Diesel Engines Particulates Formation
A State of the Art

Vincent Mahieu
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1: Executive summary

This report intends to describe the present situation (early 2001) concerning DIESEL particulates emissions and formation.

First, a chapter will remind the Diesel combustion basics: combustion process and parameters, fuel sprays, ignition delay and burning rates.

Then, the next part will explain the links between Diesel combustion process and particulates matter genesis.

To follow, a review of the main known parameters affecting particulates in number, size and mass will be performed, including the common reduction strategies.

Before the conclusion, a chapter will investigate the last trends, the recent and future technologies affecting particulates formation, and more widely, Diesel combustion processes.
2: Diesel Engines Combustion Basics

2.1 Main Characteristics

2.1.1: Diesel combustion process and parameters

The essential features of the compression-ignition or diesel engine combustion process can be summarized as follows.

Fuel is injected by the fuel-injection system into the engine cylinder toward the end of the compression stroke, just before the desired start of combustion. The liquid fuel, usually injected at high velocity as one or more jets through small orifices or nozzles in the injector tip, atomizes into small drops and penetrates into the combustion chamber. The fuel vaporizes and mixes with the high-temperature, high-pressure cylinder air.

Since the air temperature and pressure are above the fuel's ignition point, spontaneous ignition of portions of the already-mixed fuel and air occurs after a delay period of a few crank angle degrees. The cylinder pressure increases as combustion of the fuel-air mixture occurs. The consequent compression of the unburned portion of the charge shortens the delay before ignition for the fuel and air which has mixed to within combustible limits, which then burns rapidly. It also reduces the evaporation time of the remaining liquid fuel.

Injection continues until the desired amount of fuel has entered the cylinder. Atomization, vaporization, fuel-air mixing, and combustion continue until essentially all the fuel has passed through each process. In addition, mixing of the air remaining in the cylinder with burning and already burned gases continues throughout the combustion and expansion processes.

It will be clear from this summary that the compression-ignition combustion process is extremely complex.

The details of the process depend on the characteristics of the fuel, on the design of the engine's combustion chamber and fuel-injection system, and on the engine's operating conditions. It is an unsteady, heterogeneous, three-dimensional combustion process. While an adequate conceptual understanding of diesel engine combustion has been developed, to date an ability to describe many of the critical individual processes in a quantitative manner is lacking.

Some important consequences of this combustion process on engine operation are the following:

1. Since injection commences just before combustion starts, there is no knock limit as in the spark-ignition engine resulting from spontaneous ignition of the premixed fuel and air in the end-gas. Hence a higher engine compression ratio can be used in the compression-ignition engine, improving its fuel conversion efficiency relative to the SI engine.

2. Since injection timing is used to control combustion timing, the delay period between the start of injection and start of combustion must be kept short (and reproducible). A short delay is also needed to hold the maximum cylinder gas pressure below the maximum the engine can tolerate. Thus, the spontaneous ignition characteristics of the fuel-air mixture must be held within a specified range. This is done by requiring that diesel fuel have a cetane number (a measure of the ease of ignition of that fuel in a typical diesel environment) above a certain value.

3. Since engine torque is varied by varying the amount of fuel injected per cycle with the engine airflow essentially unchanged, the engine can be operated unthrottled. Thus, pumping work requirements are low, improving part-load mechanical efficiency relative to the SI engine.
4. As the amount of fuel injected per cycle is increased, problems with air utilization during combustion lead to the formation of excessive amounts of soot, which cannot be burned up prior to exhaust. This excessive soot or black smoke in the exhaust constrains the fuel/air ratio at maximum engine power to values 20 percent (or more) lean of stoichiometric. Hence, the maximum indicated mean effective pressure (in a naturally aspirated engine) is lower than values for an equivalent spark-ignition engine.

5. Because the diesel always operates with lean fuel/air ratios (and at part load with very lean fuel/air ratios), the effective value of \( \gamma = c_p/c_v \) over the expansion process is higher than in a spark-ignition engine. This gives a higher fuel conversion efficiency than the spark-ignition engine, for a given expansion ratio.

The major problem in diesel combustion chamber design is achieving sufficiently rapid mixing between the injected fuel and the air in the cylinder to complete combustion in the appropriate crank angle interval close to top-centre. The study of various parameters indicates that mixing rates control the fuel burning rate.

Commercial diesel engines are made with a very large range of cylinder sizes, with cylinder bores varying from about 70 to 900 mm. The mean piston speed (proportional to the stroke x rpm multiplication) at maximum rated power is approximately constant over this size range, so the maximum rated engine speed will be inversely proportional to the stroke. For a fixed crank angle interval for combustion (of order 40° to 50° to maintain high fuel conversion efficiency), the time available for combustion will, therefore, scale with the stroke.

Thus, at the small end of the diesel engine size range, the mixing between the injected fuel and the air must take place on a time scale some 10 times shorter than in engines at the large end of this range. It would be expected, therefore, that the design of the engine combustion chamber (including the inlet port and valve) and the fuel-injection system would have to change substantially over this size range to provide the fuel and air motion inside the cylinder required to achieve the desired fuel-air mixing rate. As engine size decreases, more vigorous air motion is required while less fuel jet penetration is necessary. It is this logic, primarily, that leads to the different diesel combustion chamber designs and fuel injection systems found in practice over the large size range of commercial diesel engines.

![Injection Diagram](image)

**Fig.1: Direct Diesel Injection**
2.1.2: Synthesis

- Direct injection of liquid fuel into combustion chamber
- Heterogeneous combustion
- Self ignitable fuel
- Combustion with turbulent diffusion flames
- Load (torque) tuning by fuel mass injected. Air mass remains constant (not throttled).
- Important air excess. Lean combustion.

The $\Phi$ adimensional parameter explained below express the fuel to air ratio.

$$\Phi = \frac{\{ m_{\text{fuel}} / m_{\text{air}} \}_{\text{real}}}{\{ m_{\text{fuel}} / m_{\text{air}} \}_{\text{stoichiometry}}}$$

For Diesel combustion,

$$0.3 < \Phi_{\text{global}} < 0.8$$

2.1.3: Combustion process time description

**A: Cylinder pressure data**

![Cylinder pressure data graph](image)

Fig.2: Direct Diesel Injection Model

1: Ignition delay – no significant heat release
2: “Premixed”, or rapid combustion phase
3: “Mixing controlled” combustion phase
4: “Late” combustion phase
B: Rate of heat release

![Graph showing rate of heat release and combustion phases](image)

**Fig.3: DI Diesel engine heat release rate**

*Ignition delay (ab).* The period between the start of fuel injection into the combustion chamber and the start of combustion determined from the change in slope on the $p\alpha$ diagram, or from a heat-release analysis of the $p(\alpha)$ data, or from a luminosity detector.

*Premixed or rapid combustion phase (bc).* In this phase, combustion of the fuel which has mixed with air to within the flammability limits during the ignition delay period occurs rapidly in a few crank angle degrees. When this burning mixture is added to the fuel which becomes ready for burning and burns during this phase, the high heat-release rates characteristic of this phase result.

*Mixing-controlled combustion phase (cd).* Once the fuel and air which premixed during the ignition delay have been consumed, the burning rate (or heat-release rate) is controlled by the rate at which mixture becomes available for burning. While several processes are involved—liquid fuel atomization, vaporization, mixing of fuel vapor with air, preflame chemical reactions—the rate of burning is controlled in this phase primarily by the fuel vapor-air mixing process. The heat-release rate may or may not reach a second (usually lower) peak in this phase; it decreases as this phase progresses.

