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ABSTRACT

The diesel engine is the most efficient user of fossil fuels for vehicle propulsion and seems to best fulfill the requirements of the future. It is for this reason that Volkswagen has initiated a very broad research programme for diesels.

The purpose of this paper is to build a bridge between fundamental research and technical developments which could allow evaluation of the prospects of direct-injection diesels as powerplants of choice for passenger cars in the turn of the century.

The current knowledge on mixture formation, combustion and pollutant formation in diesel engines is presented and discussed with special emphasis given to the concept of the direct-injection diesel engine.

1. INTRODUCTION

The future development of engines for motor vehicles is expected to be based on a compromise between the following important criteria [1]:

* As far as performance is concerned, the focus will shift towards the accelerating capability and driving flexibility whilst at the same time retaining current top speeds.

* Concerning fuel efficiency, one of the consequences will be that as the price of fuel rises, more and more costly measures for lowering consumption will pay off. Developments should concentrate on reducing fuel consumption at part-load engine operation.

* In urban areas, engine emissions should not reach concentrations that are likely to damage people's health and the environment.

* Research into future powertrain units should aim at achieving the lowest possible levels of exterior noise.

The diesel engine is the most efficient user of fossil fuels for vehicle propulsion and seems to best fulfill the requirements of the future. Therefore, Audi and Volkswagen support the development of diesel engines, because of four good reasons: they are efficient, economical, environmentally friendly, and future-oriented.

The advantages in fuel consumption and exhaust emissions offered by diesels and above all by the direct-injection diesel have been presented and discussed in [2] in view of their implications for future development. The challenges facing diesel engines have been approached by setting-up a very broad research programme named IDEA [3,4] in order to improve the fundamental understanding of the complex diesel combustion process, to define the potential for further improvement and to form the basis for a modern diesel engine technology [5].

The purpose of this paper is to discuss the latest fundamental understanding of the diesel combustion process and the consequences for further development of diesel engines.

2. COMPARISON OF THE EFFICIENCIES OF DIESEL AND GASOLINE ENGINES

There are two combustion systems in production today: the gasoline and the diesel engine. A consideration and comparison of their thermodynamical processes may explain their fundamental differences.

The efficiency of the gasoline engine remains constant with mean effective pressure (fig.1a) and is reduced by scavenging losses, friction losses and losses caused by incomplete combustion. With increasing load the efficiency becomes higher.

On the other hand the efficiency of the prechamber diesel engine (fig. 1b) decreases with increasing mean effective pressure because of the Seilinger cycle. The theoretical efficiency value is reduced by friction and incomplete combustion losses but the scavenging loss is nearly zero.

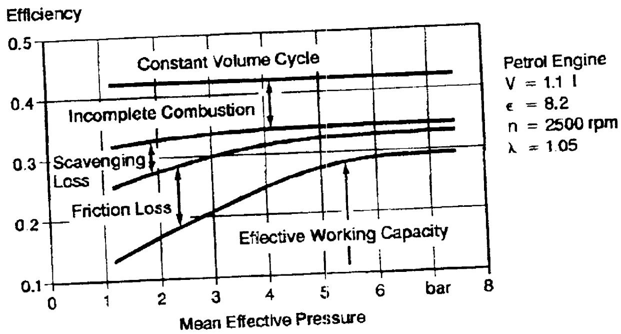


Fig. 1a: Gasoline engine efficieny

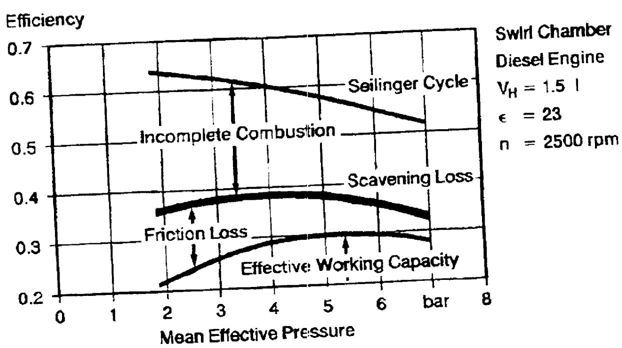


Fig. 1b: Diesel engine efficiency

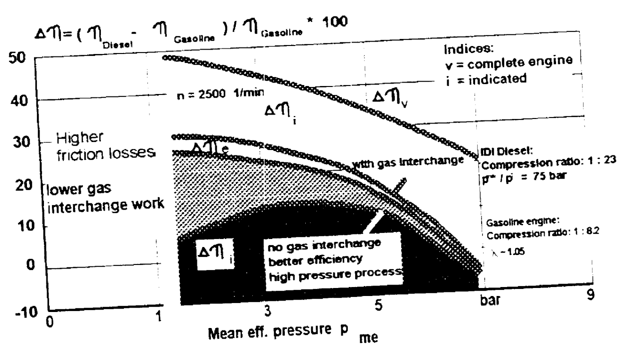


Fig. 1c: Comparison of efficiencies of gasoline and diesel engines

Direct comparison between gasoline and diesel engines (fig. 1c) shows a decrease of the advantages of the diesel process with increasing mean effective pressure. This advantage is obviously reduced by incomplete combustion and friction losses which are lower for the gasoline engine. However, the high scavenging losses of the petrol engine result in up to 25% better efficiency for diesels. Although under real driving conditions the fuel consumption benefit of a diesel vehicle may even reach 30%. The modern turbocharged direct injection diesel (TDI) can improve the fuel economy by a further 15% (resulting in up to 45%).

