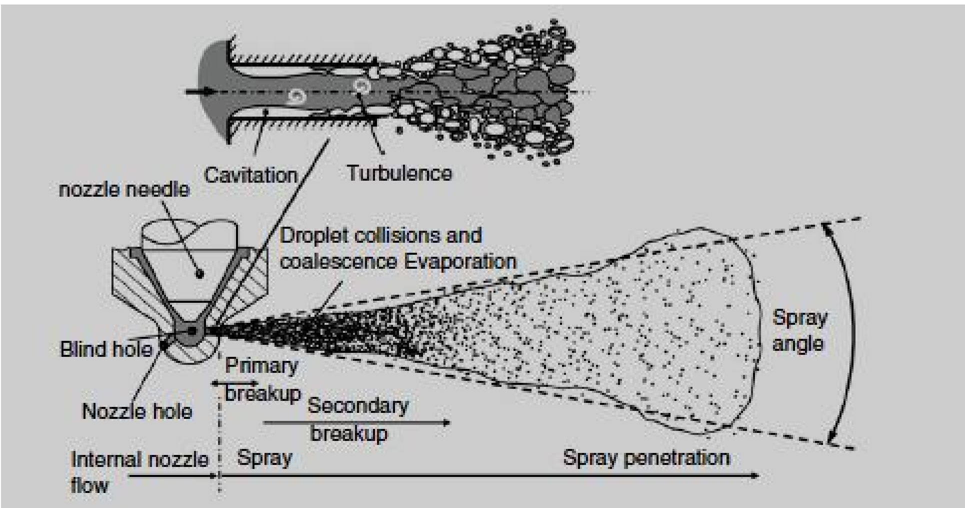

Figure 3: Schematic representation of internal nozzle flow and spray propagation

The change of flow direction on the edge of the nozzle hole leads to the formation of a "**vena contracta**", which further lowers the **static pressure** in the fluid. This reduction is strongly dependent on the geometry of the nozzle and especially on the radius of curvature of the inlet edge of the nozzle hole. **If the pressure at the vena contracta falls below the vapor pressure of the fluid, hydrodynamic cavitation is initiated and vapor bubbles are created**. Depending on the flow parameters, the cavitation can either be stabilized to reach the nozzle hole outlet or the flow can fully or partly reattach. Cavitation reduces both the effective flow area of the nozzle as well as friction. In case of small needle lifts, cavitation structures can also arise in needle seat areas which either break up in the blind hole and thus increase turbulence or enter the nozzle holes, thus promoting further cavitation. **When they leave the nozzle holes, the cavitation bubbles collapse very quickly due to the high pressures in the combustion chamber, which leads to an increase of turbulence and faster primary spray breakup**.

Break-up of already existing droplets into smaller droplets due to **aerodynamic forces** caused by the relative speed between droplet and the environment is called **secondary spray breakup**. In addition, droplets can collide with each other and **coalesce**.

The spray momentum leads to air entrainment of the surrounding combustion chamber air into the spray. The droplets are heated up as a result of convective heat transfer and temperature radiation of the hot chamber walls, and the fuel finally begins to evaporate. Besides the physical properties and the combustion chamber conditions (pressure, temperature), the **rate of fuel evaporation is determined by the size of the droplet surface formed and thus on primary and secondary breakup as well as on the amount of air entrained into the spray**.

In the case of the diesel engine, mixture formation cannot be considered independently of combustion. It is indeed the distinctive feature of diesel engine combustion that spray propagation, mixture formation, and combustion progress in partial simultaneity. Only a small amount of the injected fuel mixes nearly homogeneously with the air in the combustion chamber during ignition delay. After ignition, this amount burns almost instantly. Afterwards, mixture formation and combustion proceed simultaneously, and combustion is controlled by the mixture formation processes.

Spray propagation and mixture formation are understood quite well at present, at least qualitatively, and can be described approximately with semi-empirical models.

7.2 Autoignition and the Combustion Sequence

The period of time between injection start and combustion start is called the **ignition delay**. The physical and chemical processes occurring during this time are very complex. The essential physical processes are the atomization of the fuel, vaporization, and mixing of fuel vapour with air, forming an ignitable mixture. The chemical processes that lead to autoignition of the hydrocarbons contained in the fuel under typical diesel conditions are characterized by a highly complex, degenerated chain branching mechanism.

