Chapter 3
Guidelines for Improving Diesel Engine Characteristics

Diesel engine characteristics depend significantly on the engine type. But, even for a given engine type, the engine characteristics can still be varied in a wide range in dependence on *engine management*, *exhaust gas after treatment*, and usage of *alternative fuels* (Fino et al. 2003; Gray and Frost 1998; Maiboom et al. 2008; Peng et al. 2008; Stanislaus et al. 2010; Twigg 2007) (Fig. 3.1). Engine management and alternative fuels usage offer a possibility to *reduce* the *formation* of harmful emissions. On the other hand, exhaust gas after treatment techniques enable a *reduction* of harmful emissions *already produced* by the engine.

3.1 Engine Management

Diesel engine management is mostly related to the fuel injection process. Modern fuel injection systems must in general fulfill the following requirements:

- *High pressure capability* and *injection pressure control*
- *Flexible timing control*
- *Injection rate control*

Increased *injection pressure* generally contributes to decreased fuel droplet size and improved combustion, resulting in a reduction of smoke emission. Low injection pressure, on the other hand, is required to reduce noise at idling and in the very low load range. In other words, optimum injection pressure must be determined in accordance with engine load and speed. Regarding the fuel consumption, it was found out that some improvement might be achieved by increasing the injection pressure. More precisely, the fuel should be injected with low pressure at the initial phase of injection and later with high pressure. According to above requirements, a fuel injection system should have a wide *pressure controllability*.

Injection timing influences to a great extent the combustion process and is especially important for the control of nitrogen oxidant (NO_x) and particulate matter (PM) emissions. Flexible injection–timing control, which considers both,

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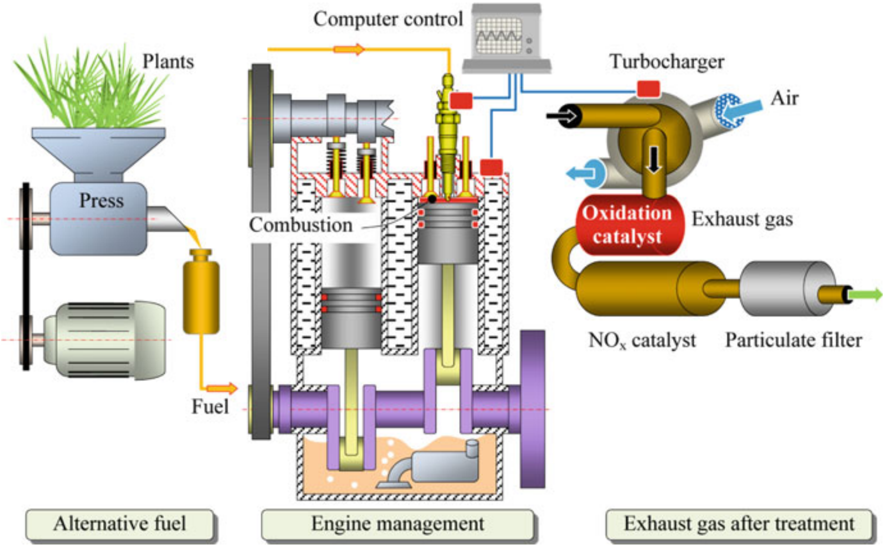


Fig. 3.1 Techniques to improve engine characteristics

engine speed and load, is therefore a highly desired feature of the injection system. The *injection rate* history variations can also be used to achieve reasonable NO_x–PM trade-offs. Besides of the NO_x and PM emission, the injection rate history also affects fuel consumption.

Once the injection characteristics are optimally matched to each other, they typically result in a fuel spray with small droplet size (good atomization), long tip penetration, and narrow spray angle. These spray characteristics play an important role in the efforts to improve engine performance and to reduce fuel consumption and harmful emissions. This is especially true for the fuel atomization (Kook et al. 2008).

To reduce harmful emissions in a wide engine operating range, multiple fuel injections can be used. Multiple fuel injections along with high fuel injection pressures represent an effective way of diesel engine combustion improvements (Kim et al. 2008; Wang et al. 2009). These are the main reasons for the development of electronically controlled high pressure fuel injection systems like the common rail system shown in Fig. 3.2. The common rail system basically consists of a supply pump, common rail, injectors, an ECU that controls these components, and sensors (Fig. 3.2).

The fuel pressure in the common rail is controlled by a solenoid valve that controls the fuel volume from the supply pump. This pressure is detected by a sensor in the common rail and set accordingly to the engine speed and load. This enables high pressure injection also at low speed regimes, allowing fuel to be well atomized at all operating regimes. Consequently, it is possible to reduce significantly the black smoke, being characteristic for diesel engines.

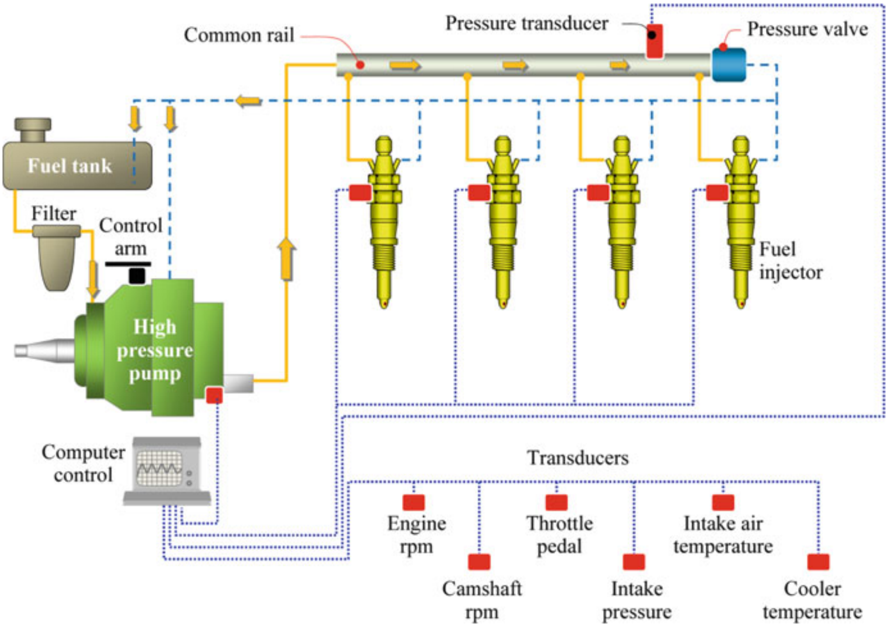


Fig. 3.2 Common rail diesel fuel injection system

The injection process in a common rail system can be roughly described as follows. The *injection volume* and *timing* are controlled by opening and closing of the solenoid valve (Fig. 3.3). When the solenoid valve is closed, the pressures in the working chamber and in the needle chamber are equal to the *common rail fuel pressure* and the needle is in the "closed" position. When the solenoid valve opens, the pressure in the working chamber falls and the common rail pressure in the needle chamber lifts the needle. This starts the injection. As the solenoid valve closes, the working chamber is pressurized again. This causes the needle to return to the "closed" position and terminates the injection. That means that the whole process is controlled by the solenoid valve, which, in turn, is controlled by the ECU. This means, for example, that by adding a power pulse to the solenoid valve prior to the main injection, one can easily get a pilot injection. This can be used to shorten the ignition delay and suppresses premixed combustion to address NO_x, combustion noise, and vibration reduction and to improve startability and fuel efficiency.

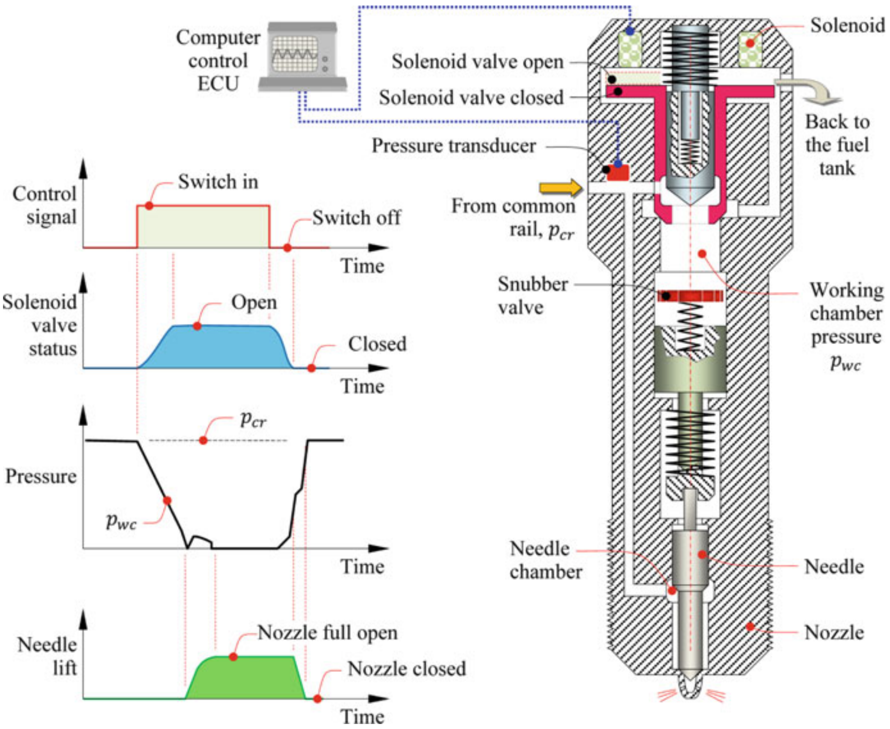


Fig. 3.3 Common rail diesel fuel nozzle

3.2 Exhaust Gas After Treatment

The exhaust of a diesel engine combustion process mainly contains the following harmful emissions: HC, CO, NO_x, and particulate matter (PM). Therefore, diesel engines should be equipped with *HC* and *CO oxidation control systems*, *lean–NOx control systems*, and *PM control*. In practice, these systems are typically *catalytic convertors* and *particulate filters* (Fig. 3.4) (Fino et al. 2003; Gray and Frost 1998; Peng et al. 2008; Stanislaus et al.2010; Twigg 2007). Additionally, *exhaust gas recirculation* may be successfully employed to reduce harmful emissions.

The most common of these devices are *catalytic convertors*. Physically, a catalytic convertor is an open monolithic structure that allows easy flow of gases, with the active catalyst coating applied to the channel walls of the monolith (Fig. 3.5).

The catalyst is typically platinum, palladium, rhodium, or some alloy. These catalysts can eliminate HC, CO, and NO_x emissions, and partially reduce particulates (Gray and Frost 1998; Sitshebo et al. 2009; Twigg 2007). The convertor oxidizes carbon monoxide and hydrocarbons to carbon dioxide and water and simultaneously reduces nitrogen oxides to nitrogen.

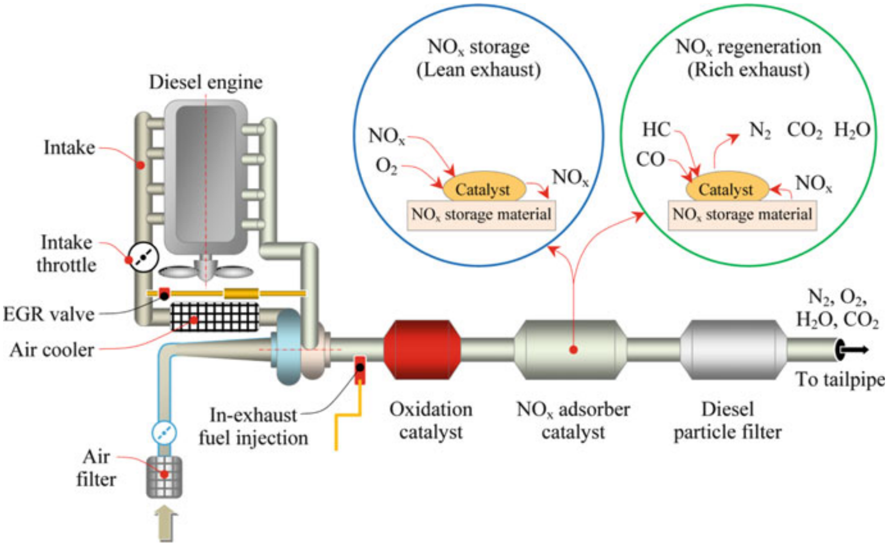


Fig. 3.4 Exhaust gas after treatment

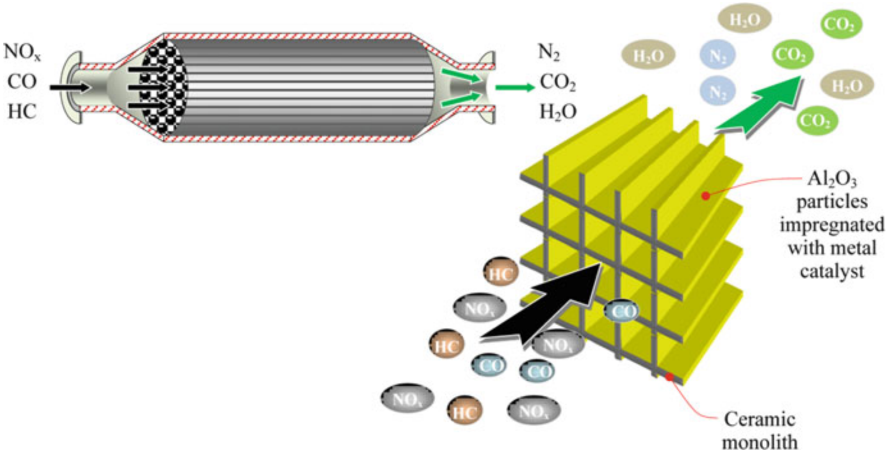


Fig. 3.5 Catalytic converter

Diesel catalytic converters are *oxidation catalyst* and *SCR catalyst* (Alkemade and Schumann 2006; Sitshebo et al. 2009; Forzatti et al. 2010). The diesel *oxidation catalyst* removes up to 90 % of carbon monoxide and hydrocarbons. The *SCR catalyst* removes NO_x.