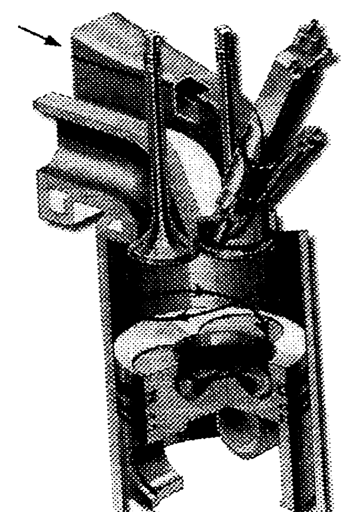
The above analysis demonstrates that there is a high potential for further improvement of the diesel engine. Prerequisite is a deeper understanding of the mixture formation and diesel combustion process. The IDEA programme proved to be an important step in this direction [3,4] by providing valuable insight into the air flow, diesel sprays, auto-ignition, combustion and pollutant formation mechanisms in the engine cylinder.

3. FUEL INJECTION AND MIXTURE FORMATION

3.1 Air Motion

Diesel engines employ helical and/or directed ports in order to generate a swirling motion in the cylinder during the compression stroke (fig. 2); at around 330-340 a TDC the swirling air is forced towards the piston-bowl as a result of the squish motion generated above the piston crown. The competing action between swirl and squish combined with the conservation of angular momentum of swirl give rise to high swirl velocities inside the piston-bowl and high turbulence levels near the bowl entry plane.

Swirl Port



Piston Bowl

Fig. 2: Swirl generation with a helical port

Measurements of the spatial distribution of swirl at different crank angles during intake obtained with laser Doppler velocimetry (LDV) in a VW optical 1.9L TDI engine equipped with helical ports [6] have revealed the presence of two counter-rotating jets of unequal strength whose interaction lasts for most of the stroke (fig. 3a). It is around BDC that the weaker vortex ceases to exist and the flow starts developing towards a single vortex occupying the whole diametrical plane. The r.m.s. of the velocity fluctuations are very high during early induction, due to the interaction of the counter-rotating jets, but decay to levels of about 1 $\overline{V}_p$ ($\overline{V}_p$ = mean piston speed) as the mean flow becomes more ordered around BDC. During the compression stroke swirl profiles are asymmetric with the centre of rotation performing a helical motion in space and in time and the swirl velocity distribution becoming nearly flat close to the wall while resembling solid-body motion around the cylinder axis. Towards the end of the stroke and around the time of fuel injection, the onset of squish encourages the centre of swirl to move towards the off-centre piston-bowl while the velocity fluctuations are nearly homogeneous in space, with values of 0.5-0.6 $\overline{V}_p$, with the exception of the bowl entry plane near the lip where turbulence can reach values in excess of 1 $\overline{V}_p$ (fig. 3b).

3.2 Fuel Injection and Sprays

In order to achieve good mixture formation the fuel must be supplied to the combustion chamber via the injection nozzle at high pressures (presently for passenger car engines up to 1000 bars, but in case of direct injection diesel higher than this in the near future up to 2000 bars). In addition an extremely precise fuel-metering system is necessary (milligrams of injected mass) to prevent excessive soot levels. In order to achieve an optimal and constant operation of the diesel engine, the start of injection must occur with a precision in the millisecond range. This is very important because load and speed control of the diesel is maintained via the injected fuel quantity.

Today's passenger car diesel engines are equipped in most cases with distributor-type fuel injection pumps. Unit injector (fig. 4a) and common rail systems (fig. 4b) represent the future trend towards high pressure injection especially for direct injection diesels and are presently under intense investigation.