In the diesel injection spray, ignition occurs in areas with local air fuel ratios of about $0.25 < F/A < 0.65$. The ignition delay can be controlled by means of temperature and pressure at the start of injection, which in turn depend on the inlet temperature and pressure, the compression ratio, injection start and the wall temperatures. In addition, the ignitability of the fuel (cetane number) and further parameters such as injection pressure, the geometry of the nozzle holes and the in-cylinder flow have a major effect on the ignition delay duration and the ignition location.

Figure 4 provides a schematic representation of the injection and combustion sequence of a diesel engine with direct injection. As we can see, the sequence of diesel engine combustion can be subdivided into three phases.

Phase 1: Initial Premixed Combustion

The first phase follows immediately after injection. The fuel injected during ignition delay mixes with the air in the combustion chamber and forms a nearly homogeneous and reactive mixture. After the ignition delay, which is physically and chemically controlled, this mixture burns very quickly. Since areas with premixed combustion arise in the main combustion phase as well, this phase is called initial premixed combustion. The rate of heat release is controlled in this combustion phase by the speed of the chemical reactions and by the amount of fuel/air mixture formed during ignition delay. The combustion noise typical of diesel engines is caused by the high speed of pressure rise at the start of combustion. This speed of pressure rise can be influenced by changing the timing of injection, in which case the following rule applies: an early injection start leads to a “hard” combustion and a late start to a “soft” injection (**see Fig. 5**). Moreover, the combustion noise can be considerably reduced by a preinjection. In this case, at first only a small fuel amount of about 2% is injected which leads after the ignition delay to only a small amount of heat release and to a small pressure increase. The increased temperatures lead however to a significant reduction of the ignition delay of the main injection, which leads to a reduction of the amount of premixed combustion with a positive effect on noise.

Phase 2: Main Combustion

In the second phase, heat release is controlled by the turbulent mixing processes between the fuel and air and is therefore also called mixture-controlled combustion. In this phase, injection, spray breakup, droplet evaporation, mixing with air, combustion, and pollutant formation all take place simultaneously. **Figure 6** shows a cross-section through a reacting diesel injection spray following the conceptual model of Dec and Flynn et al.. The model describes the quasi-steady phase during main combustion and is, strictly speaking, only valid in quiescent conditions. The fluid fuel spray penetrates into the combustion chamber, mixes with air and evaporates. The air ratio in the spray increases both with increasing distance to the injection nozzle as well as with the distance to the spray axis.

Downstream of the liquid penetration length, a rich mixture zone is formed which leads to partial oxidation of the fuel and temperatures up to 1,600 K. According to Flynn et al., the air ratio in this zone is in the range of $0.25 < F/A < 0.5$, and about **15%** of the total heat is released in this zone. Among the partially oxidized products of premixed combustion are also found precursor species, which lead to particle formation further downstream in the middle of the flame. A diffusion flame is formed around the injection spray on an iso-surface with a stoichiometric air fuel ratio. The partially oxidized products of the rich premixed combustion and the particles formed move further downstream and are transported into the diffusion flame, where they are completely oxidized into carbon dioxide and water.

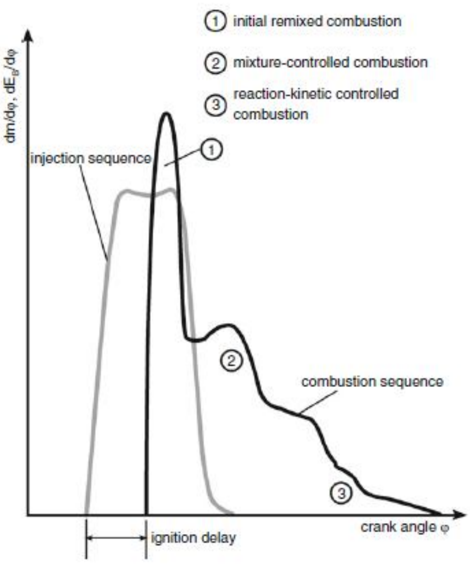

Figure 4: Injection and combustion sequence in diesel engine

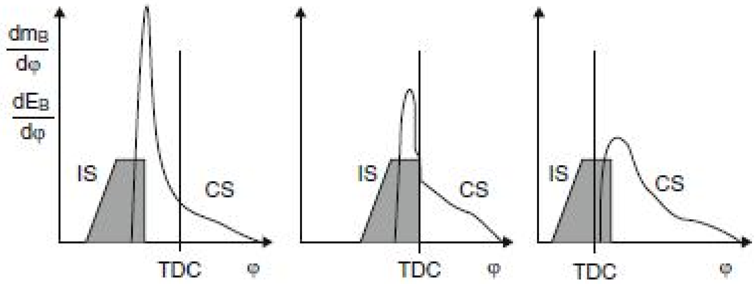

Figure 5: Injection rate (IR) and heat release rate (HRR) with early (left) and late (right) combustion.