*Late combustion phase (de).* Heat release continues at a lower rate well into the expansion stroke. There are several reasons for this. A small fraction of the fuel may not yet have burned. A fraction of the fuel energy is present in soot and fuel-rich combustion products and can still be released. The cylinder charge is nonuniform and mixing during this period promotes more complete combustion and less-dissociated product gases. The kinetics of the final burnout processes become slower as the temperature of the cylinder gases fall during expansion.

2.1.4: Disadvantages of Diesel engines

- Cold start difficulty
- Noisy (because of sharp pressure rise)
- Inherently slower combustion
- Low power to weight ratio
- Expensive components
- NOx and Particulates emissions
2.2: Injectors and Fuel sprays [6]

2.2.1 Fuel Injection

The fuel is introduced into the cylinder of a diesel engine through a nozzle with a large pressure differential across the nozzle orifice. The cylinder pressure at injection is typically in the range 50 to 100 atm. Fuel injection pressures in the range 200 to 1700 atm are used depending on the engine size and type of combustion system employed. These large pressure differences across the injector nozzle are required so that the injected liquid fuel jet will enter the chamber at sufficiently high velocity to atomize into small-sized droplets to enable rapid evaporation and traverse the combustion chamber in the time available and fully utilize the air charge.

![Classical Diesel Injection Loop](image)

Figure 4: Classical Diesel Injection Loop

The task of the fuel-injection system is to meter the appropriate quantity of fuel for the given engine speed and load to each cylinder and each cycle. Another task is to inject that fuel at the appropriate time in the cycle at the desired rate with the spray configuration required for the particular combustion chamber employed.

![Nozzle designs](image)

Figure 5: Nozzle designs

The most important part of the injection system is the nozzle. It is important to keep the volume of fuel left between the needle and nozzle orifices (the sac volume) as small as possible to prevent any fuel flowing into the cylinder after injection is over, to control hydrocarbon emissions. Multi-hole nozzles are used with most direct-injection systems. The shape of the pin on the end of the nozzle needle controls the spray pattern and fuel-delivery characteristics.
If the pressure upstream of the injector nozzle(s) can be estimated or measured, and assuming the flow through each nozzle is quasi steady, incompressible, and one dimensional, the mass flow rate of fuel injected through the nozzle \( \dot{m}_f \) is given by:

\[
\dot{m}_f = C_D A_n \sqrt{2 \rho_f \Delta p}
\]

where \( A_n \) is the nozzle minimum area, \( C_D \) the discharge coefficient, \( \rho_f \) the fuel density, and \( \Delta p \) the pressure drop across the nozzle.

If the pressure drop across the nozzle and the nozzle open area are essentially constant during the injection period, the mass of fuel injected is then

\[
\dot{m}_f = C_D A_n \sqrt{2 \rho_f \Delta p} \frac{\Delta \theta}{360N}
\]

where \( \Delta \theta \) is the nozzle open period in crank angle degrees and \( N \) is engine speed. Equations (2.1) and (2.2) illustrate the dependence of injected amounts of fuel on injection system and engine parameters.

2.2.2 Overall Spray Structure

Nozzle diameters cover the range 0.2 to 1 mm diameter, with length/diameter ratios from 2 to 8. Typical distillate diesel fuel properties are: relative specific gravity of 0.8, viscosity between 3 and 10 kg/m.s and surface tension about 3 x 10\(^2\) N/m (at 300 K). Figure 6 illustrates the structure of a typical DI engine fuel spray. As the liquid jet leaves the nozzle it becomes turbulent and spreads out as it entrains and mixes with the surrounding air.

The initial jet velocity is greater than 10\(^2\) m/s. The outer surface of the jet breaks up into drops of order 10 \(\mu\text{m} \) diameter, close to the nozzle exit. The liquid column leaving the nozzle disintegrates within the cylinder over a finite length called the break-up length into drops of different sizes. As one moves away from the nozzle, the mass of air within the spray increases, the spray diverges, its width increases, and the velocity decreases. The fuel drops evaporate as this air-entrainment process proceeds. The tip of the spray penetrates further into the combustion chamber as injection proceeds, but at a decreasing rate.

Figure 6: Schematic of Diesel Fuel Spray
Figure 7 shows photographs of a diesel fuel spray injected into quiescent air in a rapid-compression machine which simulates diesel conditions. Two different photographic techniques, back lighting and shadowgraph, have been used to distinguish the liquid-containing core of the jet and the extent of the fuel vapor region of the spray which surrounds the liquid core. The region of the jet closest to the nozzle (until injection ceases at 3.3 ms) contains liquid drops and ligaments; the major region of the spray is a substantial vapor cloud around this narrow core which contains liquid fuel.

![Figure 7: Back lighting and shadowgraph spray visualization](image)

The highest velocities are on the jet axis. The equivalence ratio is highest on the centerline (and fuel-rich along most of the jet), decreasing to zero (unmixed air) at the spray boundary. Once the sprays have penetrated to the outer regions of the combustion chamber, they interact with the chamber walls. The spray is then forced to flow tangentially along the wall. Eventually the sprays from multi-hole nozzles interact with one another.

A schematic of the spray pattern which results when a fuel jet is injected radially outward into a swirling flow is shown in Figure 8.

![Figure 8: Schematic of fuel spray into swirling airflow](image)

---

1 The back lighting identifies regions where sufficient liquid fuel (as ligaments or drops) is present to attenuate the light. The shadowgraph technique responds to density gradients in the test section, so it identifies regions where fuel vapor exists.
Because there is now relative motion in both radial and tangential directions between the initial jet and the air, the structure of the jet is more complex. As the spray entrains air and slows down it becomes increasingly bent toward the swirl direction; for the same injection conditions it will penetrate less with swirl than without swirl. An important feature of the spray is the large vapor-containing region downstream of the liquid-containing core.

To couple the spray-development process with the ignition phase of the combustion, it is important to know which regions of the spray contain the fuel injected at the beginning of the injection process. These regions of the sprays are likely to autoignite first.

2.2.3 Atomization

Under diesel engine injection conditions, the fuel jet usually forms a cone-shaped spray at the nozzle exit. This type of behavior is classified as the atomization breakup regime, and it produces droplets with sizes very much less than the nozzle exit diameter.

This behavior is different from other modes of liquid jet breakup.

At low jet velocity, in the Rayleigh regime, breakup is due to the unstable growth of surface waves caused by surface tension and results in drops larger than the jet diameter. As jet velocity is increased, forces due to the relative motion of the jet and the surrounding air augment the surface tension force, and lead to drop sizes of the order of the jet diameter. This is called the first wind-induced breakup regime.

A further increase in jet velocity results in breakup characterized by divergence of the jet spray after an intact or undisturbed length downstream of the nozzle. In this second wind-induced breakup regime, the unstable growth of short-wavelength waves induced by the relative motion between the liquid and surrounding air produces droplets whose average size is much less than the jet diameter.

Further increases in jet velocity lead to breakup in the atomization regime, where the breakup of the outer surface of the jet occurs at, or before, the nozzle exit plane and results in droplets whose average diameter is much smaller than the nozzle diameter. Aerodynamic interactions at the liquid/gas interface appear to be one major component of the atomization mechanism in this regime.

2.2.4 Spray penetration

The speed and extent to which the fuel spray penetrates across the combustion chamber has an important influence on air utilization and fuel-air mixing rates.