The primary function of the *oxidation catalyst* is to oxidize the HC and CO produced during advanced combustion modes. For this purpose, the platinum-based oxidation catalysts are typically used (Twigg 2007). To achieve the performance and durability, the catalyst should contain the platinum in a highly dispersed form, which is well stabilized against thermal sintering.

The major reactions in a diesel oxidation catalyst are as follows:

$$\begin{gathered}
C_nH_m + \left(n + \frac{m}{4}\right) O_2 \rightarrow nCO_2 + \frac{m}{2}\ H_2O \\
CO + \frac{1}{2} O_2 \rightarrow CO_2 \\
NO + \frac{1}{2} O_2 \rightleftharpoons NO_2 \\
H_2 + \frac{1}{2} O_2 \rightarrow H_2O
\end{gathered} \tag{3.1}$$

To improve the oxidation of the hydrocarbons at the cold start of the engine, the zeolites are incorporated into the catalyst. The zeolites function is to absorb the HC that would otherwise inhibit the active platinum sites and to improve CO and HC oxidation performance. At higher temperatures, the HC is desorbed from the zeolite and oxidized over the platinum catalyst sites.

The catalytic *dissociation* of NO_x to O_2 and N_2 is not possible, although NO_x is thermodynamically unstable. The reason of this lies in the high affinity of metallic catalyst surfaces for O_2, compared to that for N_2 that leads to oxygen poisoning of the metal surface. A feature of many lean NO_x reduction reactions is that there is insufficient reduction capability on the surface to reduce NO_x completely to N_2, and a significant amount of N_2O can be formed (Twigg 2007).

Two popular alternatives to catalytically reduce the NO_x from the exhaust to harmless N_2, even at low exhaust gas temperatures, are the *lean* NO_x *trap* or *LNT catalyst* and *selective catalytic reduction* or *SCR catalyst*.

A *LNT catalyst* consists of an alkali or alkaline earth metal oxide or carbonate mixed with a noble metal (typically Pt) on the same washcoat (Sampara 2008). During lean operation (excess oxygen), the NO oxidizes to NO_2 in the presence of the noble metal catalyst. The NO_2 then chemisorbs onto the metal oxide or metal carbonate forming metal nitrate. Over a period of time all the metal oxide (carbonate) is consumed to form nitrate. Hence, it has to be periodically regenerated, where the nitrate decomposes back to metal oxide (carbonate).

The desirable reactions during lean operation are

$$\begin{gathered}
2NO + O_2 \rightarrow 2NO_2 \\
NO_2 + mCO_3(mO) \rightarrow mNO_3 + CO_2
\end{gathered} \tag{3.2}$$

Undesirable reactions during lean operation are

$$\begin{gathered}
2SO_2 + O_2 \rightarrow 2SO_3 \\
SO_3 + mCO_3(mO) \rightarrow mSO_4 + CO_2
\end{gathered} \tag{3.3}$$

and the desirable reactions during rich operation are

$$\begin{gathered} 2\mathrm{mNO_3} \rightarrow 2\mathrm{mO} + 2\mathrm{NO} + \mathrm{O_2} \\ 2\mathrm{NO} + 2\mathrm{CO(HC)} \rightarrow \mathrm{N_2} + 2\mathrm{CO_2} \\ \mathrm{mO} + \mathrm{CO_2} \rightarrow \mathrm{mCO_3} \end{gathered} \tag{3.4}$$

This regeneration is generally obtained by creating a rich environment with high levels of HC or CO or H_2. The NO formed due to the decomposition of the nitrate reacts with the reductant to form N_2. The undesirable reactions under LNT operation are the oxidation of SO_2 to form metal sulfates. NO oxidation to NO_2 occurs on the noble metal between 250 and 450 °C. This reaction is kinetically limited below 250 °C and thermodynamically limited beyond 450 °C. Also, the stability of the nitrate formed is severely limited beyond this temperature. There are two important problems with using an LNT for NO_x reduction. First, the saturated LNT with metal nitrate (mNO_3) should be periodically regenerated by providing a reducing environment which contains high levels of HC or CO or both. This is typically achieved by running the engine rich or by having a secondary fuel injection late in the exhaust stroke. This involves additional fuel penalties and complications with respect to control and design. Second, the sulfur in the fuel generally reacts with the metal carbonate or metal oxide to form metal sulfates (mSO_4). Reducing these back to its original form needs high temperatures of the order of 600 °C.

A number of various strategies have been intensively investigated to find an appropriate *SCR catalyst* of NO_x to N_2 (Kamasamudram et al. 2010; Sampara 2008; Sitshebo et al. 2009). Some of the practical techniques to reduce NO_x can be realized by using:

- Soot particulate
- Ammonia (urea)
- Hydrocarbons over zeolite-based catalysts
- Hydrocarbons over metal oxide catalysts
- Hydrocarbons over multistaged catalysts
- Hydrocarbons over noble metal catalysts

Among these options, the use of ammonia and hydrocarbons over zeolite-based catalyst are the most popular. Zeolite-based catalysts have received much attention due to their high activity and relatively wide temperature window since 1990.

SCR of NO with ammonia (NH_3) under lean conditions is a widely commercialized technology for NO_x removal from stationary sources. Vanadium-based catalysts are commonly used for this application. Sources of NH_3 can be compressed gas, or compounds such as urea ($(NH_2)_2CO$), which readily decompose to give NH_3:

$$(\mathrm{NH_2})_2\mathrm{CO} + \mathrm{HO_2} \rightarrow 2\mathrm{NH_3} + \mathrm{CO_2} \tag{3.5}$$

This hydrolysis reaction occurs at temperatures beyond 160 °C. Over an SCR catalyst, NH_3 reacts with NO_x according to the following reactions:

$$\begin{gathered} 4NH_3 + 4NO + O_2 \rightarrow 4N_2 + 6H_2O \\ 2NH_3 + NO + NO_2 \rightarrow 2N_2 + 3H_2O \\ 4NH_3 + 2NO_2 + O_2 \rightarrow 3N_2 + 6H_2O \end{gathered} \tag{3.6}$$

Among the above reactions, the reaction of NH_3 with NO and NO_2 is the most facile and therefore occurs at lower reaction temperatures. Thus, an external source which can provide a $\frac{NO}{NO_2} = \frac{1}{1}$ ratio would be ideal to achieve best SCR performance.

Metal catalysts such as platinum (Pt), copper (Cu), iridium (Ir), and more recently silver (Ag), may be mixed along with the zeolite on the same washcoat. The activity of these catalysts is closely related to the type of zeolite and their structure. As a rule of thumb, zeolite structures with lower acidity lead to smaller carbonaceous deposits, leading to higher NO_x conversions. Furthermore, HCs are trapped in the zeolite structures to increase the local HC concentration.

NO is oxidized to NO_2 at the catalyst surface which contains noble metal. The formed NO_2 then reacts with the trapped HC to produce N_2. The main obstacle to the use of this system is the hydrothermal stability of the zeolite. Furthermore, this method becomes problematic at high temperatures, where HC desorption is significant, which reduces local HC concentration at the catalyst surface.

Particulate filter is a device designed to remove diesel particulate matter or aerosolized diesel exhaust pollution particles. It looks similar to a traditional exhaust silencer, but is mounted closer to the engine. Inside of it there is a complex honeycomb ceramic structure, designed to filter the exhaust gases while minimizing the flow resistance (which would otherwise limit performance) (Fig. 3.6). By forcing the exhaust gases through the walls between the channels of the filter, particulate matter is deposited on the walls.

Diesel particulate filters are ceramic filters, fitted into the exhaust, which trap about 90 % of the exhaust gas particles. The particles are mainly a mixture of soot and ash. Soot is carbon based and is formed during combustion of the fuel; meanwhile ash is metal based and forms when engine oil is burnt.

An example of a particulate filter with a regenerative effect is shown in Fig. 3.7. The exhaust gases enter the filter and go through the porous ceramics piece that holds back the particles. As the particulate matter fills up the filter, the pressure on the intake side of the filter raises. This leads to an increased portion of the residual gases in the cylinder, which affects the fuel–air mixture exchange negatively. When the pressure behind the exhaust valve reaches some critical value, the filter needs to be regenerated (the particles have to be removed by burning). For this regeneration process several systems with automatic electronic control have been developed. Fig. 3.7 shows a system with a burner that needs to be supplied with additional fuel and air. The flame from the burner destroys the particulate matter.

Exhaust gas recirculation (EGR), schematically shown in Fig. 3.8, is widely used to reduce NO_x emission from diesel engines. Effectively, it reduces the combustion flame temperature due to its dilution, thermal, and chemical effects (Peng et al. 2008).

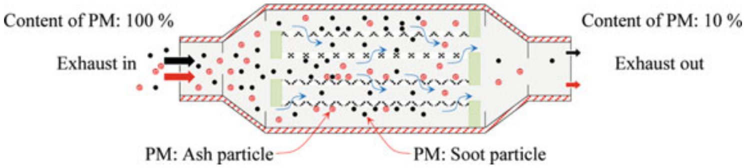


Fig. 3.6 Particulate filter

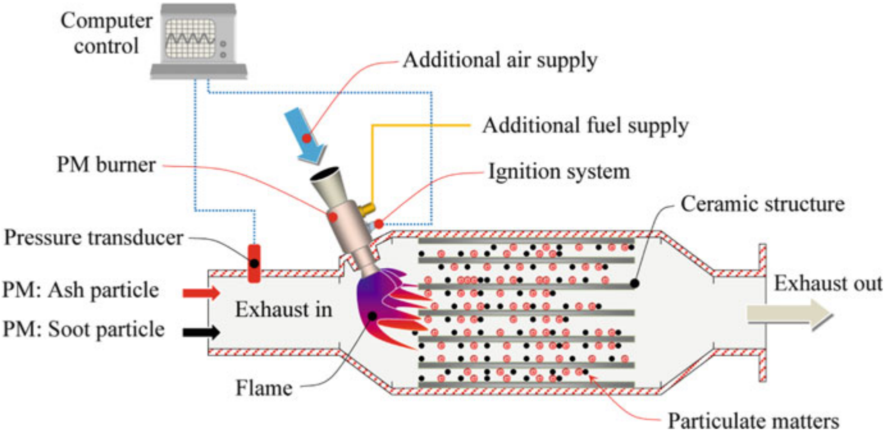


Fig. 3.7 Particulate filter with regenerative effect

The composition of the exhaust gas depends on the engine operating conditions. In the case of a cold start, the exhaust gas comprises a great deal of unburned hydrocarbons, fuel vapor, and products of partial oxidation reaction. In contrast, during normal conditions the exhaust gas contains much more products of a complete combustion. If the exhaust gas is recirculated back into the cylinder, this may have positive or negative effects on the ignition and combustion process. The quantity of recirculated exhaust gas should therefore be carefully determined in dependence of the operating conditions and mass of intake air and fuel. It is usually expressed by ψ_{EGR} [%] which is defined as the mass percent of the recirculated exhaust mass m_{EGR} in the total intake mixture mass m_i (Agarwal et al. 2011):

$$\psi_{\mathrm{EGR}}\,[\%] = \frac{m_{\mathrm{EGR}}}{m_i} \times 100 \tag{3.7}$$

An appropriate amount of EGR can improve cold startability of a diesel engine and promote combustion and emission performance during a cold start. For some operating conditions, EGR may offer a way to drastically reduce $\mathrm{NO_x}$ emissions without a significant penalty for the specific fuel consumption and soot emissions. However, under some operating conditions, the opposite effects have also been observed (Maiboom et al. 2008).

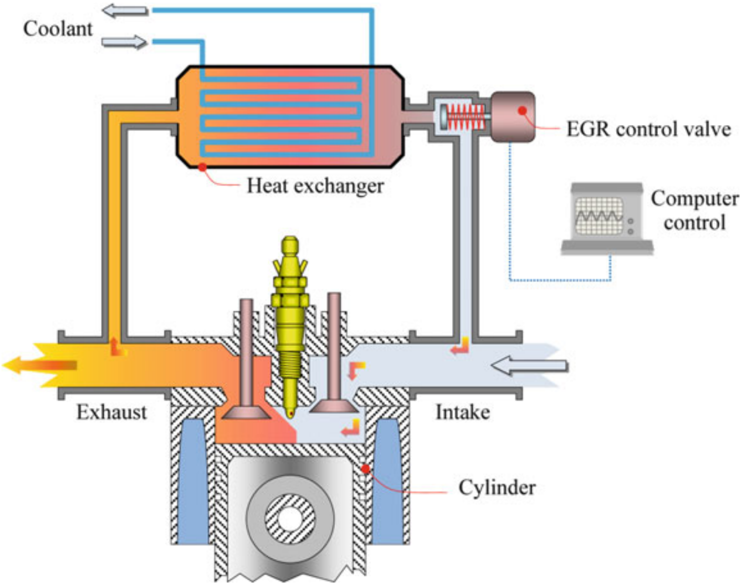


Fig. 3.8 Exhaust gas recirculation scheme

High EGR rates are known to cause large increases in PM emissions. In this view, employing a diesel particulate filter system can be considered to be a favorite solution. This is because a particulate filter provides about 90 % efficiency on reducing PM emissions and thus gives increased flexibility for NO_x control by the EGR.