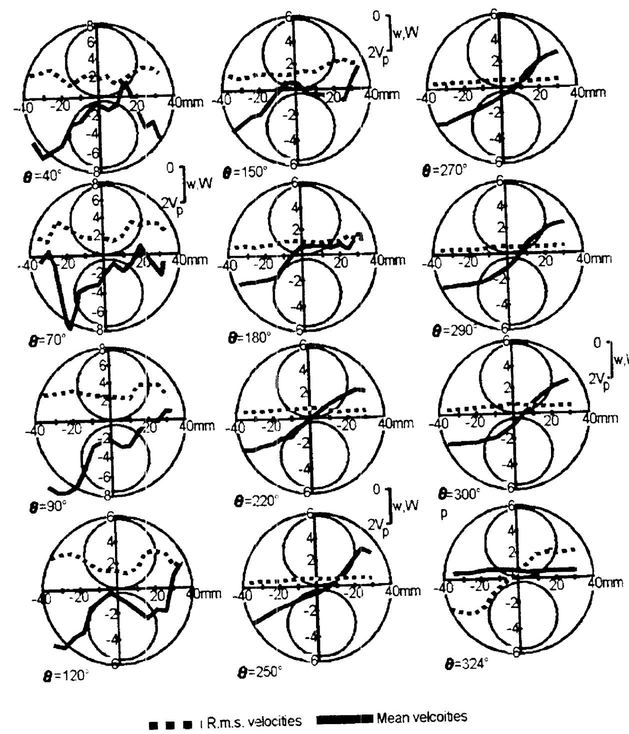


Fig. 3a: Flow in a VW 1.9l TDI engine: Counter-rotating jets during intake

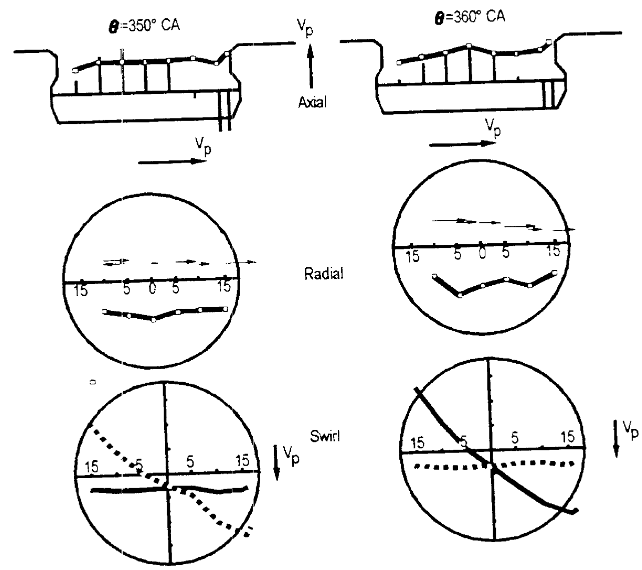


Fig. 3b: Flow in a VW 1.9l TDI engine around start of injection

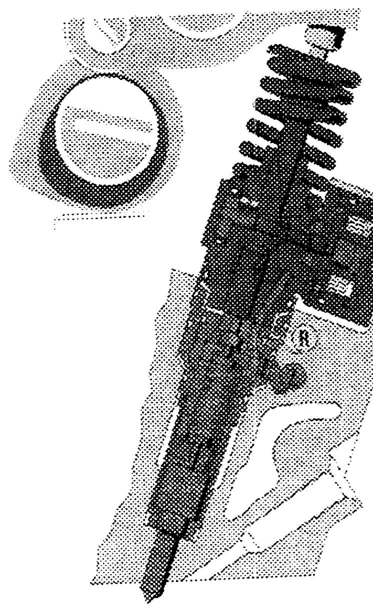


Fig. 4a: High pressure injection equipment: Unit injector

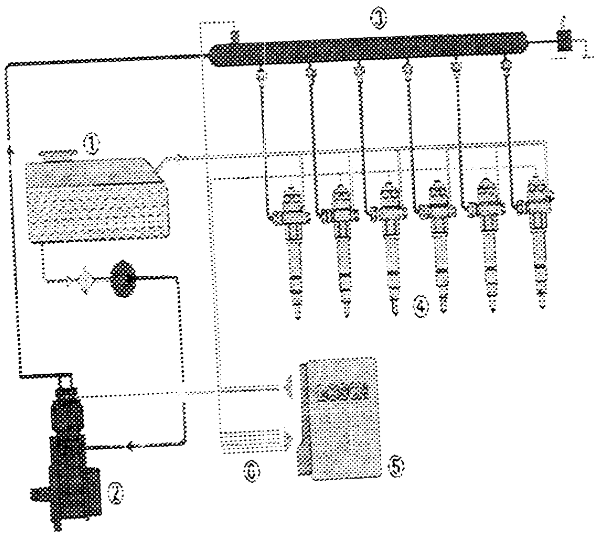


Fig. 4b: High pressure injection equipment: Common rail system

Prerequisite for an effective application of fuel injection systems is its understanding. Therefore simulation programs have been developed and validated [7, 8] which are capable of predicting the pressure and 1-D flow distribution within the pump, high pressure line and nozzle (fig. 5a).

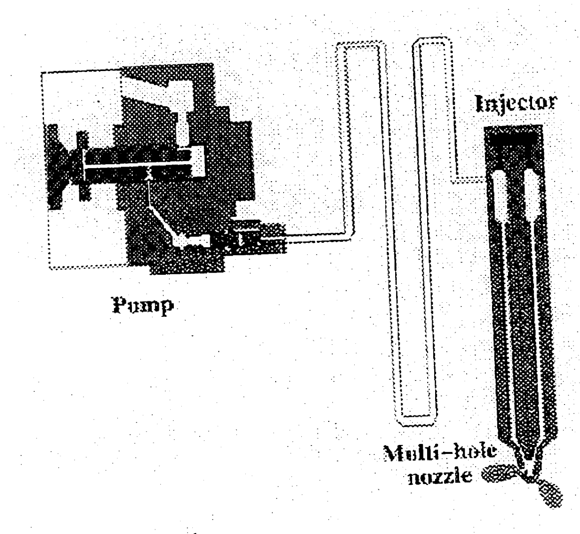


Fig. 5a: Simulation of the fuel injection: equipment

More recently, phenomenological 1-D simulation models of diesel fuel injection systems have been extended to include aspects of cavitation in the sac volume and holes and their effect on fuel atomisation [9]. These calculations have revealed that the dominant parameters influencing spray development and the mixing of fuel with the surrounding air are hole cavitation, injection velocity and droplet deformation (fig. 5b).