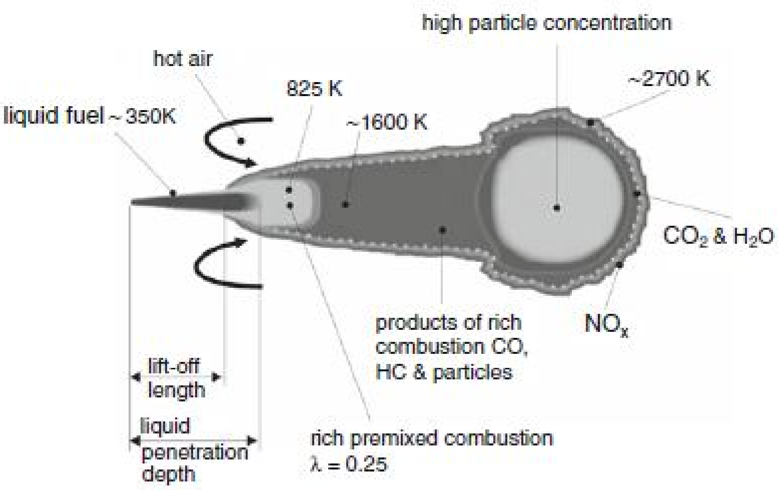

Figure 6: Conceptional model of diesel combustion

Temperatures rise to up to 2,700 K. Due to the high temperatures, nitrogen oxides are formed on the lean side of the diffusion flame. Near the injection nozzle, vaporization processes and chemical reactions in the spray determine the distance from the injection nozzle at which the diffusion flame establishes itself. The axial distance between the injection nozzle and the diffusion flame is called the lift-off length and is an important property of a diesel flame with regard to soot formation.

Phase 3: Post-combustion

After the injection process is finished, no additional momentum is added to the spray by the injection and the flame jet structure evolves into a pocket of rich premixed products surrounded by a diffusion flame. The exact properties of this zone depend on the injection system. If the nozzle needle closes very quickly, then the last fuel parcels still have high speed, so that they have a similar combustion sequence as in main combustion. On the other hand, a slow closure of the needle leads to low speeds of the last fuel parcels with low entrainment of oxygen and consequently increased formation of soot. With the expansion of the piston in the direction of the bottom dead center, the temperatures in the combustion chamber are lowered. The reaction rates go down with the temperatures, so that combustion is chemically controlled again. This phase is of extreme importance for the oxidation of the previously formed soot, of which over 90% is decomposed again. The temperatures during this combustion phase should be high, since soot oxidation is very slow below 1,300 K.

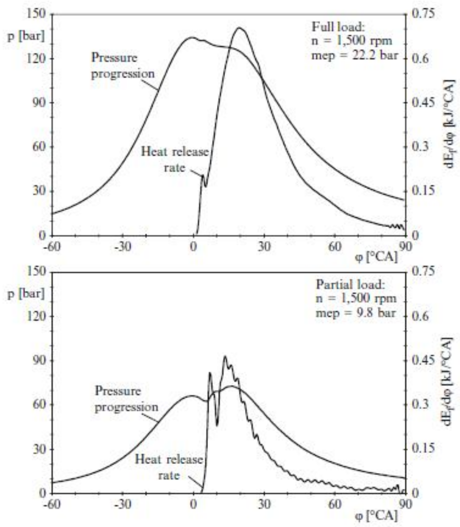

Figure 7: Pressure profile and heat release rate in a high speed diesel engine at full and partial loads.

It leads to the heating up of the air fuel mixture in the cylinder and thus to temperature and pressure increase. As an example, Figure 7 shows the pressure progression and heat release rates at full and partial load in a high speed heavy-duty diesel engine with relatively late injection.

7.3 Spark Ignition Engines

7.3 Differences between Premixed Flame and Diffusion Combustion

Mixture formation, ignition and combustion of the SI engine differ fundamentally from the diesel engine.