Two various EGR configurations

- Low Pressure Loop (LPL)
- High Pressure Loop (HPL)

are presented in Figs. 3.9 and 3.10 (Mustel 1997; Zheng et al. 2004).

The LPL EGR (Fig. 3.9) utilize the positive difference between the turbine outlet pressure p_4 and compressor inlet pressure p_1, $p_4 > p_1$. If necessary, the outlet pressure can be elevated by partial throttling to ensure sufficient driving pressure for the EGR flow.

Exhaust is usually taken directly from the filter outlet and lead to the turbo inlet. This is considered to be a clean EGR because the recirculated flow is passing through the particulate filter. It should be noted, however, that conventional compressors and intercoolers are not designed to endure the temperature and fouling of diesel exhausts.

In general, the LPL approach of EGR is not applicable, if the compressor is not designed for exhaust gases. Efforts have also been made to route the exhaust from the turbine outlet to the intercooler outlet directly, bypassing the compressor. Although this circumvents the exhaust fouling problem, an independent EGR pump becomes necessary to counteract the boost pressure. Special EGR pumps

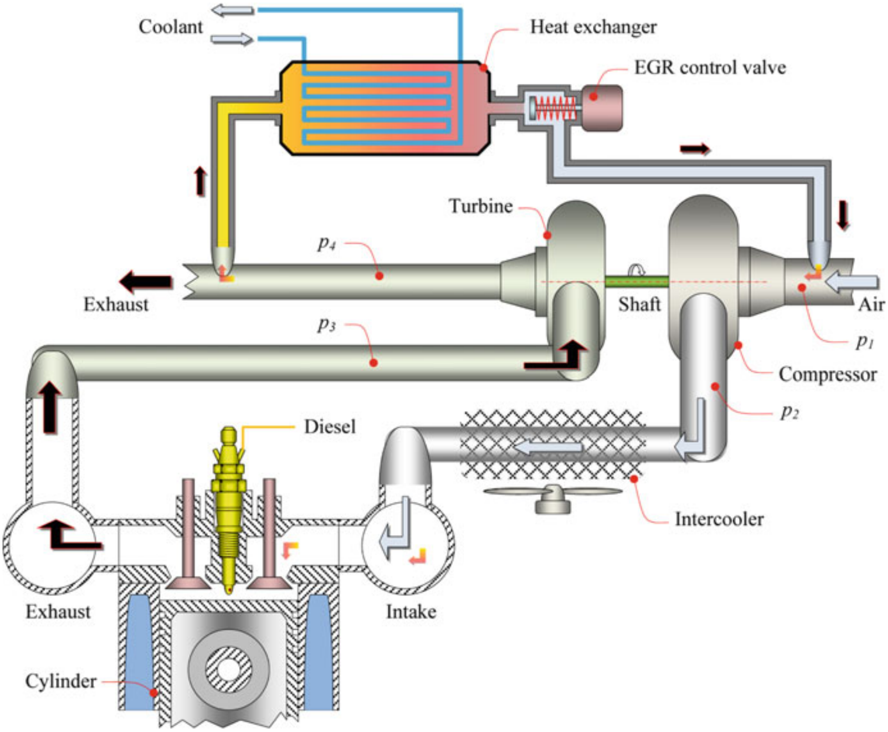


Fig. 3.9 LPL approach of EGR

are needed to withstand the exhaust heat and fouling, in addition to the substantial pumping power requirements.

For a HPL system (Fig. 3.10) it is typical to take the exhaust gases before they enter the turbocharger. After going through the EGR valves, the recirculated exhaust gases are reintroduced in the intake manifold, downstream of the compressor. The compressor and intercooler are, therefore, not exposed to the exhaust. However, such HPL EGR is only applicable when the turbine upstream pressure p_3 is sufficiently higher than the boost pressure p_2, i.e., if $p_3 > p_2$ prevails. In the case that the required pressure difference cannot be met with the original matching between the turbocharger and the engine, remedies must be made by either increasing the turbine upstream pressure or reducing the boost pressure.

Besides of the emissions, EGR may also influence engine performance, carbon deposits, and wear of various parts of a diesel engine (Al-Qurashi et al. 2011; Agarwal et al. 2011; Ghazikhani et al. 2010). EGR displaces oxygen in the intake air by exhaust gas recirculated to the combustion chamber. Exhaust gases lower the oxygen concentration in the combustion chamber and increase the specific heat of the intake air mixture, which results in lower flame temperatures. Reduced oxygen and lower flame temperatures affect performance and emissions of a diesel engine in several ways. Thermal efficiency is slightly increased and specific fuel

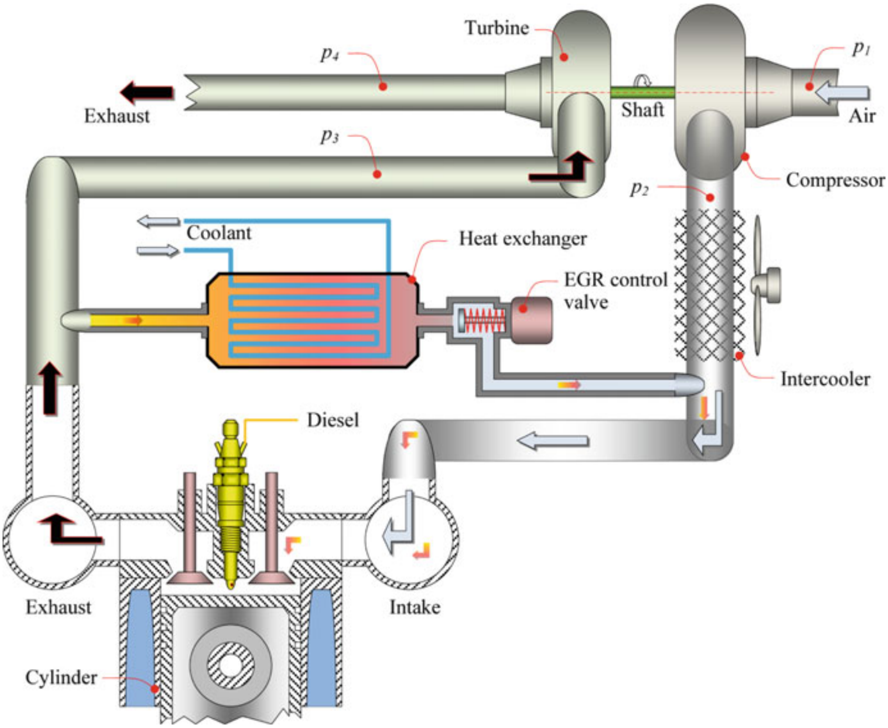


Fig. 3.10 HPL approach of EGR

consumption is decreased at lower loads. But at higher loads, thermal efficiency and specific fuel consumption are almost the same either with or without EGR.

The exhaust gas temperature is decreased with EGR. Hydrocarbons, carbon monoxide, and smoke opacity increase by EGR usage, but NO_x emissions decreases significantly. A 15 % EGR rate is found to be effective to reduce NO_x emission substantially without deteriorating significantly engine performance in terms of thermal efficiency, specific fuel consumption, and emissions. At lower loads, EGR also reduces NO_x without deteriorating performance and emissions. A higher rate of EGR, however, reduces NO_x to a great extent but deteriorates performance and emissions at higher engine loads. Thus, it follows that higher rates of EGR can be applied only at lower loads.

Related to EGR, there is another important point that needs attention. When an engine is equipped with an EGR, increased soot deposits can be expected on the cylinder head, injector tip, and piston crown. Increased wear of the piston rings was also observed. The top compression ring is not influenced significantly, but the wear of the second and the third compression rings may increase notably. The worst wear increase was observed for the oil piston ring. Unfortunately, the EGR may cause a severe wear problem on various engine components such as piston rings, cylinder liner, etc. (Suzuki 1997).

3.3 Alternative Fuels

The usage of conventional mineral diesel fuel is already quite well investigated and the problems related mainly to NO_x and PM emissions are very well known. In the quest to reduce these problems, the usage of alternative fuels may present an interesting option. The evaluation of alternative fuels utilization should include both, emissions and energy considerations. From the emissions point of view, there are of course again trade-offs to be taken into account. For example, fuels with lower PM levels can have higher NO_x or HC emission levels.

In order to evaluate their potential usage in diesel engines, alternative fuels have to be analyzed at least in relation to *cetane number*, *density*, *viscosity*, *cloud point*, *flash point*, *cold filter plugging point*, *fuel corrosiveness*, *lubricity*, *water content*, and *fuel stability*.

The *cetane number* is a measure of the ignition quality of the fuel and affects the combustion process. Up to a certain limit, an increase of the cetane level reduces the NO_x and PM emissions. However, by increasing the cetane number above the level, required for a given engine, may not improve engine performance. The minimum cetane number for mineral diesel fuel is 40.

Fuel density influences directly fuel spray development, specific fuel consumption, engine power, wear, deposits, and exhaust emissions. An increase of fuel density leads to longer spray tip penetration with a narrower spray angle.

The *kinematic viscosity* of fuel is also an important property which impacts the performance of the injection system. Some injection pumps can experience excessive wear and power loss due to injector or pump leakage, if viscosity is too low. If fuel viscosity is too high, it may cause too much pump resistance, filter damage, and adversely affect fuel spray patterns. In general, fuels with low viscosity have poorer lubrication properties. The kinematic viscosity is measured according to the corresponding standard, which defines this property as the resistance to flow of a fluid under gravity at some predefined temperature.

The *cloud point* of fuel is the temperature at which the amount of precipitated wax crystals becomes large enough to make the fuel appear cloudy or hazy. Wax may form because normal paraffins often occur naturally in a fuel. As the temperature of the fuel is lowered, these paraffins become less soluble in the fuel and precipitate out as wax crystals. In some fuel systems, cloud point can indicate the onset of fuel-filter plugging.

The *pour point* is the lowest temperature at which the fuel will flow and is used to predict the lowest temperature at which the fuel can be pumped.

The *cold filter plugging point* (CFPP) is the lowest temperature at which the fuel flow through the filter becomes problematic. This parameter is used to predict the fuel's low temperature operability properties.

The *flash point* is not related directly to engine performance. It has to be verified to meet the safety requirements for fuel handling and storage. The flash point is the lowest fuel temperature at which the vapor above a fuel sample will momentarily ignite under the prescribed test conditions.

The *fuel corrosiveness* properties indicate possible problems with copper, brass, or bronze fuel system components.

The *fuel lubricity* is a very important property, since the diesel fuel injection system relies on the fuel to lubricate the moving parts. If the fuel's lubricating properties are inadequate, this will lead to increased wear on injectors and pumps.

The *water content* in fuel may lead to increased corrosion. During fuel transport and storage, water and sediments can contaminate the fuel. This contamination can contribute to filter plugging and fuel injection system wear.

In order to improve *fuel stability*, adequate additives, fuel mixing, or fuel heating (in case that low temperatures promote gel formation) may be necessary. Unstable fuels can form soluble gums or insoluble organic particulates. Both gums and particulates may contribute to injector deposits, and particulates can clog fuel filters. The formation of gums and particulates may occur gradually during long-term storage or quickly during fuel system recirculation caused by fuel heating.

A number of fuels have been investigated as possible alternatives for diesel engines. Today's most frequently investigated and used alternative fuels are:

- *Water in diesel emulsion*
- *Natural gas* and *liquefied petroleum gas*
- *Methane* and *propane*
- *Dimethyl ether* and *dimethyl carbonate*
- *Fischer–Tropsch diesel*
- *Hydrogen*
- *Alcohols*
- *Vegetable oils*, *bioethanol*, and *biodiesel*

Some alternative fuels can be used pure, while others have to be mixed with mineral diesel. In any case, the diesel engine has to be modified to some extent.

The engines, which use mineral diesel and some gaseous fuel, are referred to as *dual fuel engines*. Natural gas and bio-derived gas appear to be quite attractive alternative fuels for dual fuel engines in view of their environment-friendly nature. In dual fuel gas diesel engine a mixture of air and gaseous fuel is prepared in an external mixing device and compressed in the cylinder. The compressed mixture is then ignited by energy from the combustion of the diesel fuel spray, which is called the *pilot fuel*. The amount of pilot fuel needed for this ignition is between 10 and 20 % of the amount needed for diesel-only operation at normal working loads. This amount varies with the point of engine operation and engine design parameters.