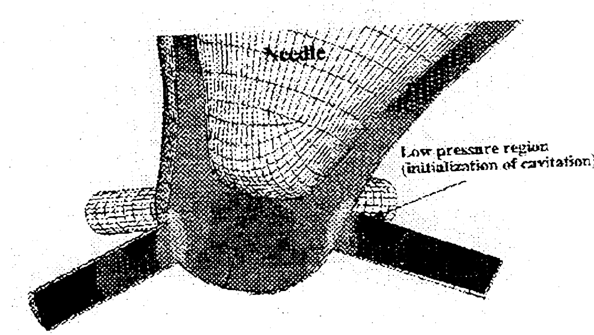


Fig. 5b: Simulation of the fuel injection: nozzle flow

Summarising, the relevant processes for the description of fuel sprays (fig. 6) are:

* nozzle flow phenomena including inlet hole geometry, diameter and length, cavitation, injection rate and number of holes,
* liquid core atomisation, droplet break up, collisions, coalescence, and turbulent dispersion,
* droplet deformation and evaporation,
* interaction with the gas phase (momentum, mass and heat exchange),
* interaction with the combustion chamber wall,
* chamber gas density and temperature and
* hole-to-hole variation in inclined injectors.

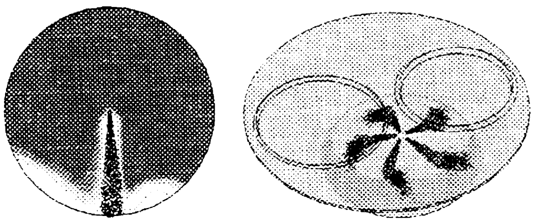


Fig. 6: Fuel spray propagation and interaction with combustion chamber wall in a 1.9l TDI engine

The recent trend towards four-valve cylinder heads will allow improved mixing between the fuel injected from the vertical, centrally-located nozzle towards the axisymmetric piston-bowl with much reduced hole-to-hole spray variations and swirl angular momentum losses.

Concerning mixing of the injected sprays with the surrounding swirling air, this is governed by the droplet size and velocity characteristics, the penetration rate of the spray, the distance between the nozzle and the piston-bowl wall, the wall temperature and the strength (velocity distribution) of swirl. For the case of two-stage injection (with pre-injection), obviously the amount of fuel injected in the first phase will control noise and NO_x levels but the timing of the second peak is equally important for soot formation and oxidation.

4. AUTO-IGNITION

During the compression stroke intake air is compressed to 30 - 60 bars and thereby its temperature increases to 700 - 900 °C. This temperature is sufficient to induce auto-ignition in the non-homogeneous turbulent mixture (fig. 7). Because the fuel auto-ignites, it must have a high ignition quality which is mainly characterized by a high cetane number (higher than 50 and towards 58). Auto-ignition sites in direct-injection diesels are usually located downstream of the spray axis in the direction of swirl and are concentrated near the wall especially at high loads.

Fig. 7: Auto-ignition of a fuel spray inside the combution chamer of a 1.9l TDI engine

The theoretical description of the auto-ignition process for non-premixed fuels under diesel conditions includes an extensive list of elementary reactions (for example: 1011 reactions with 171 species for the component n-heptane). To handle this problem, kinetic mechanisms based on only few representative and global reactions have been established. This reduced kinetic scheme still retains the essential chemical kinetic information while at the same time makes combustion calculations in CFD codes possible (fig. 8).

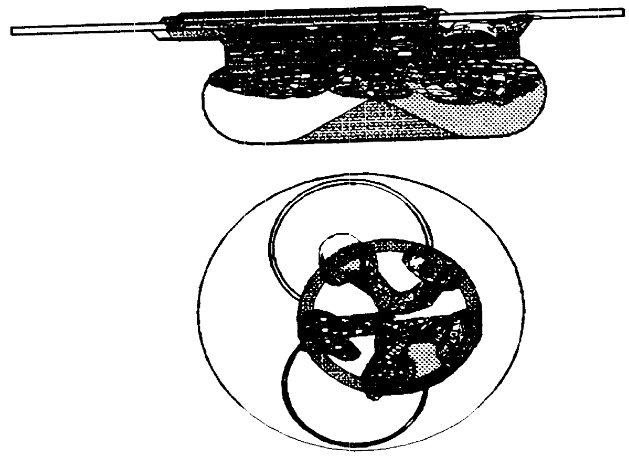


Fig. 8: CFD simulation of combustion in a 1.9l TDI engine

5. PHASES OF DIESEL COMBUSTION

According to Pischinger [10] the fundamentals of the diesel combustion process can be analyzed from the states of the mixture and the burned fuel and the transition between them. The key problem in the optimization of diesel combustion is the reduction of soot formation while maintaining low NO_X levels. Recent flame investigations have shown that soot is only formed at temperatures exceeding 1500 K and air/fuel ratios λ below 0.6 [11].

In figs. 9a-c the states of the gases at different times during the combustion process are indicated as a function of the air/fuel ratio λ and the gas temperature. The areas of possible soot formation and the concentration plots for NO_X formation are presented, as well as the states of the air/fuel mixture before auto-ignition and of the burned fuel assuming adiabatic combustion. The first conclusion is that to prevent the formation of soot and considerable amounts of NO_X during combustion, the mixture should have an air/fuel ratio of $0.6 < \lambda < 0.8$. The diesel combustion process can thus be divided into three different phases:

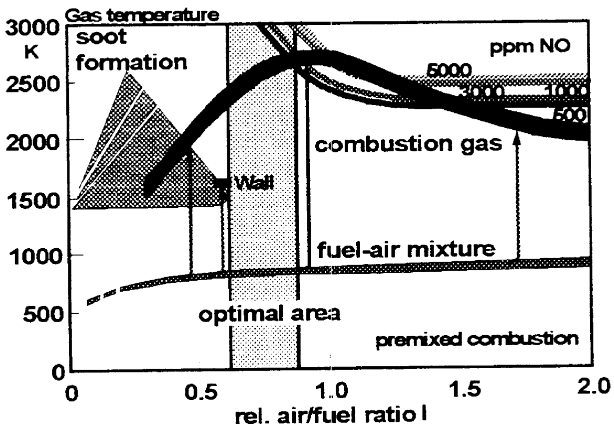


Fig. 9a: First phase of diesel combustion

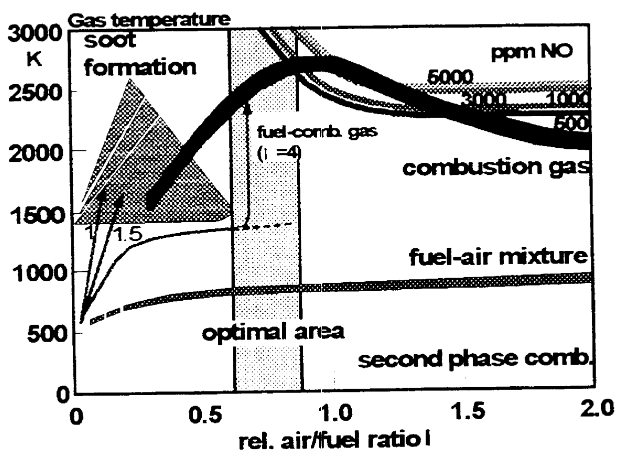


Fig. 9b: Second phase of diesel combustion

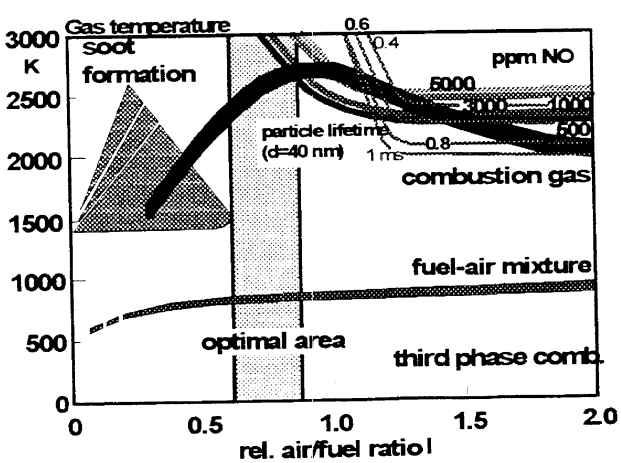


Fig. 9c: Third phase of diesel combustion

1. Premixed combustion (fig. 9a): During mixture formation the fuel is injected into the cylinder which is filled with highly compressed air. Depending on ignition delay and the injected mass, a considerable amount of fuel evaporates and mixes with air. Measurements have confirmed that auto-ignition starts in premixed zones with $\lambda \approx 0.7$.

2. Main phase of combustion (fig. 9b): Following auto-ignition the remaining fuel is injected into burning or burned gas and combustion takes place as a partially premixed diffusion flame. This means that most of the fuel burns like a gaseous layered mixture cloud in which areas with different air thermodynamic conditions are present. The extent of these areas depends on ignition delay and the burning rate of the diffusion flame of the turbulent mixture. By mixing with combustion gas (not with fresh air), mixture elements may reach high temperatures and low air/fuel ratio states at the same time. These conditions yield to a high soot formation. A fuel element that mixes with combustion gas of about $\lambda = 4$ may be able to avoid the soot threshold of $\lambda = 0.7$ during mixing and burning.

3. Burn-out (fig. 9c): During diesel combustion a burn-out of just formed soot particles may happen towards the end of the process. For soot particles with a diameter of 40 nm their lifetime graphs are given in fig.2c. Since the area for soot burn-out is overlapping with the region of No_x, formation it seems a better strategy to minimize soot formation only during the first and second phases of the combustion process.

Following the injection of fuel into the combustion chamber and its mixing with air, an inhomogeneous fuel-air mixture with various local states is formed which often exceeds the optimum described previously. This is a typical problem for diesel combustion but at the same time its optimization becomes obvious which offers research opportunities for more accurate control of the air flow and fuel injection in the piston-bowl of DI diesel engines.

6. POLLUTANT FORMATION

6.1 FORMATION OF NO_x

The formation of NO_x cannot be prevented in the combustion of fuel with air, as nitrogen is the main component of ambient air, unless gas temperatures are significantly reduced. Higher temperatures (high efficiency combustion systems operate at high temperatures) lead to higher NO_x formation rates.

As the temperature of the intake air increases, there is a subsequent rise in the combustion temperature result and thus in the formation of NO_x. In engines fitted with turbochargers cooling of the compressed air is very effective to reduce NO_x emissions.

Amongst the various developments, exhaust gas recirculation (EGR) has proven to be the most cost-effective method for reducing NO_x emissions in passenger car diesel engines, without any strong adverse effects on ignition delay, particulate emissions and fuel consumption. Although there is a consensus that EGR leads to reduced flame temperatures, there are a number of unresolved questions about the mechanisms that influence combustion and pollutant formation. It is for this reason that, within the IDEA-EFFECT research programme, emphasis was given to investigating the effect of EGR on spray development, combustion and emissions and the results obtained in the optical VW 1.9L TDI engine were presented in [12,13] and are summarised below. In general, increasing EGR in the range 0-50% leads to:

* core flame temperatures lower, on average, by about 100K depending on the engine operating condition
* similar spray tip penetration and angle
* reduction in the rate of soot oxidation
* increase in the number and spread of the auto-ignition sites
* reduction in NO_x and O_2 levels
* increase in soot, CO, CO_2 and HC concentrations
* small effect on ignition delay as a function of enginespeed, load, injection timing and level of turbocharging

In addition, cooling the EGR stream can give even lower NO_x emissions although this trend may not be true over the whole range of EGR-rates.