In some engine designs, where the walls are hot and high air swirl is present, fuel impingement on the walls is desired. However, in multispray DI diesel combustion systems, overpenetration gives impingement of liquid fuel on cool surfaces which, especially with little or no air swirl, lowers mixing rates and increases emissions of unburned and partially burned species. Yet underpenetration results in poor air utilization since the air on the periphery of the chamber does not then contact the fuel. Thus, the penetration of liquid fuel sprays under conditions typical of those found in diesel engines has been extensively studied.

Many correlations based on experimental data and turbulent gas jet theory have been proposed for fuel spray penetration.

These predict the penetration S of the fuel spray tip across the combustion chamber for injection into quiescent air, as occurs in larger DI engines, as a function of time.
An evaluation of these correlations is given below, based on a gas jet mixing model for the spray:

\[ S = 3.07 \left( \frac{\Delta p}{\rho_g} \right)^{1/4} \left( \frac{td_n}{T_g} \right)^{1/2} \left( \frac{294}{T_g} \right)^{1/4} \]

where \( \Delta p \) is the pressure drop across the nozzle, \( t \) is time after the start of injection, and \( d_n \) is the nozzle diameter. All quantities are expressed in SI units: \( t \) in seconds, \( S \) and \( d_n \) in meters, \( \Delta p \) in pascals, \( \rho_g \) in kilograms per cubic meters, and \( T_g \) in kelvins.

More detailed studies have examined the spray tip location as a function of time, following start of a diesel injection process in high-pressure bombs. Fig. 9 illustrate the sensitivity of the spray tip position as a function of time to ambient gas state and injection pressure for fuel jets injected into quiescent air at room temperature. These data show that the initial spray tip penetration increases linearly with time \( t \) and, following jet breakup, then increase as \( \sqrt{t} \). Injection pressure has a more significant effect on the initial motion before breakup; ambient gas density has its major impact on the motion after breakup.

![Figure 9: Spray tip penetration sensitivity to ambient and injection pressures](image)

2.2.5 Droplet Size Distribution

While the distribution of fuel via the spray trajectory throughout the combustion chamber is important, atomization of the liquid fuel into a large number of small drops is also necessary to create a large surface area across which liquid fuel can evaporate. Here we review how the drop size distribution in the fuel spray depends on injection parameters and the air and fuel properties. Since the measurement of droplet characteristics in an operating diesel engine is extremely difficult, most results have come from studies of fuel injection into constant-volume chambers filled with high-pressure quiescent air at room temperature.

During the injection period, the injection conditions such as injection pressure, nozzle orifice area, and injection rate may vary. Consequently, the droplet size distribution at a given location in the spray may also change with time during the injection period. In addition, since the details of the atomization process are different in the spray core and at the spray edge, and the trajectories of individual drops depend on their size, initial velocity, and location within the spray, the drop size distribution will vary with position within the spray. None of these variations has yet been adequately quantified.
However, within the dense early region of the spray, secondary atomization phenomena – coalescence and breakup – occur which will change the droplet size distribution and mean diameter. The downstream drop size in the solid-cone sprays used in diesel-injection systems is markedly influenced by both drop coalescence and breakup. Eventually a balance is reached as coalescence decreases (due to the expansion of the spray) and breakup ceases (due to the reduced relative velocity between the drops and the entrained gas).

![Droplet size distribution in diesel fuel spray](image)

Figure 10: Droplet size distribution in diesel fuel spray

The distributions shown in the figure are frequency distributions of drop volume. The peak in the distribution shifts to larger drop diameters as the radial position decreases: on average, the drops are smaller at the periphery of the spray.

2.2.6 Spray Evaporation

The injected liquid fuel, atomized into small drops near the nozzle exit to form a spray, must evaporate before it can mix with air and burn. Figure 7 showed the basic structure of an evaporating diesel spray under conditions typical of a large direct-injection engine. Back illumination showed that a core exists along the axis of the spray where the liquid fuel ligaments or drops are sufficiently dense to attenuate the light beam. Additional insight into the physical structure of evaporating sprays can be obtained from the schlieren photographs taken just after the end of injection in a prechamber engine with air swirl, shown in Fig.11. The lowest magnification picture (A) shows the overall structure of the spray. The only liquid-containing region evident is that part of the core nearest the nozzle which shows black on the left of the photograph. The spreading, vapor region of the spray; carried around the chamber by the swirling air flow, appears mottled due to local turbulent vapor concentration and temperature fluctuations. The dark region within the spray vapor region is due to soot formed where the fuel vapor concentration is sufficiently high. It is probable that, after the breakup length, the dense black liquid core of the spray is composed of individual droplets but the concentration is so high along the optical path that the light beam is fully extinguished. However, the last part of the core close to the nozzle tip (B) disperses sufficiently for individual features to be resolved. The small black dots are liquid fuel drops in the size range 20 to 100 μm. Fuel drop vapor trails can be observed in the highest magnification photo (C) corresponding to various stages of evaporation. These range from drops showing little surrounding vapor to vapor trails with little liquid remaining at the head. The vapor trails show random orientations relative to the spray axis, presumably due to local air turbulence. The process of droplet evaporation under normal engine operating conditions appears to be rapid relative to the total combustion period.
Figure 11: Shadowgraph of diesel fuel spray just after the end of injection

Let us examine the drop evaporation process in more detail. Consider a liquid drop at close to ambient temperature injected into air at typical end-of-compression engine conditions. Three phenomena will determine the history of the drop under these conditions:

1. Deceleration of the drop due to aerodynamic drag
2. Heat transfer to the drop from the air
3. Mass transfer of vaporized fuel away from the drop
As the droplet temperature increases due to heat transfer, the fuel vapor pressure increases and the evaporation rate increases. As the mass transfer rate of vapor away from the drop increases, so the fraction of the heat transferred to the drop surface which is available to increase further the drop temperature decreases. As the drop velocity decreases, the convective heat-transfer coefficient between the air and the drop decreases. The combination of these factors gives the behavior shown in Fig.12 where drop mass, temperature, velocity, vaporization rate, and heat-transfer rate from the air are shown schematically as a function of time following injection. Analysis of individual fuel drops 25 μm in diameter, injected into air at typical diesel conditions, indicates that evaporation times are usually less than 1 ms.

![Schematic variations of droplets properties](image)

Figure 12: Schematic variations of droplets properties

To quantify accurately the fuel vaporization rate within a diesel fuel spray requires the solution of the coupled conservation equations for the liquid droplets and the air within the combustion chamber. Studies with such models indicate that, under normal diesel engine conditions, 70 to 95 percent of the injected fuel is in the vapor phase at the start of combustion. Evaporation is more than 90 percent complete after 1 ms. However, only 10 to 35 percent of the vaporized fuel has mixed to within flammability limits in a typical medium-speed DI diesel engine. Thus combustion is largely mixing-limited, rather than evaporation-limited. Of course, under cold-starting conditions, evaporation becomes a major constraint.