3.3.1 Water in Diesel Emulsion

In the last decade, water addition to mineral diesel before or after injection has been quite intensively investigated (Tesfa et al. 2012; Subramanian 2011; Maiboom and Tauzia 2011; Tauzia et al. 2010; Lif and Homlberg 2006; Armas et al. 2005; Samec et al. 2002; Kadota and Yamasaki 2002; Kegl and Pehan 2001). Namely, it can be

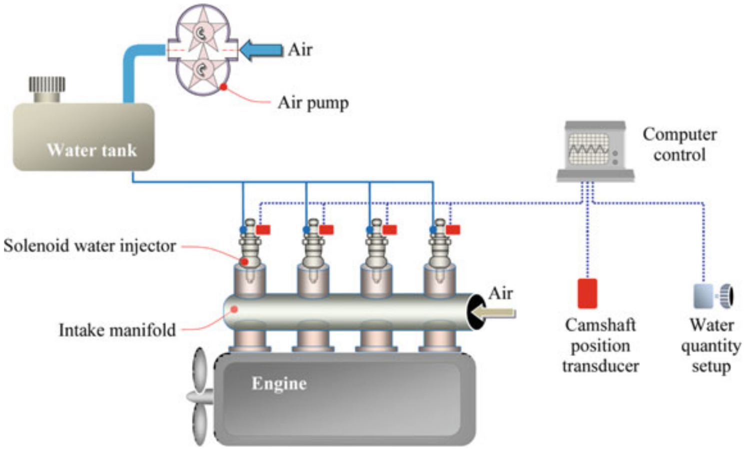


Fig. 3.11 Scheme of a multi-point water injection into the intake manifold

shown that by adding water to mineral diesel, the peak combustion temperatures are reduced which leads to a high reduction of NO_x. Till today, various methods of water addition have been developed, such as:

- *Water injection into the intake manifold*
- *Water injection directly into the combustion chamber*
- *Water injection into emulsion with the fuel*

Water Injection into the Intake Manifold. In order to inject water into the intake manifold, several systems have been investigated till now. The first one is the *multi-point water injection system* (Brusca and Lanzafame 2001; Imahashi et al. 1995; Odaka et al. 1991), which basically consists of the following parts: injectors, electronic control unit, pump, water tank, and pipes (Fig. 3.11).

The water tank is loaded by air pressure, which is controlled by a pressure valve. The injectors are all the time connected with the water tank and directly loaded by water pressure. The electronic control unit (ECU) manages the injectors according to the signals from impulse sensors related to piston position. The injectors have integrated electric coil valves that are activated by an electric signal from the ECU. In the absence of an electric signal from the ECU, the injector valve is closed. The ratio between air and water is adjusted by a potentiometer. The amount of water injected into the air stream depends on pressure that is adjusted by the air pressure in the water tank. The air pressure in the water tank is controlled by an electronic regulator. The injection procedure starts immediately after the intake valve is open and finishes just before the intake valve is closed.

A very cost-effective solution to reduce NO_x emission and thermal loading of the engine is the *mono-point water injection*. In this system, water is added to the air stream with one injector, located just *before* the *turbocharger*. The main advantage of this system is the simplicity of its construction (Fig. 3.12).

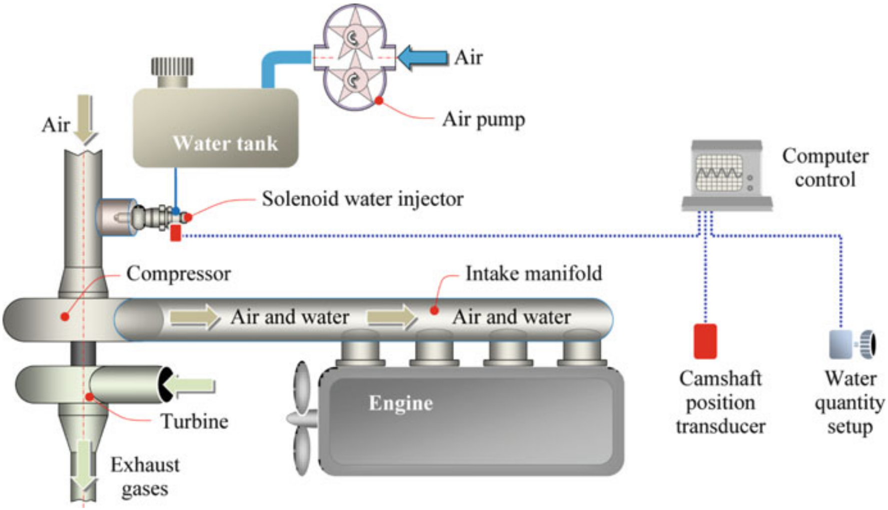


Fig. 3.12 Scheme of mono-point water injection before turbocharger

There are no significant differences in measurable engine characteristics between the mono-point and multi-point water injection systems. Both systems reduce NO_x emission effectively. Some authors also report a positive influence of water injection on PM emission, but this topic is not well investigated.

Water Injection Directly into the Combustion Chamber. Various strategies have been proposed till now to inject water directly into the combustion chamber, with the aim of reducing NO_x emissions and keeping the water quantity small in comparison to water injection into the intake manifold. A good example is the *stratified fuel/water injection system* (Miyano et al. 1995). By this method, water is fed into the injector's dead volume when fuel injection does not take place. When injection begins, fuel and water are injected into the combustion chamber in a stratified condition.

At the end of the previous injection process, the high pressure tube and injector dead volumes contain only pure fuel. At an appropriate moment, determined by the ECU, water is pumped into the injector's dead volume at some pressure, that is higher than the opening pressure of the non-return valve and lower than the needle nozzle opening pressure. The delivered water pushes the fuel in the high pressure system toward the fuel pump. The water quantity is defined by the water supply pressure and by the duration of the solenoid valve opening. At the moment of the solenoid valve closing, water delivery is finished and the non-return valve closes. At this moment, the middle part of the injector's dead volume is filled with water, while the rest is filled with fuel. This means that the injection process starts with a small amount of fuel. This is followed by the injection of water and finally again fuel.

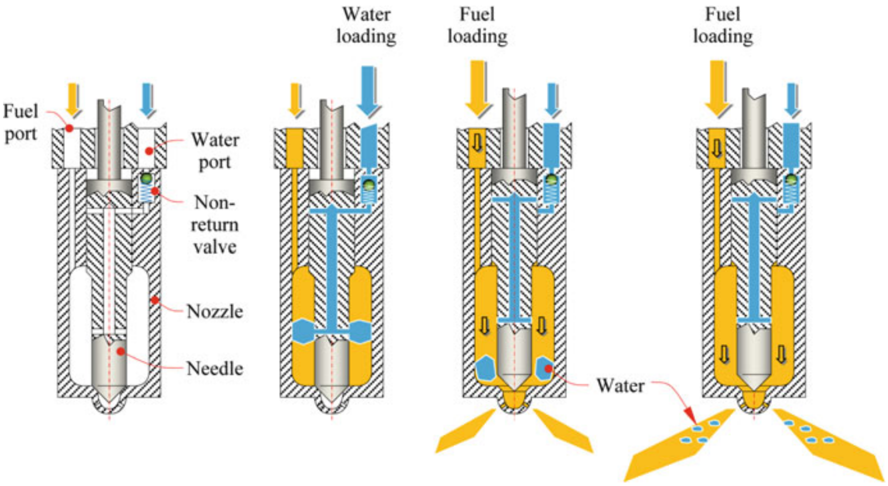


Fig. 3.13 Scheme of the stratified fuel/water injection system

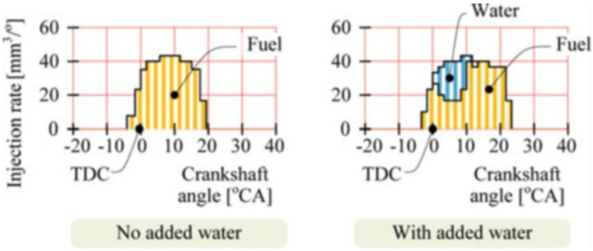


Fig. 3.14 Stratified fuel/water injection process

The stratified fuel/water injection system is shown schematically in Fig. 3.13.

The injection rate history ($dq/d\alpha$) of fuel/water injection in dependence on crankshaft angle after top dead center (ATDC) is presented in Fig. 3.14. The graph looks like a three-phase injection; in both graphs the total quantity of injected fuel is the same. This means that water addition leads to increased injection duration.

Injecting water in this way delays slightly the ignition but this delay can be kept at a low level, even if a large quantity of water is injected and significant NO_x reduction is achieved. An important advantage of direct water injection is the possibility to change the water to fuel ratio in dependence on engine parameters (speed and load) or during engine warm-up (cold start) (Stanglmaier et al. 2008). This system allows for varying the percentage of water in the mixture on a cycle-resolved basis. Consequently, a considerable improvement in NO_x and PM emissions can be obtained, both, under steady-state and transient conditions. The control of water percentage on a cycle-resolved basis has also been shown to be an effective method for mitigating NO_x and smoke emissions over step-load transients.

Water/Fuel Emulsion Injection into the Combustion Chamber. In recent years, water/fuel emulsion has also been investigated with the aim to achieve improvements in fuel economy and reduction of NO_x and PM emissions. The emulsion fuel is determined by an appropriate water/diesel ratio and a corresponding stabilizing agent (Park et al. 2000). Some minimal amount of the stabilizing agent has to be added to the fuel to avoid any negative effects on combustion. For example, 0.1 % of a frequently used stabilizing agent, Sekiemal SA, is typically added to water and mixed with a centrifugal mixer (Tauzia et al. 2010; Park et al. 2000). Additional stirring fans and circulating pipelines have to be installed at the fuel tank and just ahead of the injector, as water drops tend to combine and submerge in the inherently unstable emulsion fuel. The droplet size of the emulsion fuel is one of the most important factors determining the subsequent combustion characteristics.

It has to be noted that the ignition locations of emulsion fuels differ from those of mineral diesel (Park et al. 2000). With mineral diesel, ignition typically occurs in the middle of the combustion chamber, while for water/diesel emulsions it may occur in the bottom region or at multiple points simultaneously. The flame of a water/diesel emulsion propagates slowly from the ignition locations, so that it takes twice as long or longer for the luminous flame to propagate over the whole chamber. Images of water/diesel emulsion show that strong micro-explosions of a group of droplets may occur in the luminous flame near the spray tip. They affect the local shape and brightness of the flame by dark and round regions due to explosions of superheated water in the droplets. The sizes of the micro-explosions range from barely identifiable small ones to those with diameter of a few millimeters. Micro explosions of the emulsion fuels seem to enhance mixing of the fuel with the surrounding air for faster and more efficient combustion.

The influence of water injection on combustion (pilot and main injections) has also been studied and shows similar effects as EGR. However, these effects are much more significant for water injection than for EGR for a given dilution ratio. To keep the cycle efficiency, the readjustment of fuel injection parameters, when increasing the dilution ratio, is thus higher in the case of water injection than in the case of EGR (Park et al. 2000).

In general, the results reported suggest that the water emulsification has a potential to slightly improve the brake efficiency and to significantly reduce the NO_x, soot, unburned HC and PM emissions of a diesel engine. However, in order to optimize the emulsion formulation in terms of water content and internal structure, experimental work is practically unavoidable (Armas et al. 2005). By using a water/diesel emulsion in a diesel engine, the cold-start may become impossible. Therefore, the injection system should be purged before engine stop in order to use pure diesel fuel for the engine start (Maiboom and Tauzia 2011). Furthermore, reliability problems may occur at some injection components, in particular in high pressure pumps that might need some specific modifications. Long-term emulsion stability could also turn out to be problematic in a water/diesel emulsion distribution network. Maybe, a possible alternative could be an onboard emulsion fabrication, which could allow the water/fuel ratio to vary.

3.3.2 Natural Gas and Liquefied Petroleum Gas

Natural gas (*NG*) is one of the most important energy carriers today, because it is available in large quantities and its reserves are of the same magnitude as the crude oil reserves. Typical compositions of NG are approximately 92.7 % methane, 3.3 % ethane, 2.2 % nitrogen, 0.5 % carbon dioxide, 0.7 % propane, 0.1 % isobutane, 0.2 % N-butane, 0.1 % pentane, and 0.1 % hexane (Nwafor 2000). NG offers several advantages such as clean combustion, high availability, and an attractive price (Poompipatpong and Cheenkachorn 2011). Additionally, its relatively high auto-ignition temperature is suitable for higher compression engines (Selim 2001). Due to a low cetane number, an engine using natural gas requires injection of mineral diesel fuel as the pilot ignition fuel. Such engines are *dual fuel engines*, which need two fuel systems (Kowalewicz and Wojtyniak 2005). Previous studies investigated the characteristics of *dual fuel* operation in unmodified or slightly modified diesel engines (Selim 2001). For dual fuelling, the in-cylinder pressure and heat release rate are lower than for neat diesel fuelling. By increasing the pilot diesel fuel injection quantity, the pressure in the cylinder and the rate of its rise also increase. Furthermore, an increase of pilot fuel quantity extends the lean burning limit and decreases HC and CO emissions, which are generally higher than for diesel fuelling. The ignition delay is longer for dual fuelling (Cordiner et al. 2008; Kowalewicz and Wojtyniak 2005). The combustion characteristics of a diesel engine with natural gas/diesel fuels show positive effects on thermal efficiency, total specific fuel consumption, soot, and NO_x emissions (Selim 2001). However, the reports indicate that such dual operation system cannot reach high speed operation that is obtainable by a diesel-only engine. Therefore, adequate modifications might be necessary to mitigate this negative effect.

Liquefied petroleum gas (*LPG*) is a mixture of propane and butane and comes with significant variations of its composition in various countries. An engine can be fuelled with a lean homogenous mixture of LPG and air and pilot diesel fuel injection can be used for ignition. In general, LPG usage delivers lower smoke, lower CO at high load, and lower NO_x at low and middle loads (Qi et al. 2007; Kowalewicz and Wojtyniak 2005). The usage of LPG requires only a minor modification of the diesel engine.