6.2. POLLUTANT FORMATION: SOOT

To influence soot formation in diesel combustion, the individual steps of the formation process under different operating conditions (especially at the high pressure and highly turbulent non-premixed environment) must be known; these steps are complicated by speed determining factors and the simultaneous occurrence of soot formation and oxidation.

Soot formation resulting from combustion of fuel hydrocarbons will be discussed below on the basis of the concepts of premixed and diffusion flames.

The general idea of the soot formation process in a premixed flame is presented in fig. 3a together with the specifications of the flame. It is evident that in this flame soot formation is nearly completed after about 15 ms. The soot volume fraction reaches a plateau after which it increases very slowly. The main volume of soot is formed during a phase in which the number density of the soot particles decreases significantly which implies that soot does not primarily result form the steady formation of new particles. In the early phase of soot formation there is a short phase where formation of new particles can be observed; these particles coagulate, which results in an increase in the volume of the particles. Simultaneously the volume of the total particles increases mainly as a result of the gaseous component growing on the particle surface. This particle growth velocity is approximately one order of magnitude higher than that during the onset of soot formation.

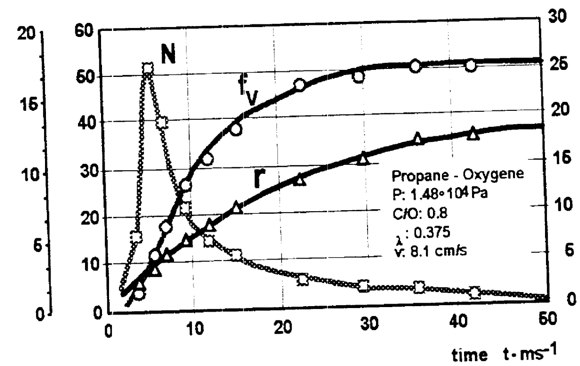


Fig. 10a: Soot formation in a premixed propane-oxygen flame

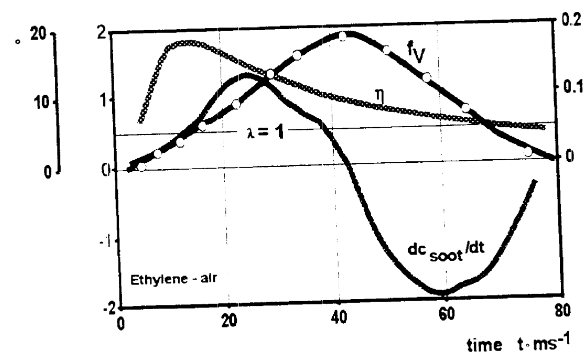


Fig. 10b: Soot formation in an ethylene-air diffusion-flame

The soot formation in a diffusion flame is shown in fig. 10b. In this type of flame the carbon/oxygen atomic ratio does not remain constant along a particle path due to the mixing process between the fuel, air and the reaction products. The mixture fraction η is the normalized mass fraction of carbon in the mixture; thus the stoichiometric ratio (λ = 1) corresponds to η = 0.052. In this flame the soot formation is completed after a few milliseconds as in the premixed flame. The particles first appear in the fuel rich area (η > 0.052) where the soot formation rate and the soot volume fraction increases. With increasing time, more combustion air is entrained into the flame which leads to a decrease in η and a slowing down of the soot formation rate. Finally the particles reach the fuel lean area where soot oxidation and therefore negative formation rates become dominant.

The results of these fundamental flame studies imply that soot formation is a kind of phase transition of the type gas-solid. The solid phase has no standard physical and chemical structure. The soot formation has to be divided into the formation of new particles, particle coagulation, condensation, and surface growth in addition to oxidation of particles, i.e.

$$
\begin{aligned}
(dN_i/dt)_{total} = \; & (dN_i/dt)_{growth} + (dN_i/dt)_{coagulation} \\
& + (dN_i/dt)_{condensation} + (dN_i/dt)_{surface\ growth} \\
& + (dN_i/dt)_{oxidation}
\end{aligned}
$$

These processes (see also fig. 11b) occur to a large extent simultaneously - and, in addition and in competition to each other, the chemical reactions of the fuel including those involving the intermediately formed hydrocarbons are taking place. For the premixed flame shown in fig. 10a, only approximately 1% of the carbon atoms follow the path leading to soot; the rest can be found in the combustion products. In modern diesel engines it is even less: only about 0.2% of the burned fuel is converted into particulates.