### 2.3: Fuel-Air Mixing and Burning Rates

#### 2.3.1 Introduction

The relations between fuel spray behavior, flame structure, and fuel burning rate are difficult to quantify. Depending on the spray configuration, the visible flame may then fill almost the entire combustion chamber. The flame and spray geometries are closely related. Mixing processes are also critical during the ignition delay period: while the duration of the delay period is not influenced in a major way by the rates of spray processes which together control "mixing," the amount of fuel mixed with air to within combustible limits during the delay (which affects the rate of pressure rise once ignition has occurred) obviously is directly related to mixing rates. Thus substantial observational evidence supports the mixing-controlled character of diesel engine combustion.
2.3.2 Spray and Flame Structure

The structure of each fuel spray is that of a narrow liquid-containing core (densely filled with drops of order 20 pm in diameter) surrounded by a much larger gaseous-jet region containing fuel vapor (see Fig.7). The fuel concentration in the core is extremely high: local fuel/air equivalence ratios near the nozzle of order 10 have been measured during the injection period. Fuel concentrations within the spray decrease with increasing radial and axial position at any given time, and with time at a fixed location once injection has ended. The fuel distribution within the spray is controlled largely by turbulent-jet mixing processes. Fuel vapor concentration contours determined from interferometric studies of unsteady vaporizing diesel-like sprays, confirm this gaseous turbulent-jet-like structure of the spray, with its central liquid-containing core which evaporates relatively quickly once fuel injection ends.

Figure 13: Outer boundary of liquid fuel spray (glass engine)

The location of the autoignition sites and subsequent spreading of the enflamed region in relation to the fuel distribution in the spray provides further evidence of the mixing-controlled character of combustion. Figure 13 shows how this process occurs under conditions typical of direct-injection quiescent-chamber diesel engines. It shows tracings of the liquid fuel spray and flame boundaries taken from high-speed movies of the injection and combustion processes with central injection of five fuel jets into a disc-shaped chamber in a rapid-compression machine. These and other similar studies show that autoignition first occurs toward the edge of the spray, between the spray tip and the injector nozzle. Experiments where air/fuel ratio contours and where the autoignition sites recorded on movies, showed that autoignition occurred in a concentration band between the equivalence ratios of 1 and 1.5. Subsequent flame development, along mixture contours close to stoichiometric, occurs rapidly, as indicated in Fig.13. Combustion movies such as those in Fig.13 show that flame rapidly envelops each spray once spontaneous ignition occurs.

2.3.3 Burning Rates

The model of diesel combustion obtained from heat-release analyses of cylinder pressure data identifies two main stages of combustion (see Fig.3). The first is the premixed-combustion phase, when the fuel which has mixed sufficiently with air to form an ignitable mixture during the delay period is consumed.
The second is the mixing-controlled combustion phase, where rates of burning (at least in naturally aspirated engines) are lower. Experimental evidence from heat-release analysis indicates that the majority of the fuel (usually more than 75 percent) burns during the second mixing-controlled phase. Such evidence forms the basis for the heat-release models used in diesel engine cycle simulations.

2.3.4 Summing-up and Pertinent Features of Diesel Mixing Process

- Fuel spray phenomena

The injected liquid fuel spray atomizes; the droplets produced vaporize and the fuel vapor and air then mix. The fuel spray contains widely varying fuel concentrations and mixture temperatures. The spray interacts with the combustion chamber walls and the charge motion, and it has a major influence on the fuel air mixing process.

- Spontaneous ignition

Under normal conditions, ignition occurs at approximately 1 ms after the start of injection. The ignition characteristics of the fuel air mixture depends strongly on the mixture temperature and the fuel composition. The diesel fuel property is controlled to give good ignition behaviors which are characterized by the cetane number.

- Effects of fuel-jet and charge motion on mixing-controlled combustion

The diffusion burning is limited by the fuel air mixing rate, which is strongly influenced by the fuel jet interaction with the combustion chamber geometry and the charge motion. The combustion behavior is designed for best trade off for specific fuel consumption and emissions.

Features of Diesel Combustion Process

- Ignition delay
  - Auto-ignition in different parts of combustion chamber
  - After ignition, fuel sprays into hot burned gas
  - Major part of combustion is controlled by fuel air mixing process

Mixing dominated by flow field formed by fuel-jet interacting with combustion chamber walls during injection

- Highly luminous flame:

Substantial soot formation in the fuel rich zone by pyrolysis, followed by substantial subsequent oxidation
Air Fuel Mixing Process

- Importance of air utilization - Smoke limit A/F - 20
- Fuel jet momentum / wall interaction has a larger influence on the early part of the combustion process
- Charge Motion has a larger influence on the later part of the combustion process

Charge Motion Control

- Intake created motion : swirl, etc.
- Piston created motion : squish effect

Challenges in Diesel Combustion

Heavy Duty Diesel Engines

- NOx emission
- Particulate emission
- Power density
- Noise

High Speed Small Passenger Car Diesel Engines

- All of the above, plus: Fast burn rate

3.1 Introduction

Particulates consist of soot particles, formed during the diesel combustion process, on which high molecular weight hydrocarbons and sulfates condense in the exhaust system. The soot particles form in the extremely fuel-rich zones of the burning fuel spray as the fuel molecules pyrolyze or break down and then form increasingly higher molecular weight polycyclic aromatics and polyacetylenes. These form the nuclei for soot particles which grow and agglomerate. A substantial fraction of the soot formed oxidizes within the cylinder. Hydrocarbons, and sulfates, then condense on the soot particles in the exhaust. Particulate reduction is being achieved through high-pressure and improved control of injection to reduce the fuel-rich zones in the spray, reductions in oil consumption to reduce the condensing HC, and control of fuel sulfur levels.

Figure 14 illustrates how various parts of the fuel jet and the flame affect the formation of soot (or particulates) during the “mixing-controlled” phases of diesel combustion in a direct injection engine with swirl.

Figure 14: Soot formation during the “mixing-controlled” phase

Soot forms in the rich unburned-fuel-containing core of the fuel sprays, within the flame region, where the fuel vapor is heated by mixing with hot burned gases. Soot then oxidizes in the flame zone when it contacts unburned oxygen, giving rise to the yellow luminous character of the flame.
3.2 Particulates Composition and Structure

Diesel particulates consist principally of combustion generated carbonaceous material (soot) on which some organic compounds have become absorbed. Most particulate material results from incomplete combustion of fuel hydrocarbons; some is contributed by the lubricating oil. The emission rates are typically 0.2 to 0.6 g/km for light-duty diesels in an automobile. In larger direct-injection engines, particulate emission rates are 0.5 to 1.5 g/brake kW·h. The composition of the particulate material depends on the conditions in the engine exhaust and particulate collection system. At temperatures above 500°C, the individual particles are principally clusters of many small spheres or spherules of carbon (with a small amount of hydrogen) with individual spherule diameters of about 15 to 30 nm. As temperatures decrease below 500°C, the particles become coated with adsorbed and condensed high molecular weight organic compounds which include: unburned hydrocarbons, oxygenated hydrocarbons (ketones, esters, ethers, organic acids), and polynuclear aromatic hydrocarbons. The condensed material also includes inorganic species such as sulfur dioxide, nitrogen dioxide, and sulfuric acid (sulfates).

![Figure 15: Photomicrographs of Diesel Particulates](image1)

![Figure 16: Spherules distribution](image2)

The structure of diesel particulate material is apparent from the photomicrographs shown in Fig.15 of particulates collected from the exhaust of an IDI diesel engine. The samples are seen to consist of collections of primary particles (spherules) agglomerated into aggregates (hereafter called particles). Individual particles range in appearance from clusters of spherules to chains of spherules. Clusters may contain as many as 4000 spherules. The spherules are combustion generated soot particles which vary in diameter between 10 and 80 nm, although most are in the 15 to 30 nm range. Figure 16 shows a typical distribution of spherule size (solid line) determined by sizing and counting images in the photomicrographs. The number mean diameter \( = \sum N_d d_i^2 / N \) is 28 nm. The volume contribution of these spherules is shown as the dashed curve in Fig.16. The volume-mean diameter is 31 nm. \( \left( = \sum N_d d_i^2 / N \right)^{1/3} \)
Determination of the particle size distribution with a similar technique involves assigning a single dimension to a complex and irregular aggregate, and introduces uncertainties arising from only having two-dimensional images of particles available. Other approaches based on inertial impactors and electrical aerosol analysers have been used. Some of the data suggest that the particle size distribution is bimodal. The smaller-size range is thought to be liquid hydrocarbon drops and/or individual spherules characterized by number-mean diameters of 10 to 20 nm; the larger-size range is thought to be the particles of agglomerated spherules characterized by number-mean diameters of 100 to 150 nm. However, other particulate samples have not shown a bimodal distribution; volume-mean diameters ranged from 50 to 220 nm with no notable trend with either speed or load.