3.3.3 Methane and Propane

Compared to mineral diesel, methane has a higher heating value and lower adiabatic flame temperature. Many existing diesel engines can be relatively easily converted to dual fuel operation with methane, keeping the same compression ratios and a diesel-like efficiency, but with much lower emissions of NO_x and PM. On the other hand, direct methane injection is characterized by some practical complications, due to the necessity of a high pressure supply and to the problem of

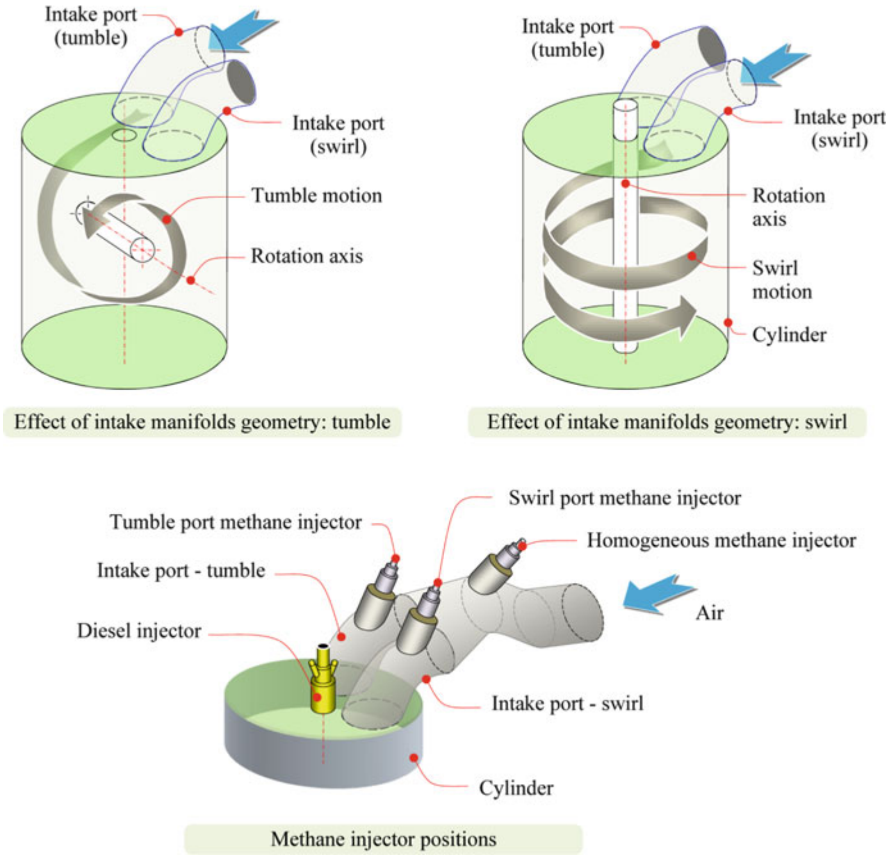


Fig. 3.15 Intake manifold geometries and methane injection positions

positioning another injector into the cylinder head. One possibility to overcome these technical difficulties in *dual fuel* mode is the injection of methane and preparation of *air–methane mixture* in the intake manifold (Carlucci et al. 2011; Sahoo et al. 2009; Abd-Alla et al. 2001).

Several options for methane injection are possible as illustrated in Fig. 3.15. In the case of *homogeneous injection*, a *homogeneous-like air–methane mixture* is produced by injecting methane along the intake air flow at a distance of about 40 cm from the cylinder axis. On the contrary, in order to obtain a certain degree of *mixture stratification* up to the end of the compression stroke, a *port injection* is proposed, in which methane is injected near the inlet valves. In the case, when engine has two intake pipes, two different injection positions are possible (Fig. 3.15). In the case of *swirl port injection*, methane is injected near the swirl inlet valve, while, in the case of *tumble port injection*, the injector is placed very close to the tumble inlet valve.

In Carlucci et al. (2011) the interactions of the methane injection strategies with the various in-cylinder charge motions have been investigated. For this purpose, the

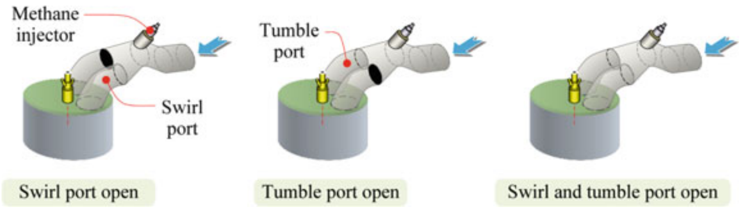


Fig. 3.16 Activation/deactivation of the engine intake ports

swirl helicoidal port and the tumble direct port of the engine intake were activated and deactivated in order to obtain three different in-cylinder bulk motions, denominated as the *swirl*, *tumble*, and *swirl/tumble* ports (Fig. 3.16).

At *low load*, no relevant effects of the methane supply method on the heat release histories were observed (Abd-Alla et al. 2001; Carlucci et al. 2011). On the contrary, the analysis of the combustion images and the luminance curves demonstrated that, for several combinations of the engine-operating parameters, port injection can probably induce some stratification effects on the in-cylinder charge, modifying the intensity and the spreading of the combustion flame. This happened especially when a suitable in-cylinder bulk motion was produced. Concerning pollutant emissions at low load, it was revealed that pilot diesel fuel injection pressure is a very important factor. At high pilot injection pressure, the *swirl port* delivers the lowest levels of HC and NO_x.

At *high load*, the swirl port injection engine shows higher peaks on the heat release curves and a generally more rapid development of the combustion with respect to the swirl/tumble port, while the tumble port shows an intermediate trend (Carlucci et al. 2011). At high loads, the improving effect of swirl port bulk motion affects the combustion, starting from the first steps of the oxidation process. No relevant effects of the methane supply method on the heat release histories were observed. Concerning the pollutant emissions at high load, it was revealed that, with respect to low load tests, the variation of pilot diesel fuel quantity affects less seriously the NO_x production. Moreover, the swirl inlet configuration showed the highest NO_x levels, the lowest ones were obtained with the swirl/tumble ports (especially with methane port injection), while the tumble port showed intermediate values. Regarding the swirl/tumble ports inlet configuration, by changing from homogeneous to port methane injection, a decrease of the NO_x levels was observed, especially when the port injection was performed into the swirl intake duct.

By comparing the low and high load emission results, it is important to note that, for several combinations of the operating parameters, methane port injection was always associated with the lowest values of pollutants emissions. This demonstrates that the port injection, as a methane supply method for dual fuel engines, is a very effective strategy to reduce unburned HC and NO_x concentrations, especially when implemented with variable intake geometry systems to produce suitable in-cylinder bulk motion and turbulence intensity for various engine operating parameters.

The dual fuel engines can also use *mineral diesel fuel* and *propane*. The results show that under the same operating condition, the effective thermal efficiency increases with the increase of propane proportion in the blends (Sahoo et al. 2009; Ma et al. 2008). Increased propane part also leads to decreased ignition delay and combustion duration. Furthermore, the maximum in-cylinder pressure, maximum rate of pressure rise, maximum rate of heat release, and maximum mean combustion temperature increase. The exhaust HC and smoke emissions decrease, while only minor variations of CO levels have been observed, except at high engine load, where CO levels also decreased. The NO_x emission increases with the increase of propane proportion in the blend.

3.3.4 Dimethyl Ether and Dimethyl Carbonate

Dimethyl ether (*DME*) is a liquefied gas that can be produced from a variety of feed stock. It is nontoxic and environmentally benign. DME has a high cetane number, low density, low lubricity, and it is more corrosive than mineral diesel (Arcoumanis et al. 2008). Fast evaporation of DME allows for good mixing with air in the engine cylinder and its high oxygen content can assure smokeless combustion through low formation and high oxidation rates of particulates. DME requires a high injected volume to deliver the same amount of energy as the diesel fuel, due to its lower density and combustion enthalpy. Furthermore, DME-fueled systems also need lubricity enhancing additives and anticorrosive sealing materials to assure leakage-free operation.

The DME spray characteristics and the corresponding combustion process have been investigated for various fuel injection systems, ranging from mechanically controlled injection systems to advanced common rail systems, in a variety of prototype engines with minor modifications (Arcoumanis et al. 2008; Kim et al. 2011; Youn et al. 2011; Zhu et al. 2012).

The DME engine is smoke free and this offers an opportunity to reduce NO_x emission without the trade-off, related to increased PM. By retarding the injection timing of a *mechanically controlled injection system*, a significant decrease of NO_x emission can be achieved, while the CO and formaldehyde emission increase to some extent (Zhu et al. 2012). In *common rail systems*, the injection quantity of DME fuel at the same injection conditions (injection pressure, energizing duration) is larger than that of mineral diesel due to the high return fuel pressure. Although the DME fuel has low fuel density, the high return fuel pressure prolongs the real injection duration of DME. Also, the spray tip penetration and spray cone angle of DME are longer and wider than those of mineral diesel. Among the combustion characteristics, the peak combustion pressure and the ignition delay of DME fuel are higher and shorter than those of mineral diesel, respectively. The DME fuel causes fast burning and reduces the ignition delay because it has a more oxygenated chemical structure and higher evaporating characteristics than diesel. The NO_x emission of DME is slightly higher than that of mineral diesel at the same engine load condition due to the high peak combustion pressure, the higher oxygen content, and the active combustion of the DME. The oxygenated component and volatility of DME result in HC and CO emissions being lower than those of mineral diesel (Kim et al. 2011; Youn et al. 2011).

Overall, DME has been found to be a very promising alternative fuel for diesel engines, capable of providing high thermal efficiency, low combustion noise, and soot free combustion; it thus merits further research and development to evaluate better its potential as a mass production fuel for the automotive market (Arcoumanis et al. 2008).

Dimethyl carbonate (*DMC*) is an oxygenated renewable fuel, which is usually used as an oxygenated additive to blend with mineral diesel in order to improve the combustion and reduce emissions (Cheung et al. 2011; Xiaolu et al. 2006). DMC reduces smoke almost linearly with its concentration, which is directly related to the oxygen content of the fuel. With 10 % of DMC content in mineral diesel, a smoke reduction of 35–50 % is attainable. Furthermore, reductions of HC and CO levels can be expected, but NO_x emissions may increase slightly (Xiaolu et al. 2006).

Attempts have also been made to use DMC as a primary fuel. In this context it should be noted that it is difficult to fuel diesel engines with DMC only, due to its low cetane number and high latent heat of vaporization. Some investigations of DMC combustion in diesel engines show the advantages of combining EGR with a small injection of mineral diesel by an auxiliary pump in order to ignite the DMC. Experiments on a single cylinder two-stroke diesel engine show that DMC also has a positive influence on the spray development (Xiaolu et al. 2006). DMC sprays have smaller atomization particles and more uniform distribution with respect to mineral diesel sprays. A DMC-fueled engine may have a 2–3 % higher effective thermal efficiency than a mineral diesel-fueled engine at moderate and high load conditions, partially due to its lower exhaust gas temperature. At low load operation, the brake thermal efficiency of the DMC fueled engine is slightly lower than that of the mineral diesel fueled engine, owing to higher HC emissions. A DMC-fueled engine with EGR and a small amount of mineral diesel for the ignition has a potential for simultaneous reduction of NO_x and smoke emissions by keeping CO emission almost the same. Moreover, the smoke density can be brought almost to the zero level. It has to be kept in mind that the mineral diesel injection timing influences the NO_x, CO, and HC emissions of the DMC-fueled engine. With the advanced diesel injection timing the NO_x emissions increase, while the CO and HC emissions decrease.

3.3.5 Fischer–Tropsch Diesel

Fischer–Tropsch diesel (*FTD*) is called the fuel obtained from synthetic gas in the presence of a catalyst, at high pressure and temperature. The basic mechanism of the *Fischer–Tropsch process* is $2nH_2 + nCO = n(-CH_2-) + nH_2O$.

In order to reduce NO_x emissions in diesel engines, a typical strategy is to retard the injection at the cost of penalizing the fuel consumption. Because FTD has a higher cetane number than mineral diesel, FTD usage leads to shorter ignition delays (Bermúdez et al. 2011; Gill et al. 2011). This results in an improved specific fuel consumption and thermal efficiency, while maintaining NO_x at an acceptable level. However, the advantages associated with a high cetane number diminish at higher compression ratios, where the soot levels increase due to the reduced premixed combustion. The overall combustion parameters of FTD fuels are very sensitive to high levels of EGR, particularly soot emission which increases dramatically as the level of EGR increases. Since FTD has lower distillation characteristics, the improvement in atomization and dispersion of fuel spray enables faster evaporation, accelerating the fuel mixing with air. In addition, the rates of heat release for such fuels are lower and therefore improve the combustion noise. However, using pilot injection in conjunction with a common rail injection system can also assist the reduction of combustion noise.

3.3.6 Hydrogen

Because of the high self-ignition temperature, pure hydrogen can’t be used directly in a diesel engine. Some alternative methods are the *hydrogen enrichment technique*, *hydrogen port/manifold injection*, and *hydrogen into the cylinder injection*, which use mineral diesel as a pilot fuel for the purpose of ignition (Saravanan and Nagarajan 2010).

By the *hydrogen enrichment technique* air is enriched with hydrogen by using a venturi/gas carburetor in the intake manifold. A pilot quantity of diesel is used as an ignition source. The benefit of hydrogen enrichment is that the brake thermal efficiency increases, but the power output of the engine drops due to the partial replacement of air by gaseous hydrogen.