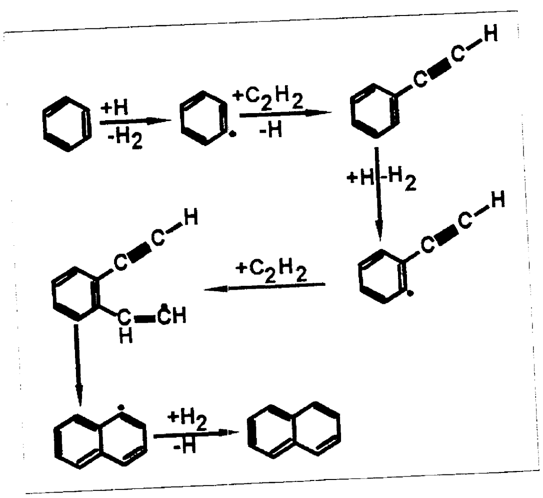


Fig. 11a: H-Abstraction-Ethine-Addition mechanism for the planar growth of polycyclic aromatics

6.2.1 Formation of New Soot Particles

A simple mechanistic scheme for the soot formation in 'molecular' terms was first presented by Frenklach and Wang [14]. Initially, the fuel molecules are oxidized in the premixed gas phase. This results in the formation of smaller molecules and among others in ethiene (acetylene) which is the basis for the formation of higher hydrocarbons and aromatics. The latter grow in a planar structure by an H-abstraction-ethiene-addition mechanism (fig. 11a).

The application of the concept to diffusion flames requires an extension. The reaction of the fuel takes place in very fuel-rich areas under pyrolysis conditions which means that the 'chemical part' of the model has to be extended by including pyrolysis reactions of the hydrocarbons. A further complication is the high pressure and turbulent environment.

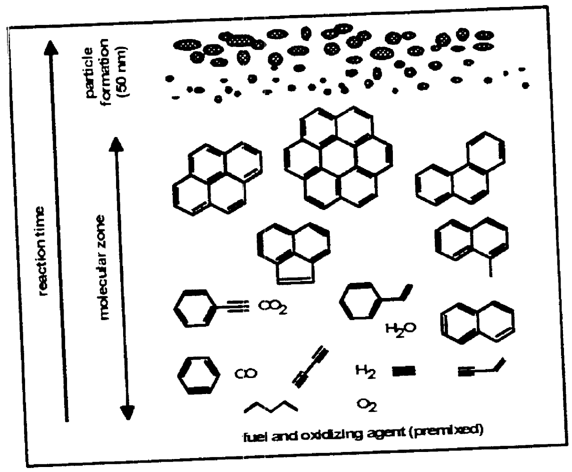


Fig. 11b: Reaction scheme of soot formation

Since there is still a lot of information on the pressure dependence of individual reactions which is missing, a full theoretical description is not feasible and the only possible approach is through simplifications.

6.2..2 Coagulation of Soot Particles

Once the soot particle density has reached a maximum, the evolution of particle sizes is mainly influenced by the coagulation process with the characteristic hyperbola-like reduction in the particle number density (see fig. 10a) after the peak. Fig. 12 shows the time dependency of the size distribution of the particles for the premixed flame of fig. 10a [15]. The relative width of the size distribution, expressed as the ratio between the second and first moments of the particle volume distribution (sixth and third moments of the size distribution) remains constant. The size distributions can be shown to be log-normal and the moment ratio 2.1. The same form of size distribution can also be applied to the soot particles from diesel engine exhaust gases.

In diffusion flames the development as a function of time of the particle number density and particle size distributions is also dominated by coagulation (fig. 10b). The most important difference from the premixed flame is the oxidation of the soot particles when the reaction products in the fuel rich zone are mixed with combustion air. The oxidation of the particles leads to a decrease of the particle size by the same amount per unit time for each order of magnitude. In this way the particle size distribution is skewed at the end towards smaller sizes and the relative width of the distribution increases.

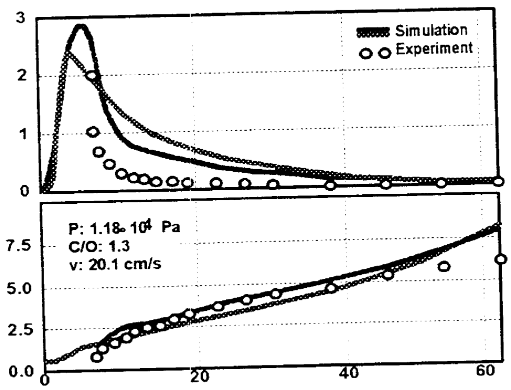


Fig. 12: Time dependent development of particle size distribution of the premixed flame of fig. 10a

For combustion at the high pressures present in diesel engines the theoretical description of the coagulation in principle, remains unchanged. However, appropriate corrections in the coagulation coefficients have to be applied [11].

6.2.3 Surface Growth of Soot Particles

The surface growth of soot particles in premixed hydrocarbon flames can be described empirically by a first order law of the type:

$$df_V/dt = k_{SG}\,(f_{V,\infty} - f_V)$$

where: df_V/dt = soot formation rate
$f_{V,\infty}$ = final soot volume fraction
f_V = soot volume fraction
k_{SG} = velocity coefficient

Because of the low volume which results from the formation of new soot particles, the soot formation rate can be considered to be equal to the surface growth rate.

The surface growth rate in premixed flames is influenced by k_{SG} which mainly depends on the maximum flame temperature. The other limiting factor is f_V which depends on the fuel specifications, the fuel concentration, the pressure and the maximum flame temperature. In fig. 13 [11] the dependence of the final soot volume fraction $f_{V,\infty}$ on the fuel concentration (represented by the C/O ratio of the mixture) and the maximum flame temperature for ethylene/air mixtures is plotted. A temperature dependence of $f_{V,\infty}$, with a maximum at about 1600 K and a non-linear increase of $f_{V,\infty}$ with C/O ratio are the main characteristics. If the pressure rises, these dependencies qualitatively are maintained with $f_{V,\infty} \div p^2$ for pressures up to 1 MPa and $f_{V,\infty} \div p$ for above 1 MPa. The shape of $f_{V,\infty}$ and the limited area from fig. 13 confirm qualitatively the soot formation area indicated in fig. 9.