The exhaust particulate is usually partitioned with an extraction solvent into a soluble fraction and a dry-soot fraction. Two commonly used solvents are dichloromethane and a benzene-ethanol mixture. Typically 15 to 30 mass percent is extractable, though the range of observations is much larger (~ 10 to 90 percent). Thermogravimetric analysis (weighing the sample as it is heated) produces comparable results. Dry soot has a much lower H/C ratio than the extractable material. Although most of the particulate emissions are formed through incomplete combustion of fuel hydrocarbons, engine oil may also contribute significantly. Radioactive tracer studies in a light-duty IDI diesel have shown that the oil was the origin of between 2 to 25 percent by mass of the total particulate and 16 to 80 percent of the extractable organic portion, the greatest percentages being measured at the highest engine speed studied (3000 rev/min). All of the oil contribution appeared in the extractable material. The contributions from the different individual compounds in the fuel have also been studied. All the compounds tested-paraffins, olefins, and aromatics-contributed to the particulate emissions; as a group, aromatics were the greatest contributors. Eighty percent of the carbon-14 used to tag individual fuel compounds was found in the insoluble fraction and 20 percent in the soluble particulate fraction.

In addition, trace amounts of sulfur, zinc, phosphorus, calcium, iron, silicon, and chromium have been found in particulates. Sulfur and traces of calcium, iron, silicon, and chromium are found in diesel fuel; zinc, phosphorus, and calcium compounds are frequently used in lubricating oil additives.

![Image of Diesel Particulates](image_url)

Figure 17: Lattice imaging of Diesel Particulates

A lattice image of a diesel particle is shown in Fig.17; it suggests a concentric lamellate structure arranged around the center of each spherule. This arrangement of concentric lamellas is similar to the structure of carbon black. This is not surprising; the environment in which diesel soot is produced is similar to that in which oil furnace blacks are made. The carbon atoms are bonded together in hexagonal face-centered arrays in planes, commonly referred to as platelets.
As illustrated in Fig.18, the mean layer spacing is 0.355 nm (only slightly larger than graphite). Platelets are arranged in layers to form crystallites. Typically, there are 2 to 5 platelets per crystallite, and on the order of $10^3$ crystallites per spherule. The crystallites are arranged with their planes more or less parallel to the particle surface. This structure of unordered layers is called turbostratic. The spherules, 10 to 50 nm in diameter, are fused together to form particles as shown in Fig.17. A single spherule contains $10^3$ to $10^6$ carbon atoms.

![Figure 18: Substructure of carbon particle](image)

A surface area of about 200 m$^2$/g has been measured for diesel soot. A smooth-surfaced 30-nm diameter sphere with a density of 2 g/cm$^3$ would have a surface/mass ratio of 100 m$^2$/g, so the measured value is about twice the superficial area. Approximating a particle of agglomerated spherules by a single sphere of 200 nm diameter gives a surface/mass ratio of 15 m$^2$/g. These data and estimates of superficial area per unit mass indicate that diesel soot has low porosity.

### 3.3 Soot Formation Fundamentals

The soot particles, whose characteristics have been described form primarily from the carbon in the diesel fuel. Thus, the formation process starts with a fuel molecule containing 12 to 22 carbon atoms and an H/C ratio of about 2, and ends up with particles typically a few hundred nanometers in diameter, composed of spherules 20 to 30 nm in diameter each containing some $10^5$ carbon atoms and having an H/C ratio of about 0.1.

Most of the information available on the fundamentals of soot formation in combustion comes from studies in simple premixed and diffusion flames, stirred reactors, shock tubes, and constant-volume combustion bombs. Moreover, the characteristics of diesel combustion which make it unsuitable for more fundamental studies—the high gas temperatures and pressures, complex fuel composition, dominance of turbulent mixing, the unsteady nature of the process, and the three-dimensional geometry—also make it difficult to interpret fundamental ideas regarding soot formation in the diesel context.

Soot formation takes place in the diesel combustion environment at temperatures between about 1000 and 2800 K, at pressures of 50 to 100 atm, and with sufficient air overall to burn fully all the fuel. The time available for the formation of solid soot particles from a fraction of the fuel is in the order of milliseconds. The resulting aerosol—dispersed solid-phase particles in a gas—can be characterized by the total amount of condensed phase (often expressed as the soot volume fraction, $F_v$, the volume of soot/total volume), the number of soot particles per unit volume ($N$), and the size of the particles (e.g., average diameter $d$). $F_v$, $N$, and $d$ are mutually dependent [e.g., for spherical particles $F_v = (\pi/6)Nd^3$], and any two of these variables characterize the system. It is most convenient to consider $N$ and $F_v$ as the independent variables since they each relate to the "almost-separate" stages of soot particle generation (the source of $N$) and soot particle growth (the source of $F_v$).
These stages can be summarized as follows:

1. Particle formation, where the first condensed phase material arises from the fuel molecules via their oxidation and/or pyrolysis products. These products typically include various unsaturated hydrocarbons, particularly acetylene and its higher analogues (C\(_{2n}H_2\)), and polycyclic aromatic hydrocarbons (PAH). These two types of molecules are considered the most likely precursors of soot in flames. The condensation reactions of gas-phase species such as these lead to the appearance of the first recognizable soot particles (often called nuclei). These first particles are very small (d < 2 nm and the formation of large numbers of them involve negligible soot loading in the region of their formation.

2. Particle growth, which includes both surface growth, coagulation, and aggregation. Surface growth, by which the bulk of the solid-phase material is generated, involves the attachment of gas-phase species to the surface of particles and their incorporation into the particulate phase. Figure 19, where the log of the molecular weight of a species is plotted against its hydrogen mole fraction \(\bar{x}_H\), illustrates some important points about this process. Starting with a fuel molecule of \(\bar{x}_H \geq 0.5\) it is apparent that neither purely polyacetylene chain growth nor purely PAH growth would lead to soot particles which have \(\bar{x}_H\) in the range 0.1 to 0.2. What is required is condensation of species with the right hydrogen content, or condensation of species with higher hydrogen content followed by dehydrogenation, or a combination of both these processes. Obviously some polyacetylenes and some PAH can satisfy these requirements, as can saturated platelets (e.g., C\(_{27}H_{27}\)). Surface growth reactions lead to an increase in the amount of soot \(F_S\) but the number of particles \(N\) remains unchanged. The opposite is true for growth by coagulation, where the particles collide and coalesce, which decreases \(N\) with \(F_S\) constant. Once surface growth stops, continued aggregation of particles into chains and clusters can occur.