By *hydrogen port/manifold injection*, hydrogen is injected in the intake port by using a mechanically or electronically operated injector. The position of the injector on the manifold determines whether the system is a port fuel injection system or a manifold injection system. For both systems, mineral diesel is usually taken as the ignition source. The advantage of hydrogen injection over the carbureted system is that with proper injection timing some problems, related to backfire and preignition, can be eliminated.

By *hydrogen into the cylinder injection*, hydrogen is injected directly into the combustion chamber at the end of the compression stroke. During idling or part load conditions, the efficiency of the engine may be reduced slightly. In spite of that, compared to other methods of hydrogen usage, this method is the most efficient. Two types of injectors are available for the use in direct injection systems.

The *low pressure direct injector* injects the fuel as soon as the intake valve closes when the pressure is low inside the cylinder. The *high pressure direct injector* injects the fuel at the end of the compression stroke.

The major problem associated with the use of direct injection is that it must be able to withstand higher combustion temperature in addition to preventing the injector from corrosion due to exhaust gases. Lubrication between the injector's moving parts also makes the design of the direct injector more complicated (Saravanan and Nagarajan 2010).

By using hydrogen as a supplementary fuel in a diesel engine, the brake thermal efficiency increases due to a better combustion process (Saravanan et al. 2008) and the specific energy consumption decreases due to the operation of hydrogen fueled engine under lean burn conditions. The effect of hydrogen addition on the combustion process of heavy duty diesel engines depends on the load and the amount of hydrogen added. Addition of a small amount of hydrogen has only a minor or negligible effect on the cylinder pressure and combustion process. To change this situation, a relative large amount of hydrogen is needed (Liew et al. 2010). Hydrogen addition yields modest emission reductions with a limited penalty on engine performance (Lilik et al. 2010).

3.3.7 Alcohols

Alcohols, mainly *methanol* and *ethanol*, have been widely investigated in combination with mineral diesel, with the aim to reduce NO_x and particulate emissions (Agarwal 2007; Kowalewicz and Wojtyniak 2005; Abu-Qudais et al. 2000; Surawski et al. 2010). Methanol is biodegradable, toxic, and corrosive, while ethanol is not considered toxic and is also biodegradable. Methanol is produced mostly from coal and natural gas, but meanwhile it can also be produced from renewable sources, such as wood or waste paper. Ethanol is produced from biomass, such as potatoes, beets, sugar cane, wood, brewery waste and many other agricultural products, and food wastes in the process of fermentation. Additionally, it can also be produced from natural gas and crude oil.

Alcohols and mineral diesel are mostly applied together, either in the *blended mode* or in the *fumigation mode*. In comparison with the blended mode, the fumigation approach seems to be more flexible, despite of the extra fuel injection system required. This is because, firstly, it allows the amount of injected alcohol to vary in dependence on actual requirements. Secondly, since the alcohol is not premixed with mineral diesel, an emulsion additive, to ensure proper mixing of alcohol and diesel, is not required.

A number of studies have demonstrated that NO_x and particulate matter can both be reduced with *methanol* or *ethanol fumigation* (Cheng et al. 2008; Zhang et al. 2009). In general, *methanol fumigation* (Fig. 3.17) was found to decrease NO_x emission and smoke opacity and to have a beneficial effect on fuel efficiency at high engine loads. Some investigation results show a reduction of NO_x and particulate emissions, but an increase of HC and CO emissions (Zhang et al. 2010; Wang et al. 2008; Yao et al. 2008). Furthermore, *ethanol fumigation* influences gaseous emissions and the in-cylinder pressure shows that HC and CO emissions may increase severely,

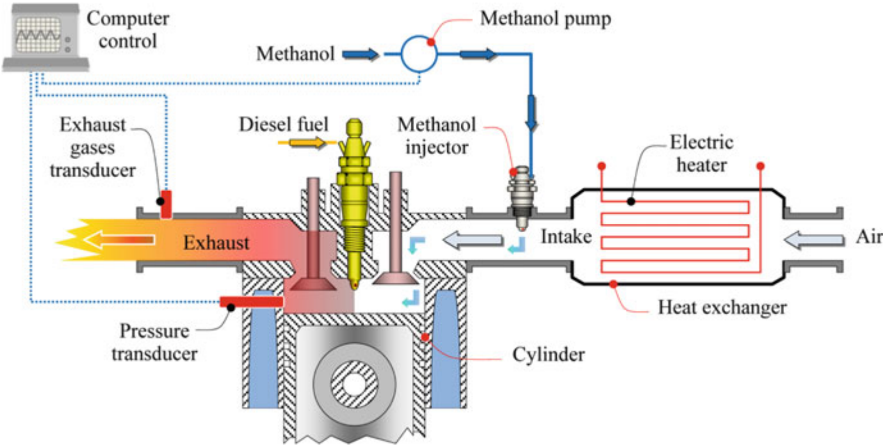


Fig. 3.17 Methanol and diesel fuel injection systems

compared with diesel fuel, but the NO_x emissions usually decrease (Hayes et al. 1988). The maximum rate of pressure rise and peak pressure is also significantly higher (Jiang et al. 1990). Ethanol fumigation could be a viable method to offset mineral diesel while lowering diesel soot emissions and avoiding lubricity problems, potentially associated with ethanol–diesel-blended fuels (Leahey et al. 2007).

3.3.8 Vegetable Oils, Bioethanol, and Biodiesel

Vegetable Oils. Vegetable oils also offer a potential environmental advantage. The most frequently investigated raw *vegetable* oils for diesel engine usage are among edible oils: *corn*, *palm*, *peanut*, *rapeseed*, *sesame seed*, *soybean*, and *sunflower* oil, and among nonedible oils: *cottonseed*, *jatropha*, and *rubberseed* oil (No 2011; Hossain and Davies 2010; Demirbas 2007; Kalam et al. 2003). The chemical and physical properties of these vegetable oils may differ from mineral diesel significantly, as shown in Fig. 3.18.

On average, the *density* of vegetable oils is about 12 % higher than that of mineral diesel, while their *energy content* is around 10 % lower as a result of lower hydrogen content (Hossain and Davies 2010). For most vegetable oils the *cetane number* is around 10–20 % lower than that of mineral diesel. The cetane number depends on the locations and number of double bonds in the molecular structure of the oils. Lower cetane number implies a larger ignition delay and tends to result in lower efficiency. The *viscosity* affects the flow of fuel and spray characteristics. Due to the large molecular size of the triglycerides, making up about 98 % of vegetable oils, viscosity is higher and volatility lower than for mineral diesel. Higher viscosity of vegetable oils leads to poorer combustion. The *flash point temperature* indicates the overall flammability hazard in the presence of air and

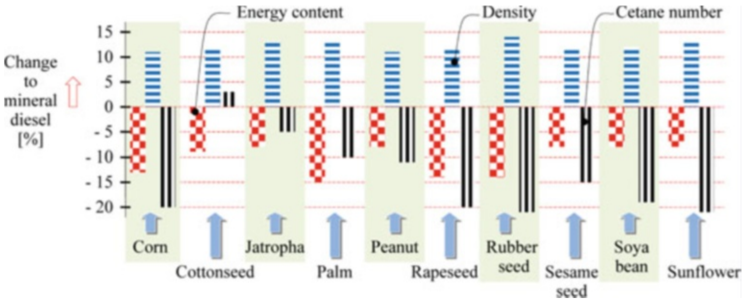


Fig. 3.18 Comparison of vegetable oils properties with respect to mineral diesel

the *pour point temperature* is a measure of fuel performance under cold temperature conditions. In the case of the edible vegetable oils, the flash point temperatures are higher, whereas the pour point temperatures are lower than for mineral diesel. On the other hand, for nonedible vegetable oils both of these temperatures are much higher than for mineral diesel. The *iodine value* is a measure of the number of double bonds and indicates oxidation stability. Iodine value increases with the level of unsaturation. Higher iodine value means lower oxidation stability. Oxidation can lead to polymerization. Iodine values vary significantly among the vegetable oils considered. The *carbon residue value* correlates with the carbonaceous deposits inside the combustion chamber and injector systems. For vegetable oils it is considerably higher than for mineral diesel.

Various properties of vegetable oils lead to various injection, atomization, and combustion characteristics of a diesel engine. Several investigations show significant performance variations obtained among various types of these oils (Ramadhas et al. 2004; Nwafor 2003; Bhattacharyya and Reddy 1994). Of course, those effects vary in dependence of engine load, engine speed, feedstock homogeneity, ambient conditions, engine type, and injection type. However, some general influence of vegetable oils usage on *effective power*, *effective specific fuel consumption*, and *thermal efficiency*, can be roughly estimated as follows.

By using vegetable oils, the *effective engine power* decreases by 2–18 %. The main reasons for this lie in lower energy content of vegetable oils and in higher viscosity, which leads to poor atomization and inefficient mixing of air and fuel. It has to be noted that in some cases (e.g., when using rapeseed and coconut oils), the peak in-cylinder pressure is lower than the one obtained with mineral diesel (Hossain and Davies 2010; Kalam et al. 2003; Nwafor and Rice 1996); meanwhile in some other cases (e.g., using orange oil), the peak in-cylinder pressure and heat release are higher (Hossain and Davies 2010; Purushothaman and Nagarajan 2009).

Effective specific fuel consumption of vegetable oils is the same or higher than that of mineral diesel. Most investigations show an increase in the range of 2–15 %. *Thermal efficiency* of vegetable oils is in the range from +3 to −10 % compared to mineral diesel. High viscosity and low volatility are again the reasons for this efficiency loss. Eventual efficiency gains may be explained by the lower energy content of vegetable oils.

A comparison of data for engine emissions shows a considerable spread. This is due to the variation of vegetable oil type, engine type, injection type, fuel inlet temperature, engine operating regime, lack of feedstock homogeneity, and ambient conditions. When the engine runs on vegetable oils, the *emissions* of *CO* and *HC* may either increase or decrease. At low load operation, the CO emission is almost the same as for mineral diesel. At higher loads however, the mixture becomes richer, meaning that more CO is produced due to the lower oxygen content of vegetable oils. Some investigations show a decrease in NO_x *emission* with vegetable oils, compared to mineral diesel (Kalam et al. 2003), but the opposite has also been observed (No 2011). Furthermore, the exhaust gas temperature and smoke intensity of vegetable oils may either increase or decrease in comparison to mineral diesel.

Vegetable oils can be successfully used in a diesel engine by making adequate engine and fuel modifications. The *engine modifications* include preheating/heated fuel line, dual fueling, injection system modifications, etc. Heating and blending of vegetable oils reduces the viscosity but its molecular structure remains unchanged and hence the polyunsaturated character and low volatility problems still exist. The *fuel modifications* include pyrolysis (thermal cracking), micro-emulsion, transesterification, and hydrodeoxygenation to reduce polymerization and viscosity. It may be worth noting that transesterification, where the kinematic viscosity can be remarkably reduced, is an effective process to overcome many problems associated with vegetable oils (No 2011).

Bioethanol. Bioethanol is produced from carbohydrates (hemicelluloses and cellulose) in lignocellulosic materials (Demirbas 2007). It can be derived from wheat, sugar beet, corn, straw, and wood. Bioethanol can be used as an additive to mineral diesel. The chemical and physical properties of bioethanol differ from those of mineral diesel. Consequently, the presence of bioethanol causes various physical and chemical property variations of the blended mineral diesel, mostly related to lower cetane number, lower energy content, viscosity, flashpoint, pour point, etc. (Torres et al. 2011a; Li et al. 2005). Lower cetane number worsens cold starting ability, increases noise, and shortens engine life. Density and viscosity decrease by addition of bioethanol, thus promoting retarded injection timing in mechanically controlled injection systems (Torres et al. 2011a). High bioethanol volatility may influence significantly injection timing and ignition and combustion characteristics. Furthermore, it can cause some problems related to storage. While the pour point decreases significantly with respect to that of pure diesel, the cloud point increases. Concentrations of bioethanol above 5 % lead to unwanted behavior in the cloud point test. The addition of bioethanol, however, does not modify substantially the cold filter plugging point (CFPP).