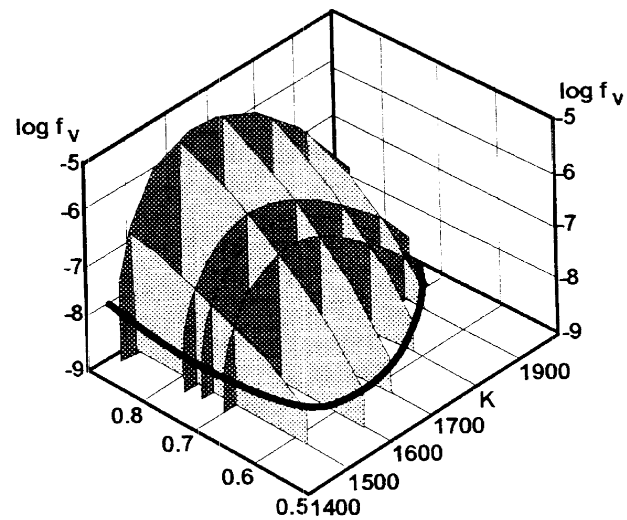


Fig. 13: Final soot volume fraction of a premixed ethylene-air flame as function of the C/O ratio and the temperature

If the above equation is integrated with a constant rate coefficient k_{SG} as a first approximation, the final soot volume fraction becomes:

$$f_V = f_{V,*} \,*(1 - \exp(-k_{SG} *t))$$

The surface growth of the soot particles in premixed flames is thus a process with an exponentially decreasing rate. For diffusion flames this behavior is observed by changing the C/O ratio along the particle trajectory, (see fig. 10b).

Surface growth represents the most important mechanism for most of the soot mass formed during combustion. Some helpful information for the understanding of soot formation is available but, for a more detailed analysis, a chemical interpretation is required to explain the exponential decay of the soot formation rate. At present, only various hypotheses exist.

6.2.4 Oxidation of Soot Particles

The oxidation of soot particles is the dominant process for the last phase of diesel combustion (see fig. 9c). The particles reach an oxygen-rich environment because of the mixing of the combustion products with a lean air/fuel mixture. For diffusion flames this phase is characterized by negative formation rates along the particle trajectories in areas with $\eta < \eta_V$ (fig. 10b).

The oxidation rate is generally represented by the following equation:

$$- dc_{soot} / dt = A * S$$

where S is the surface specific oxidation rate and A the specific surface of the particles.

Soot particles consist neither of carbon only nor does the gas phase around the particles contain oxygen only. The oxidation velocity is determined by a complex reaction scheme (applicable for high pressure and high temperature) and transport processes in the turbulent environment. For reliable calculation of the oxidation rate under diesel engine conditions the elementary steps have to be clarified in detail. At present mainly phenomenological information (which can be used for engineering purposes) is available.

Within the IDEA Programme [3,4] a lot of research on pollutant formation under diesel engine conditions was initiated or intensified. Promising seems to be the application of the flamelet concept to diesel combustion simulation [16,17] - but further work is required. However, it may not be possible to design a soot-free diesel engine since non-homogeneous combustion always produces soot and thus, particulates in the engine exhaust.

CONCLUSIONS

The fundamental processes taking place in diesel engines and, in particular, direct-injection diesels were presented and discussed with emphasis on mixture preparation and combustion. Aspects of NO_x and soot formation were highlighted in view of their significant importance in the effort of diesel engines to meet the forthcoming emissions legislation. The improved understanding of pollutant formation mechanisms obtained in various European research programmes complemented by recent developments in exhaust after-treatment technology represent the most important factors in securing the position of diesel engines in the passenger car market.

The fundamental experiments were performed with model fuels (often one-component substances). Besides engine technology fuel qualitites play a major role and influence the engine emissions as demonstrated in EPEFE [18] and other research programmes [2].

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

Air/fuel ratio λ = quantity of air supplied / theoretical requirement. For a stoichiometric mixture $\lambda = 1$. A lean mixture ($\lambda > 1$) contains excess air, while a rich mixture ($\lambda < 1$) contains less air.

EPEFE = European Programme on Emissions, Fuels and Engine Technologies: Research programme in the framework of a much larger initiative known as the European Auto-Oil Programme. This initiative was the result of the new approach adopted by the European Commission aimed at addressing more rationally the question related to air quality requirements in Europe. The objectives of the Auto-Oil Programme are to determine the most cost-effective package of measures which, on implementation in 2000, would achieve the necessary air quality targets set for the year 2010[2,18].

IDEA = Integrated Diesel European Action: detailed experimental and theoretical studies of the fundamental phenomena in diesel engines like air flow, fuel injection, mixture formation, auto-ignition, combustion and pollutant formation (jointly funded by Fiat, Peugeot SA, Renault, Volvo and Volkswagen, the Commission of the European Communities and the Swedish National Board of Technical Development [3].

IDEA EFFECT = Integrated Development on Engine Assessment with Environment Friendly Fuel Efficient Combustion Technology: Continuation of the IDEA Programme with inclusion of the homogeneous combustion (gasoline engine) [4].