![Figure 19: Soot formation paths](image)

These stages of particle generation and growth constitute the soot formation process. At each stage in the process oxidation can occur where soot or soot precursors are burned in the presence of oxidizing species to form gaseous products such as CO and CO\(_2\). The eventual emission of soot from the engine will depend on the balance between these processes of formation and burnout. The emitted soot is then subject to a further mass addition process as the exhaust gases cool and are diluted with air. Adsorption into the soot particle surface and condensation to form new particles of hydrocarbon species in the exhaust gases occurs in the exhaust system and in the dilution tunnel which simulates what happens in the atmosphere.
Figure 20: Net production of particulates process

Figure 20 illustrates the relationship between these processes. Although they are illustrated as discrete processes, there is some overlap, and they may occur concurrently in a given elemental mixture region within the diesel combustion chamber. Of course, due also to the nonhomogeneous nature of the mixture and the duration of fuel injection and its overlap with combustion, at any given time different processes are in progress in different regions or packets of fluid.

Two next sections will detail particles growth and adsorption/condensation phenomenon.

SOOT PARTICLE GROWTH

Nucleation produces a large number of very small particles with an insignificant soot loading. The bulk of the solid-phase material is generated by surface growth, which involves the gas-phase deposition of hydrocarbon intermediates on the surfaces of the spherules that develop from the nuclei. A qualitative description of the changes that occur as a function of time in a promised flame during nucleation and surface growth is shown in Fig.21. The soot fraction $F_v$, in units of soot volume per unit volume of gas, is related to the number density $N$ and the volume-mean diameter of the soot particles by

$$F_v = (\pi/6)Nd^3$$

$d$ is the actual diameter of the spherules, or the diameter of a sphere of equivalent volume to an agglomerated particle.
The rate of change of particle number density with time $t$ can be written

$$\frac{dN}{dt} = \dot{N}_n - \dot{N}_a$$

where $\dot{N}_n$ is the rate at which fresh nuclei appear and $\dot{N}_a$ is the rate of agglomeration of spherules or particles that collide and stick.

![Figure 21: Variations with time in a flame](image)

At the peak of the $N$ curve, $\dot{N}_n = \dot{N}_a$. To the left of the peak, $\dot{N}_n > \dot{N}_a$, the particle diameter remains essentially constant at the minimum detectable diameter and the (small) rise in soot volume is dominated by nucleation. To the right of the peak in the $N$ curve, $\dot{N}_a > \dot{N}_n$. The number of agglomerating collisions is high because of the high number density; at the same time nucleation ends because there is enough dispersed surface area for gaseous deposition of hydrocarbon intermediates so the probability of generating new nuclei falls to zero. With nucleation halted slightly to the right of the $N$ curve peak, all the subsequent increase in soot volume fraction (the majority) stems from surface growth. To the right of the $N$ curve peak, the number density falls in the cast illustrated by three orders of magnitude. This is the result of agglomeration, which is responsible for a portion of the increase in particle diameter.

Agglomeration does not contribute to the rise in soot volume fraction, $F_v$. Surface growth that takes place on nuclei and on spherules is responsible for forming the concentric shells (somewhat distorted and warped) that constitute the outer portions of spherules and which are distinct from the less-organized spherule enter (see Figs. 17 and 18). Surface growth on agglomerated particles may partly fill in the crevices at the junctures of adjoining spherules to provide the nodular structure evident in Fig. 17.

Once particles have formed, interparticle collisions can lead to agglomeration, thereby decreasing the number of particles and increasing their size. Three types of agglomeration have been identified in soot formation. During the early stages of particle growth, collision of two spherical particles may result in their coagulation into a single spheroid. This is easy to visualize in hydrocarbon pyrolysis where the beginnings of a soot particle may have the viscosity of a tarry liquid. Also, when the individual particles are small, rapid surface growth will quickly restore the original spherical shape. This process occurs up to diameters of about 10 nm.
On the other hand, if spherules have solidified before collision and surface growth rates have diminished, the resulting particles resemble a cluster in which the original spherules retain much of their individual identity. After surface growth essentially ceases, continued coalescence of the soot particles results in the formation of chainlike structures of discrete spherules. This suggests electrostatic forces are significant. Positive charge measured on these particle chains is claimed to be the cause of their chainlike structure. This latter coalescence once surface growth ceases is termed aggregation.

It has been shown experimentally that during coagulation the rate of decrease of particle number density was proportional to the product of a coagulation coefficient and the square of the number density:

\[-\frac{dN}{dt} = KN^2\]

Throughout the soot formation process in a flame, the H/C ratio of the hydrocarbons formed in the pyrolysis and nucleation process and of the soot particles continually decreases. The H/C ratio decreases from a value of about 2, typical of common fuels, to of order 1 in the youngest soot particles that can be sampled, and then to 0.2 to 0.3 once surface growth has ceased in the fully agglomerated soot. The latter stages of this process are indicated in Fig. 21. The addition of mass to the soot particles occurs by reaction with gas-phase molecules. The reacting gas-phase hydrocarbons appear to be principally acetylenes, with larger polymers adding faster than the smaller. Small polycyctenes undergo further polymerization in the gas phase, presumably by the same mechanism leading to nucleation. As a result of preferential addition of the larger polymers, the H/C ratio of the particles decreases toward its steady-state value. Thus most of the polycyctenes added must be of very high molecular weight or dehydrogenation must also take place.

**ADSORPTION AND CONDENSATION**

The final process in the particulate formation sequence illustrated in Fig. 20 is adsorption and condensation of hydrocarbons. This occurs primarily after the cylinder gases have been exhausted from the engine, as these exhaust gases are diluted with air. In the standard particulate mass emission measurement process this occurs in a dilution tunnel which simulates approximately the actual atmospheric dilution process.

The effect of the dilution ratio on collected particulate mass is shown in Fig. 22 for a standard dilution tunnel, where the total sample is partitioned into extractable and nonextractable fractions.

![Figure 22: Effect of dilution ratio on particulate mass](image)

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The nonextractable fraction is the carbonaceous soot generated during combustion and is not affected by the dilution process. With no dilution (dilution ratio of unity) the difference between the total and nonextractable mass is small; the bulk of the extractable fraction is acquired after the exhaust gas is mixed with dilution air. Extensive studies of this dilution process have shown that both adsorption and condensation occur. Adsorption involves the adherence of molecules of unburned hydrocarbons to the surfaces of the soot particles by chemical or physical (van der Waals) forces. This depends on the fraction of the available particle surface area occupied by hydrocarbons and on the partial pressure of the gaseous hydrocarbons that drives the adsorption process.

As the dilution ratio increases from unity, the effect of decreasing temperature on the number of active sites dominates and, as shown in Fig. 22, the extractable fraction increases. At high dilution ratios, the sample temperature becomes insensitive to the dilution ratio but the decreasing hydrocarbon partial pressure causes the extractable mass to fall again. Condensation will occur whenever the vapor pressure of the gaseous hydrocarbon exceeds its saturated vapor pressure. Increasing dilution decreases hydrocarbon concentrations and hence vapor pressure. However, the associated reduction in temperature does reduce the saturation pressure. High exhaust concentrations of hydrocarbons are the conditions where condensation is likely to be most significant, and the hydrocarbons most likely to condense are those of low volatility. Sources of low-volatility hydrocarbons are the high-boiling-point end of the fuel, unburned hydrocarbons that have been pyrolyzed but not consumed in the combustion process, and the lubricating oil.

Experiments with a passenger car IDI diesel, where the oil was tagged with a radioactive tracer, have shown that the oil can contribute from 2 to 25 percent of the total particulate rows, with the greatest contribution occurring at high speed. On average, over half of the extractable mass was traceable to the oil. All the material traceable to the oil was found in the extractable fraction, indicating that the oil did not participate in the combustion process. However, the oil is not always a significant contributor: in another engine and tests, fuel was the dominant source of extractable material.