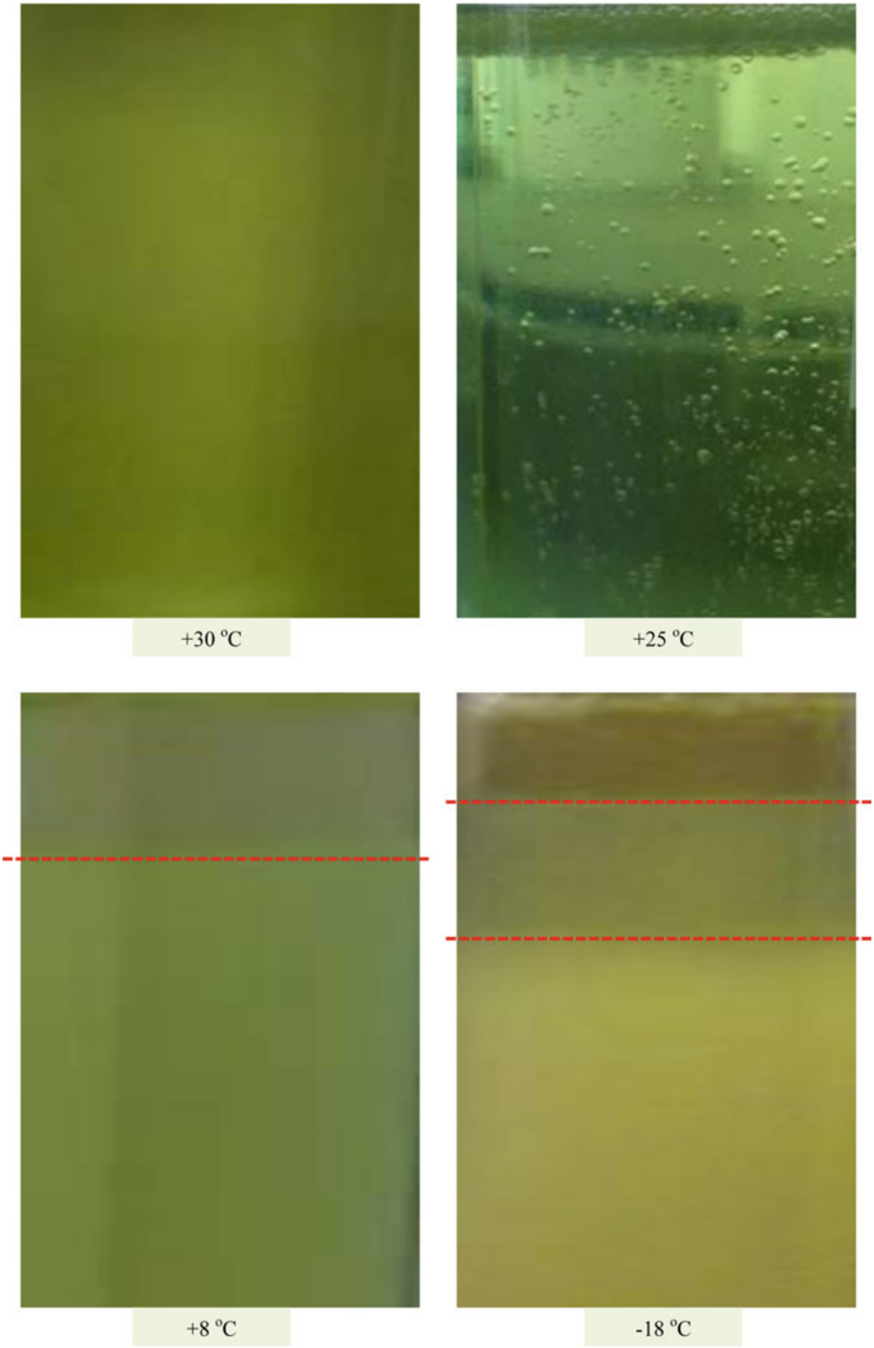
Another point worth notion is the *ethanol–diesel mixture stability*. In this context experimental tests are necessary in order to determine whether additional mixing, additives, or heating (in case that low temperatures produce gel formation) are necessary. In one of such experiments, (Torres et al. 2011a), samples of 5, 10, and 15 % of bioethanol (obtained by fermentation of sugars) in mineral diesel were tested at various temperatures: +25 °C (normal ambient temperature), +30 °C (summer), +8 °C (not critical winter), and −18 °C (critical winter in central Europe). Each sample was checked once a week during a 5-week period; it was assumed that this is enough time to estimate the blend behavior during long periods of storage. Figure 3.19 shows some photographs of the 15 % bioethanol and 85 % diesel mixture.

At +30 °C and after 4 weeks, no changes in stability, color, and aggregation state of the blends could be observed in any of the samples. However, already at +25 °C and after 4 weeks, the 10 % sample separated into bioethanol and diesel. Bioethanol bubbles, traveling to the surface, could be observed. At +8 °C in the 15 % sample bioethanol separated from diesel after only 1 week; two layers have been observed. At −18 °C in the 15 % sample three layers have been observed after 5 weeks; two layers were observed within the diesel fuel.

Obviously bioethanol/diesel separation depends on temperature, bioethanol concentration, and water content (Hansen et al. 2005; Lapuerta et al. 2007). The situation can be improved by using various kinds of additives (Weber et al. 2006; Reyes et al. 2009; Ribeiro et al. 2007). Nowadays, the addition of biodiesel, acting as a blend stabilizer, is becoming the preferred method (Lapuerta et al. 2009; Chotwichien et al. 2009; Lebedevas et al. 2009; Kwanchareon et al. 2007) because it has the advantage of increasing the biofuel concentration in the fuel, which is one of the targets proposed by the European Community in the Directive 2003/30/EC to promote the use of biofuels for transport.

Bioethanol addition to mineral diesel influences the injection and spray characteristics, combustion performance, and engine emissions (Torres et al. 2011b, 2010; Huang et al. 2009; Rakopolulus et al. 2008a, b; Rakopoulus et al. 2007; He et al. 2004). Investigations show that bioethanol addition to mineral diesel up to 30 % reduces smoke emissions at high loads, decreases NO_x emissions at low loads, but can somewhat increase NO_x emissions at medium and high loads. Bioethanol increases HC and CO emissions with the exception of CO emission at high loads. The combustion duration shortens and ignition retards by increasing the bioethanol content in mineral diesel. Bioethanol also improves the effective specific fuel consumption at full load.

Biodiesel. Biodiesel is a transesterified vegetable oil. Biodiesel is technically competitive with mineral diesel or even offers some technical advantages (Demirbas 2007). This can be even further improved by adding bioethanol. In this context the addition of bioethanol up to 15 % to biodiesel has been investigated. The *bioethanol–biodiesel blends* have some specific properties. In general, however, investigations show that bioethanol addition improves the most important fuel properties, related to the injection process and engine characteristics. It was also observed that *filter plugging tendency* improves proportionally to the bioethanol concentration. Owing to the hygroscopic nature of bioethanol and biodiesel, special attention must be paid during storage and transportation to avoid the absorption of water from the ambient humidity. Properties related to cold climates, such as cloud point, pour point, and CFPP, improve by bioethanol addition because phase separation does not occur and bioethanol shows good behavior at low temperatures. When testing fuel blends, it is important to note that the flash point will show the value of the more volatile of the components, i.e., bioethanol in this case. Therefore, additives are typically necessary to raise the blend flash points within the limits required by the standards.



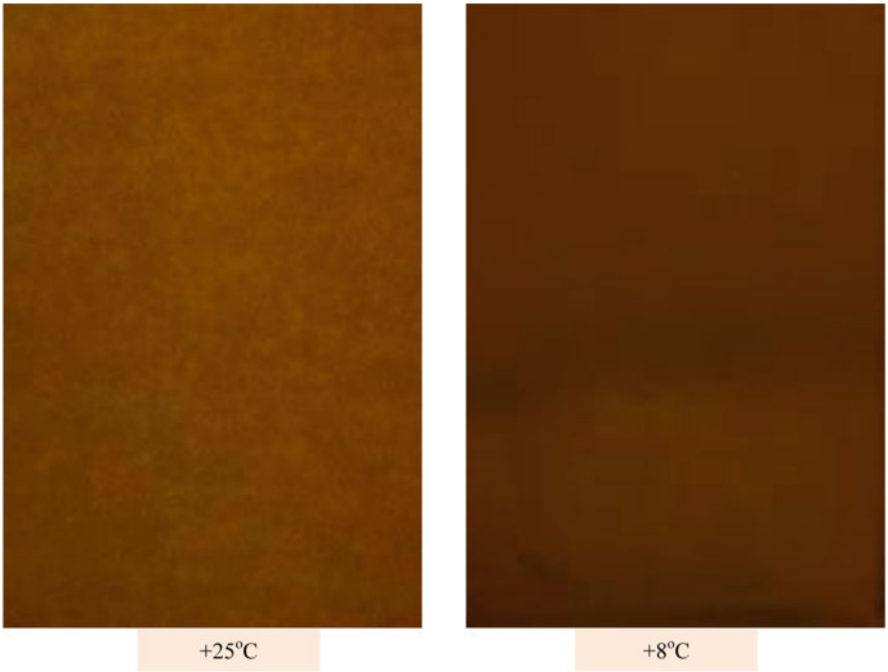
+30 °C

+25 °C

+8 °C

-18 °C

Fig. 3.19 Stability of bioethanol–diesel blend D85E15


+25°C

+8°C

Fig. 3.20 Stability of bioethanol–biodiesel blend B85E15

In order to evaluate the bioethanol–biodiesel blend stability, samples with up to 15 % of bioethanol were tested at +25 °C, +30 °C, +8 °C, and −18 °C (Torres et al. 2011b). Each sample was checked once a week during a 5-week period. When neat biodiesel warmed up from the solid state and became liquid, some white particles could easily be observed (Fig. 3.20). These particles are paraffins. In case of bioethanol–biodiesel blends, bioethanol changes the melting process in such a way that paraffins do not become visible. Invisibility of the paraffins is an improvement from the point of winter properties, such as the cloud point and cold filter plugging point. Namely, these properties are linked to the moment at which paraffins appear during the solidification process. This means that bioethanol also acts as a winter additive. Furthermore, bioethanol–biodiesel blends need less time to become liquid than neat biodiesel.

3.4 Discussion

To improve the worldwide acceptance of diesel engines, it is necessary to increase engine effective power and torque and reduce fuel consumption and harmful emissions, especially NO_x and smoke emissions.

The NO_x concentration can be reduced by a reduction of the flame temperature, which can be achieved, for example, by lower oxygen content or increased content of inert gas in the intake air (as with EGR) or by better premixing in order to achieve partly homogeneous lean mixture before the start of combustion. Reduction of smoke concentration in the exhaust gas can be achieved either by decreasing smoke formation or increasing smoke oxidation. Smoke formation can be mitigated by more thorough mixing that prevents the appearance of rich flames with local relative air/fuel ratios below 0.6. Oxidation can be improved by sustaining a high temperature level for a sufficiently long time period in the presence of oxygen. Unfortunately, this is counterproductive with regard to the NO_x problem. Anyhow, because there are many trade-offs in the quest for a better diesel engine, the development process is quite tedious. In spite of this, some general guidelines may still be given and the measures that can be implemented can be outlined as follows:

- Improved in-cylinder gas flow such as controlled swirl and turbulence through a variable intake geometry, multivalve cylinder head, and optimized bowl shape
- Improved injection system with a vertical nozzle, injection rate shaping such as pre-injection and split injection or a shorter deactivation period, and finer sprays, achieved by an electronically controlled injection system with injection pressures up to 2,000 bar (e.g., by unit-injector or common-rail systems)
- Improved and controlled combustion procedure through reduced flame temperature (higher EGR level, cooled EGR, better premixing), avoidance of over-rich zones, and faster start of oxidation
- Improved fuel quality and usage of alternative fuels with lower sulfur and aromatic content and increased cetane number

The influence of exhaust gas after treatment, engine management, and alternative fuels on engine characteristics was and still is a lively investigated topic. The interdependent relationships between the involved quantities are quite sophisticated. In spite of that, much helpful insight can be gained by a careful study of many thorough investigations, for example, like the one provided by (Armas et al. 2010). In this investigation the effects of *injection control* and *alternative fuels* on engine characteristics have been investigated by using the DDC/VM Motori 2.5 L turbocharged diesel engine with a *common rail injection system* and an ultra low sulfur mineral diesel fuel (D100), *soybean biodiesel* (SoBIO) and *Fisher–Tropsch* (FT) fuel. Test were run at 2,400 rpm and 64 Nm torque with *single* and *split* (pilot and main) *injection strategies* and without exhaust gas recirculation (EGR) In the case of single injection, the experimental results have been obtained by using advanced (5.83°CA BTC), baseline (3.83°CA BTC), and retarded injection timing (1.83°CA BTC). In the case of split injection strategy, advanced (31.4°CA BTC), baseline (29.4°CA BTC) and retarded (27.4°CA BTC) injection start was used for the pilot injection. The main injection started always at the same angle after top center (1.4°CA ATC). The results are summarized in the following.

Figure 3.21 shows that effective specific fuel consumption increases with SoBIO and decreases with FT fuelling when the injection timing and other parameters are

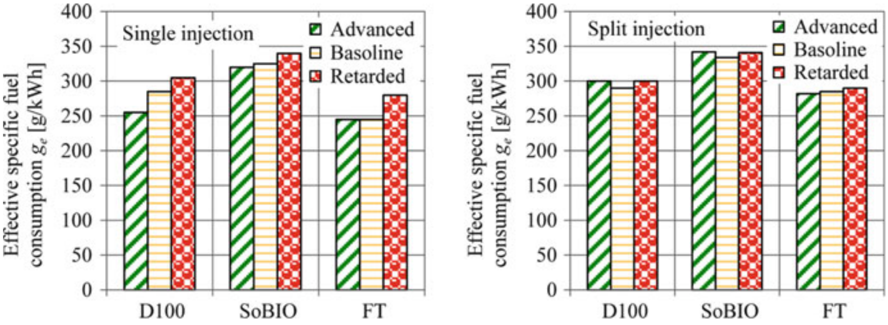


Fig. 3.21 Effective specific fuel consumption for various injection timings and fuels

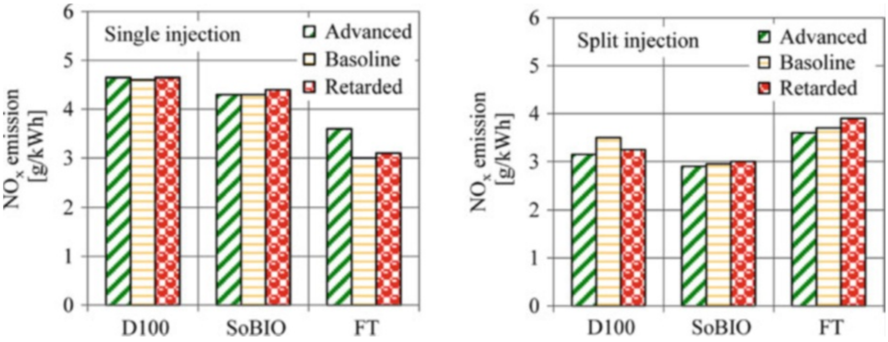


Fig. 3.22 NO_x emission for various injection timings and fuels

constant at single and split injection. Compared to D100, the effective specific fuel consumption of SoBIO fuel increased approximately by 12 % for single and 15 % for split injection. This fact is consistent with the lower energy content of SoBIO, which is about 12.9 % lower than the energy content of D100. Compared to D100, a decrease of g_e for the FT fuel (2 % for the advanced injection timing, 8 % for the delayed injection timing) was observed. This is consistent with the energy content of FT fuel, being 2.3 % higher than that of D100. The results show that in the case of split injection various injection timings have no significant effect on g_e for any of the tested fuels.