Classic SI engines with homogeneous mixture formation are characterized by their premixed flame. Fuel and air are mixed long before ignition as completely as possible. This is realized either outside the combustion chamber (intake-manifold fuel injection) or in the combustion chamber itself (homogeneous direct injection in the intake cycle). The mixture exists in a gaseous state in the combustion chamber at the time of ignition. Since SI engine fuel is not a highly ignitable fuel relatively speaking, the mixture must be externally ignited by an ignition spark from the spark plug.

A major differentiating factor between premixed flame and diffusion combustion is thus the time available for mixture formation and the question of whether there is a homogeneous mixture at the point of ignition or not. Therefore, signs of diffusion flames can arise in the SI engine under unfavorable mixture formation conditions as well. For example, if the cold combustion chamber walls are wetted after cold start by large fuel droplets and the impinged fuel cannot be completely vaporized toward the end of the compression phase, local mixture inhomogeneities result (e.g. very rich zones) leading to combustion with possible soot formation. A charge movement that has not been optimally designed can lead to similar results, e.g. if fuel droplets wet the cylinder liner due to excessive tumble flow. Since the soot particles have a characteristic radiation, such signs of unintentional diffusion combustion can be detected in the combustion chamber with fiber optical measurement technology.

7.4 Combustion in SI Engines

The combustion process in a spark ignition engine is broadly divided into three stages: (i) ignition and flame development (ii) Flame propagation and (iii) Flame quench.

Ignition and Flame Development

In SI engines, the ignition of the charge in the combustion chamber is initiated by the introduction of spark from an external source (spark plug). An important focus in this region is the time taken for the consumption a certain percentage of the charge in the combustion chamber. While some researches consider the consumption of 5% of the charge, others consider 10%. The time it takes for the consumption of this percentage of charge (10%) is known as the **delay period**. Research has shown that during this period of combustion only a negligible in-cylinder pressure rise is noticed and it is assumed that no useful work is produced in the engine.

Flame propagation

This stage is the period where 80 to 90% of the mass of the fuel is consumed (mainly between 10 and 90% of charge is burned). This is when the bulk of the fuel in the cylinder is burned and useful work is produced.

Flame quench

This stage is when the remaining 5 to 10% of the charge in the cylinder is consumed and a halt in combustion occurs. Pressure quickly decreases at this stage of combustion and combustion stops.

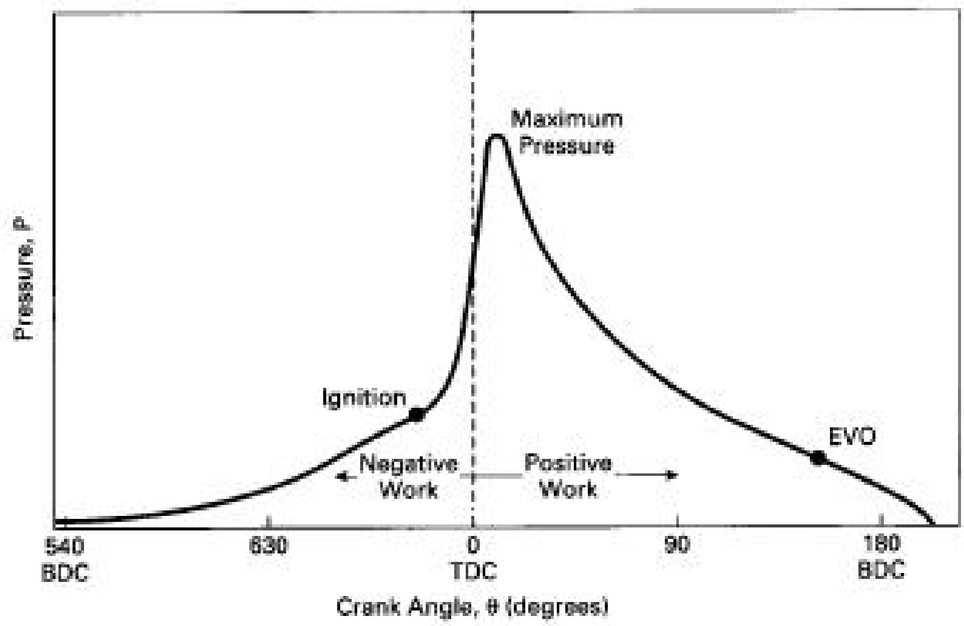

Figure 8: In-Cylinder pressure trace in an SI engine.