4: Main known parameters and reduction strategies

4.1 Introduction

COMPOSITION

From the previous development, one can summarize that Diesel particulates are mainly comprised of carbon nuclei impregnated with various heavy hydrocarbons and sulfate, produced during incomplete combustion. Sulfate derives from sulfur in the diesel fuel.

PARAMETERS

The parameters affecting soot and particulates formation can be classified into 3 categories.

1: The engine

The proportion of each elements varies according to

- the engine technology (DI, valves, injectors, ...)
- the engine control strategy (cold start, multi-injections, ...)
- its use (load, speed, driving cycles, driver, ...)
- its maintenance conditions (ageing, wearing, sooting, ...)
2: The fuel

The sensitive fuel property appears to be the sulfur content.

3: The lubricant

As previously detailed, lubricant consumption generates soot which is not participating to the combustion process. These particulates are found in the extractable mass.

4.2 Parameters review

4.2.1 The engine

As established previously, soot is mainly coming from rich mixture areas, where combustion is not complete.

Combustion process:

At the moment, technologies are focusing on air motion improvement, increasing air/fuel mixing, etc. These improvements are reached by working on swirl and tumble generation (multi-valves, air ducts, deflectors, piston and combustion chamber shapes, valve timing), and on turbulence generation (all the previous one, plus fuel spray pattern, high injection pressure, smaller holes and finally high injection velocities). The higher injection pressures lead to smaller fuel droplets, enhancing the mixing process, and reducing particulates emission rates. Knowing that particulates are subject to aggregate and agglomerate after the combustion, the particulates size reduction that is observed is not a direct consequence of the droplet size diminution, but a combination of all the aforementioned enhancements.

From size reduction derives filtration difficulties and lead to particulates lungs penetration (health effects for suspected particulates size between 0.03 and 10 μm). For a given size distribution, particulates number is also decreasing.

Engine control:

By adopting Common Rail Diesel Fuel Injection System, new generations of small Diesel engines, dedicated to passenger cars, are more accurately controlled, with precise fuel metering.

The common rail system is at the forefront of diesel fuel injection technology. The system consists of a fuel rail accumulator, injectors with rapid action solenoid valves and an electronic control system that separates the pressure generation and injection processes to abolish mutual constraints. This technology is capable of producing injection pressures of up to 1600 bar. The Electronic Control Unit (ECU) regulates the combustion process by selecting the optimum fuel line pressure, injection timing and fuel flow duration from the stored maps. Further Common Rail generations, with even higher pressure potential’s for further improvements in emission levels benefiting the environment, are under development. These new systems utilise piezo actuators instead of solenoid valves and permit even more precise control of multiple injection pulses.
Cold start conditions lead to Diesel particulates with higher size (see reference [1]), that enhanced engine control can contribute to decrease (higher injection pressure).

By changing injection timing (by advancing), a substantial part of the particulates can be reduced, but with NOx increase as counterparts. (see reference [3])

Exhaust Gases Recirculation is a common technical option to reduce NOx emissions by affecting ignition delay, heat capacity of the intake charge, and dilution of the intake charge with inerts. In this originates lower peak combustion temperatures, reducing NOx, but reducing soot oxydation rates in the same time. And thus, leading to higher soot production. A way to control and reduce this effect can be reached by pre and multi-injection strategies implementation. (see reference [3])

EGR application presents undesirable effects: engine wear, due to abrasion by the particulate present in the intake air, and lubricant contamination, due to sulfuric acid present in the recirculated gases which tends to break down the lubricating oil.

**Engine use:**

The engine use concerns primarily load and speed variations, applied by the driver, following the road and traffic conditions. From the facts that load is tuned by A/F ratio on Diesel engines, and that air motion and air/fuel mixture are the main parameters to obtain a combustion process as complete as possible, one can derive that:

- Rapid transients, particularly rapid acceleration at high load, are driving patterns that produce soot, by relative and temporary lack of oxygen.
- At the other hand, at idle or light load and speed, it is possible to collect more soot than at medium speed and load, due to insufficient air motion and turbulence.

**Engine mileage and maintenance:**

Engine mileage primarily presents wearing and leakage phenomena, as well as deposits and sooting-up processes. Wear affects also oil properties, and no replacement of damaged oil leads to higher wearing. As seen previously, EGR application accelerates wear and oil degradation.

Leakage at valve’s guides and piston rings provides oil in the combustion chamber that generates soot, which is not participating to the combustion process. These particulates are found in the extractable mass. (see adsorption and condensation paragraph)

4.2.2 The fuel

Sulfur content of Diesel fuels affects the level of particulates emissions at two stages. The first one during the formation process, knowing that soot is made of carbon nuclei impregnated with various heavy hydrocarbons and sulfate, and sulfate derives from the fuel sulfur. By decreasing sulfur level in the fuel, the sulfate impregnation is lowered as well. Note that recent studies (see reference [2]) have shown that low sulfur level affects primarily the small size particulates (nanoparticles < 0.030 μm). The second stage concerns exhaust after-treatment, especially oxydation catalyst, that reduce HC and CO emissions, and that remove hydrocarbons from particulates. These oxydation catalysts are sensitive to the fuel sulfur content. The conversion efficiency of 30% for present sulfur content (300 ppm) raises to 70-80% with a low sulfur fuel (~30 ppm). Concerning after-treatment, particulates filters and traps (see after) are also affected by the sulfur content, while it seems that it affects the required temperature for trap regeneration.
4.2.3 The lubricant

Particulates coming from lubricant originate in:

- the thin oil layer in the cylinder
- the leakage at the valve’s guides
- the leakage at the turbocompressor shaft
- the crankshaft oil vapor recovery system

All these potential sources are linked to mileage and engine wearing.

4.3 Reduction strategies

4.3.1 Introduction

The limitation of soot generation is performed at two levels. The first one focuses on combustion process and aims at controlling the particulates emissions inside the combustion chamber and just after, in the exhaust ports and lines (source limitation). This step (except for the oxidation catalyst) is illustrated just after with a flowchart (figure 23). The second step focuses on the after-treatment of emitted gases.

4.3.2 Source limitation

**PARTICULATES REDUCTION STRATEGIES**

- **Reduction of INSOF**
  - Reduction of sulfate
    - Use of low sulfur fuel
    - Optimization of air motion in cylinder
    - Optimization of combustion chamber shape
  - Reduction of soot
    - Improvement of diffusion combustion
    - Improvement of injection timing and rate

- **Reduction of SOF**
  - Reduction of oil SOF
    - Improvement of lubricating oil consumption
  - Reduction of fuel SOF
    - Reduction of conversion from oil to SOF
    - Use of oxidation catalyst

Figure 23

**SOF**: Soluble Organic Fraction
**INSOF**: Insoluble Organic Fraction
4.3.3 After-treatment

This section is summing-up the last particulates emission systems on the market, with a short characteristics description.

DIESEL PARTICULATES TRAP (DPT)

An exhaust treatment technology that substantially reduces diesel engine particulate emissions is the trap oxidizer. A temperature-tolerant filter or trap removes the particulate material from the exhaust gas: the filter is then "cleaned off" by oxidizing the accumulated particulates. This technology is difficult to implement because: (1) the filter, even when clean, increases the pressure in the exhaust system; (2) this pressure increase steadily rises as the filter collects particulate matter; (3) under normal diesel engine operating conditions the collected particulate matter will not ignite and oxidize; (4) once ignition of the particulate occurs, the burnup process must be carefully controlled to prevent excessively high temperatures and trap damage or destruction. Trap oxidizers have been put into production for light-duty automobile diesel engines. Their use with heavy-duty diesel engines poses more difficult problems due to higher particulate loading and lower exhaust temperatures.