Figure 3.22 shows that NO_x emission is lower when SoBIO and FT fuels are used instead of D100. Independent of the injection timing, the reduction of NO_x emissions was approximately 7 % when SoBIO is used. Compared to mineral diesel, the use of FT led to a reduction of NO_x emissions approximately by 22 % and 33 %, when operating with advanced and retarded injection timing, respectively. The influence of the start of pilot injection on NO_x emission is less significant than the influence of fuel composition.

Figure 3.23 shows that PM emission of SoBIO is high with the retarded injection timing in the case of single injection strategy. This is because the delayed

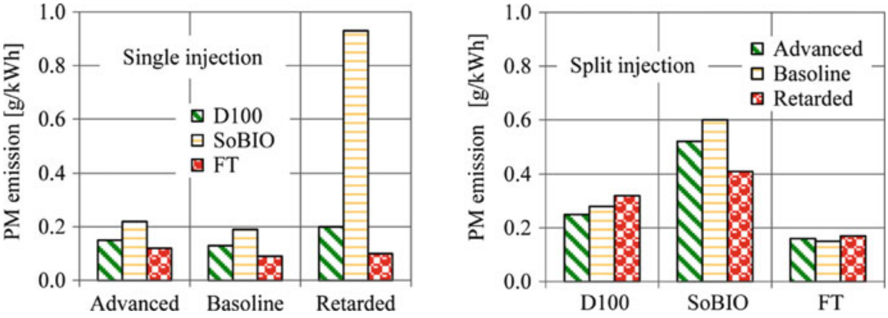


Fig. 3.23 PM emission for various injection timings and fuels

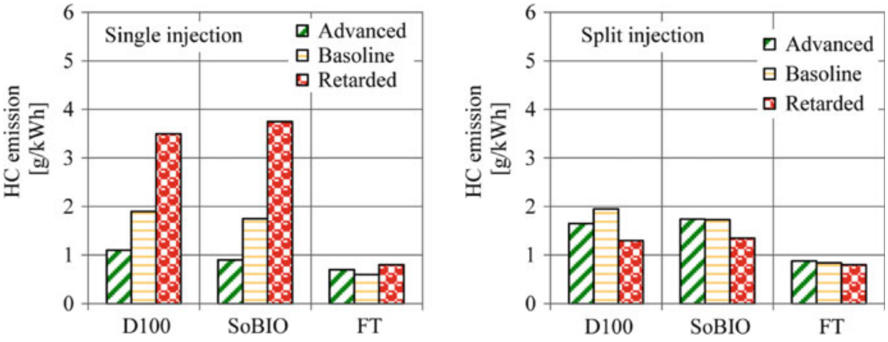


Fig. 3.24 HC emission for various injection timings and fuels

combustion process (being practically without premixed combustion phase) is combined with a flat high temperature distillation curve of SoBIO.

Considering the NO_x–PM trade-off observed for D100, SoBIO, and FT, when using the single injection strategy, the emissions results show that biodiesel usage can lead to a slight decrease in NO_x emissions when the engine is operated within the range of the considered injection timing. However, this operating condition can produce an increase in PM emissions, which increase as the injection timing is retarded. This trend can be explained by the longer injection duration as a consequence of the lower energy content of biodiesel. The FT fuel produced a significant decrease of both NO_x and PM emissions, independently of the tested injection timing. In the case of split injection strategy, fuel composition has a greater impact on the emissions than the pilot injection timing. As with the single injection strategy, the FT fuel produces the lowest PM emissions without a significant increase of NO_x emissions.

With single injection strategy, the HC emission increased by retarding the injection timing for D100 and SoBIO (Fig. 3.24). When FT fuel is used, the level of HC emissions is practically independent of the injection timing, being lower than for D100 by 38, 67, and 78 % for advanced, baseline, and retarded injection timing, respectively. With the split injection strategy, the HC emission, produced by D100

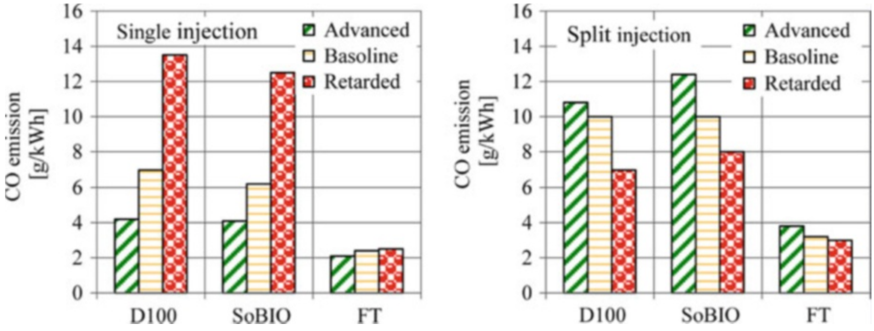


Fig. 3.25 CO emission for various injection timings and fuels

and SoBIO, slightly decreases when the pilot injection is retarded (Fig. 3.24). In all tests, the FT fuel has the lowest HC emission compared to D100 and SoBIO. SoBIO produced the highest PM emission in all the tested starts of pilot injection.

In the case of single injection, the CO emissions follow the same trend as HC emissions, being lower than those produced by D100 by 56, 70, and 81 % for advanced, baseline, and retarded injection timing, respectively (Fig. 3.25). In the case of split injection, the CO emissions produced by D100 and SoBIO show a slight decrease when the start of pilot injection is retarded. In all tests, the FT fuel has the lowest CO emission compared to D100 and SoBIO.

The tests described in the forgoing text were performed without making use of the exhaust gas recirculation (EGR). It should be noted, however, that EGR usage may, in general, reduce the total engine NO_x emissions significantly.

Diesel exhaust also contains sulfuric salts and other abrasive and corrosive substances (Zheng et al. 2004). It has been argued whether EGR should be applied to diesel engines because of the increased piston–cylinder wear. Heavy use of EGR can also deteriorate the energy efficiency, operational stability, and PM generation of the engine. In spite of that, the concerns about increased wear and deteriorated performance have been somewhat pushed into background, because of ever-stringent emission regulations. Therefore, the current concern is mostly about how aggressively EGR should be applied at various speeds and loads. It should be noted, however, that increased wear, related to EGR, continues to be a problem, affecting engine durability and performance.

Currently, EGR is still the most viable technique to reduce NO_x substantially. Energy efficient after treatment systems, dealing with NO_x and PM simultaneously, are still in the early development stages. The inability of available catalytic after treatment technologies further encourages the use of EGR.

Another topic worth of attention are *vegetable oils*. Some general advantages and disadvantages of their *usage* in *diesel engines* have been summarized in the *strength–weakness–opportunities–threat* (SWOT) analysis by (Misra and Murthy 2010; Russo et al. 2012).

Strengths

- Vegetable oils are a renewable energy source and they are classified as harmless to groundwater, according to the German system of water hazard classes.
- There exists a fuel standard for neat vegetable oils in Germany, DIN V 51605.
- Vegetable oils balance the CO_2 in the environment (CO_2 neutral to environment), oilseed-borne trees also remove carbon from the atmosphere, store it in the woody tissues, and assist in the buildup of soil carbon; they are thus environment friendly.
- The fuel production technology is simple and proven and suitable for decentralized production in small oil mills, providing increasing employment in rural areas.
- A number of nonedible oil crops is perennial and is not affected by climatic changes.
- The cetane number is similar or close to that of mineral diesel.
- The energy contents of various vegetable oils are nearly 90 % of that of mineral diesel.
- A higher flash point of vegetable oils allows them be stored at high temperatures without any fire hazard; their flashpoint makes them easy to handle and store.
- By mixing a vegetable oil into the mineral diesel at low ratios, the engine performance and exhaust emissions are, in general, improved with respect to neat mineral diesel.
- The additional oxygen molecule in its chemical structure is beneficial for the combustion process.
- Vegetable fuels seldom contain sulfur, which is not the case for mineral diesel.
- Compared to mineral diesel, NO_x emissions are drastically lower, up to around 30 % for 100 % straight vegetable oil.
- Vegetable oils are easily available in rural areas where their usage is advantageous, especially for smaller engines in agriculture.

Weaknesses

- Non-modified diesel engines can't run on neat vegetable oils.
- Vegetable oils in their natural form have higher viscosity compared to mineral diesel.
- Variable output, variable oil content, long gestation period of the crops.
- At present their availability is scattered.
- There are no large-scale production plants.
- The economic viability depends on seed yields and the income from by-products (press cake).
- In Europe there exists no common standard for neat vegetable oils.
- The presence of chemically bound oxygen in vegetable oil lowers their energy content.
- Very high viscosity and low volatility of vegetable oil lead to poor fuel atomization and lower engine performance.
- Lube oil dilution, high carbon deposits, ring sticking, scuffing of the engine liner, and injection nozzle failure are the major problems, associated with direct use of straight vegetable oils.

- The high flash point attributes to lower volatility characteristics.
- Both, cloud and pour points are significantly higher than that of mineral diesel; these high values may cause problems during cold weather.
- Long storage problems related to viscosity; long-term storage of vegetable oils may cause degradations of certain fuel properties; additives are needed.
- No commercial output is available without ample farming inputs.

Opportunities

- There are a number of nonedible oils which are good fuels; about 300 varieties of tree born oilseeds have been identified; selected crops can be grown on arid and semiarid lands which are presently not cultivable.
- Vegetative propagation is possible in many varieties.
- They have carbon credit value (Kyoto protocol).
- There is a relatively large demand worldwide owing to environmental problems.
- Mineral diesel fuel dissolves quite well with vegetable oils.
- Preheating, blending of vegetable oils with diesel fuel, and blending vegetable oils with solvents would greatly remove many problems associated with diesel engine operation with neat vegetable oil.
- A great opportunity for reduction of NO_x emissions.
- Energy policies that provide tax reductions/exemptions and biofuel obligations could largely increase use of vegetable oils,
- Their use and development can enhance energy supply security.
- Waste oil can be used as cheap feedstock for vegetable oil.
- Other oil plants than rapeseed can be cultivated in Europe (e.g., sun flower, soya).
- There is a significant potential of employment generation capacity in rural areas.

Threats

- The behavior of the plants can vary in dependence of agro climatic zones and regions.
- It is still to be established how these plants would behave once removed from its original habitat and put under high density and intensive cropping system.
- Costly input materials at present.
- As of now these oils are difficult to sustain economically without subsidies.
- No sustainable procurement mechanism available in the market.
- Requirement of seeds in large quantity, even for the modest scenario of 5 % blending with mineral diesel.
- In Europe, mainly rapeseed is used as feedstock source; rapeseed can be cultivated only every 4 years on the same field.
- No standard for vegetable oils feedstocks exists, which means the variation in feedstock properties could vary for many reasons.

Direct use of vegetable oils is impractical and unsatisfactory either for direct or indirect diesel engines (Bozbas 2008). The most important short-term and long-term problems, their probable reasons, and the potential solutions are known to a great extent. Briefly, they can be summarized as follows.

Short-term problem: *cold weather starting*
Probable cause: high viscosity, low cetane number, low flash point of vegetable oils
Potential solution: heat fuel prior to injection process chemically alter the fuel to ester

Short-term problem: *plugging and gumming of filters, lines, and injectors*
Probable cause: natural gums in vegetable oils
Potential solution: partially refine the oil to remove gums

Short-term problem: *engine knocking*
Probable cause: very low cetane number of some oils, unsuitable injection timing
Potential solution: adjust injection timing, heat fuel prior to injection, and chemically alter the fuel to ester

Long-term problem: *coking of injectors on the piston and head of the engine, carbon deposit on the piston and head of the engine*
Probable cause: high viscosity of vegetable oils, poor combustion at partial load
Potential solution: heat fuel prior to injection process chemically alter the fuel to ester

Long-term problem: *excessive engine wear*
Probable cause: high viscosity of vegetable oils, poor combustion at partial load, possible free fatty acids in vegetable oils, and dilution of engine lubricating oil due to blow-by of vegetable oils
Potential solution: heat fuel prior to injection process chemically alter the fuel to ester, increase motor oil changes and motor oil additives to inhibit oxidation

Long-term problem: *failure of engine lubricating oil due to polymerization*
Probable cause: collection of polyunsaturated vegetable oil blow-by in crankcase to the point where polymerization occurs
Potential solution: heat fuel prior to injection process, chemically alter fuel to ester, increase motor oil changes and motor oil additives to inhibit oxidation

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