Types of particulate filters include: ceramic monoliths, alumina-coated wire mesh, ceramic foam, ceramic fiber mat, woven silica-fiber rope wound on a porous tube, and also metallic fiber support. Each of these has different inherent pressure loss and filtering efficiency. Regeneration of the trap by burning up the filtered particulate material can be accomplished by raising its temperature to the ignition point while providing oxygen-containing exhaust gas to support combustion and carry away the heat released. Diesel particulate matter ignites at about 500° to 600° C. This is above the normal temperature of diesel exhaust so either the exhaust gas flowing through the trap during regeneration must be heated (positive regeneration) or ignition must be made to occur at a lower temperature with catalytic materials on the trap or added to the fuel (catalytic regeneration). Catalytic coatings on the trap reduce the ignition temperature by up to 200° C.

DIESEL OXYDATION CATALYST (DOC)

DOCs reduce HC, CO and SOF (soluble organic fraction) by oxidation over a precious metal catalyst. A concern with higher precious metal loading is the DOC’s tendency to convert SO2 in the exhaust gases to sulfates. This is an additional reason to adopt low sulfur Diesel fuels.

5: Last trends

The trends can be separated as previously into combustion process technologies and after-treatment technologies.

5.1 Combustion process

Up-to-date technologies for high specific power Diesel engines leading to less particulates, for present and future power units, are given here below.
• Turbocharging and Intercooling
• High compression ratio
• High spray and charge turbulence
• 4 valves per cylinder: concentric mixture formation
• Enhanced fuel injection systems: high pressures, electronic control, common rails
• Multi-injections strategy
• Cooled EGR, with a dynamic control
• Inlet swirl control by port deactivation

In addition, last trends concerns downsizing, air/fuel mixture closed loop control with a lambda sensor, and the development of a combustion process control, mixing the two characteristics of Diesel and gasoline engine (ignition driven by spark and pressure).

DOWNSIZING

Diesel power units will progressively gain a sport character, with high compression ratio, high injection pressures, light structures, and high revolution speeds. This trend will be carried out through downsizing and turbocharging strategies, leading to small displacements per cylinder and high power output per unit of displacement (300 cm³/cylinder, more than 70 kW/dm³ of displacement).

CLOSED LOOP CONTROL

10 years ago, Diesel engine control was assumed by 8 bits processor. Nowadays, Diesel engines are driven with 32 bits processors. With the help of new air/fuel ratio sensors, dedicated to Diesel units, it is now possible to control and measure in real time the injected mass of fuel, on each cycle, with a closed loop regulation. Combined with EGR dynamic control strategies, these new generations of compression ignited engines show drastic reductions of fuel consumption and emission.

‘PIESEL’ FUELS (for Petrol and Diesel mixed fuels)

An advanced research area originates in the fact that gasoline and Diesel engine control are beginning to merge: direct injection, common rail and dynamic injectors electronic control, similar combustion process in excess of air (lean burn engines). With the help of electronic and gaseous fuels like CNG (compressed natural gas), it seems possible to combine the advantages of the two type of power unit into a single ‘hybrid’ engine, ignited by compression and spark.

5.2 After-treatment

Here below, the present and future after-treatment technologies.

• Oxydation catalyst with low light-off temperatures
• DENOX catalyst
• Particulate trap

In addition, the use of plasma technologies to reduce particulates is under development.
PLASMA TECHNOLOGIES

Development of efficient techniques for dealing with passenger car diesel exhaust particulates is on the track, and a UK based company is developing the Electrocat Diesel Particulate Filter (DPF), which it says operates at low exhaust temperatures, is self-cleaning, does not require fuel additives, and is apparently unaffected by sulfur levels in fuel. This DPF is described as a self-cleaning system, using plasma filter regeneration to remove all types of carbonaceous particulates, including ultrafines, as well as gaseous hydrocarbons including suspected carcinogenic polyaromatic hydrocarbons.

The system uses proprietary Electrocat technology that employs a gas plasma. A plasma is a gas that has been at least partially ionized by having an electrical current passed through it, becoming chemically reactive because of the electrons and free radicals it contains. This reactivity can be engineered to perform chemical reactions under low-temperature conditions (non-thermal plasmas) in which reactivity would not normally be expected. This property makes plasma particularly suitable for the environmental treatment of gases so that, for example, an organic contaminant can be removed from a gaseous waste stream, such as an engine's exhaust, without the need for thermal or catalytic oxidation.

As this new technology needs high voltages to generate gas plasma, next generation of starter/alternator delivering 42 V will be helpful to facilitate the application.

6: Perspectives and Conclusions

After having built this study on particulates formation processes, a number of important issues have to be screened.

The first one concerns particulates size.

The study shows that particulates tend to decrease in number and size, as a consequence of combustion process and technologies improvement. The reduction in number is welcome. But the reduction in size leads to filtration difficulties and health problems (lung penetration). Another issue is to fix a limit for regulation thresholds. At the moment, PM$_{10}$ (particulates less than or equal to 10 $\mu$m in aerodynamic diameter) is the regulation limit, but some recent studies show strong effects and health correlation on animals for PM$_{2.5}$ [1], [2].

To continue, an important investigation field still remain uncoordinated, namely the problem of particulates measurement devices and procedures. Indeed, as the particulates emission standards are now scheduled, involved staffs by manufacturers and oil companies are daily confronted to the following fact:

Same instrumentation gives different results, function of the test procedures.

In the next future, particulates measurements facilities, techniques, and procedures will have to be assessed and co-ordinated [2].

Concerning Diesel fuels, and linked to the size problem just evoked, low sulfur fuels are expected by cars and trucks manufacturers. The standards are now around 300 ppm. Acceptable sulfur levels are targeted between 10 and 30 ppm, leading to efficient small sized particulates reduction. Theses standards are yet applied in Sweden and Japan.
Another issue is linked to filter regeneration, especially for heavy duty Diesel power unit. These kind of engines are supposed to work at high load (particulates emission running conditions) for long periods. It needs to develop enhanced strategies to avoid filter saturation, like active regeneration, with or without additives in the fuel, and trap catalytic coatings.

Last but not least, a galore of new sensitive technologies are now under development and close to be on the market. But a large effort is still to be made, to assess the production consistency and reliability, delivering low emission vehicles (LEV) from the beginning (reception), and that remain clean until the end of service life. Without too much maintenance sessions.

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[9] A Theoretical Investigation on the Effects of Combustion Chamber Geometry and Engine Speed on Soot and NOx Emissions

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In proceedings of the 4th INTERNATIONAL CONFERENCE ICE99
September, 12th - 16th 1999 - Napoli

[10] Diesel Emission Control – Sulfur Effects

National Renewable Energy Laboratory
U.S. department of energy
In Automotive Engineering International December 2000 ( SAE periodic )


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SAE Paper 980411

[12] Diesel Engine Smoke Measurements in the Rapid Acceleration Test

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SAE Paper 980411

Automotive Engineering (SAE), Engine Technology Int., Focus (AVL), Ingénieurs de l’Automobile (SIA), Global Automotive Network (FISITA), Automotive World, European Automotive Design.
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