İSTANBUL TECHNICAL UNIVERSITY ★ INSTITUTE OF SCIENCE AND TECHNOLOGY

IMPACT OF DIFFERENT NOZZLE SPECIFICATIONS ON ENGINE EMISSIONS

M.Sc. Thesis by

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Department : Mechanical Engineering

Programme: Automotive Engineering

HAZİRAN 2008

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Date of submission : **5 May 2008**

Date of defence examination: **9 June 2008**

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JUNE 2008

İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

FARKLI ÖZELLİKLERE SAHİP ENJEKTÖR MEMELERİNİN MOTOR EMİSYONLARINA ETKİSİ

YÜKSEK LİSANS TEZİ

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Tezin Enstitüye Verildiği Tarih : 5 Mayıs 2008

Tezin Savunulduğu Tarih : 9 Haziran 2008

Tez Danışmanı : Prof. Dr. Cem SORUŞBAY

Diğer Jüri Üyeleri Prof.Dr. Metin ERGENEMAN

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HAZİRAN 2008

ACKNOWLEDGEMENT

I would like to express my deep appreciation and thanks to my advisor Prof. Dr. Cem SORUŞBAY for his support and consultancy. This work is supported by the facilities of Ford Otosan and Ford Motor Company, Dunton.

May 2008

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ABBREVIATIONS

PM	: Particulate Matter
DI	: Direct Injection
SMD	: Sauter Mean Diameter
CA	: Crank Angle
NOx	: All Compounds of Nitrogen and Oxygen
CO	: Carbon Monoxide
HC	: Hydrocarbon
H	: Hydrogen
C	: Carbon
EGR	: Exhaust Gas Recirculation
VGT	: Variable Geometry Turbocharger
CAE	: Computer Aided Engineering
SEI	: Sharp-Edged Inlet
RI	: Rounded Inlet
FSN	: Filter Smoke Number
NEDC	: New European Driving Cycle
OBD	: On-Board Diagnostics
ECU	: Engine Control Unit
BTDC	: Before Top Dead Center
ATDC	: After Top Dead Center

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d	: Diameter
P	: Pressure
$\mathbf{Q_n}$	: Heat Release
$\mathbf{m_{fi}}$	: Fuel Injection Amount
θ	: Angle
t	: Time
L	: Length
ppm	: Parts Per Million
Φ	: Equivalence Ratio

IMPACT OF DIFFERENT NOZZLE SPECIFICATIONS ON ENGINE EMISSIONS

SUMMARY

Advancing emissions requirements and the customer demand for increased performance and fuel efficiency are forcing the diesel engine technology to keep improving 100 years after the first commercial application. It is obvious that with each new emissions requirement, there is significant change in the limits. Considering how low the emissions are compared to just a few years ago, it becomes obvious that reaching these limits require innovative solutions improved calibration and controls.

Of all the subsystems of a diesel engine, maybe the injection system has undergone the most excessive development phase over the last 20 years. Earlier designs quickly left the field for more advanced and electronically controlled counterparts, opening a big gate enabling huge steps to be taken in the diesel engine technology. Not so much ago, diesel engines were heavy, noisy, and not as powerful as spark ignition engines. In addition, most of them used to emit the famous ‘diesel black smoke’ being one of the most harmful combustion byproducts to human health. Only their longer service life and fuel economy benefits make them in common use in heavy duty and commercial market vehicles. Nowadays, especially with the advance of common rail injection systems coupled with other high technology sub-systems like VGT turbochargers and many aftertreatment systems, diesel engines are as clean, powerful and silent as spark ignition engines, if not better.

High injection pressures increased the power density of the diesel engines and with the use of VGT turbochargers, low-end torque output and turbo lag are eliminated to a greater extent, making diesel vehicles fun to drive.

In this study, four different sets of injectors having different hydraulic flow rate, cone angle and number of holes were tested on an engine dynamometer at steady-state conditions. Engine out NOx – soot trade off curves were used to compare the injectors. Two separate test points were extracted from a real life emission cycle run on a Ford Transit vehicle. Those two points were deliberately selected to have different speed and load characteristics to enable testing of the injectors over a wide range of usage area. Afterwards, a six-variable design of experiments was derived over those two points. The variables were rail and boost pressure, EGR ratio, start of main injection, pre injection separation and amount.

Of all the tested injectors, baseline injector was seen to be the best option, although lower flow and higher hole number injectors normally should exhibit better trade-off curves. The reason behind this is that the combustion system does not perfectly match with the other injectors to enable them give their intended performance.

FARKLI ÖZELLİKLERE SAHİP ENJEKTÖR MEMELERİNİN MOTOR EMİSYONLARINA ETKİSİ

ÖZET

Son zamanlarda gittikçe sıkılaşan emisyon regülasyonları ve gittikçe daha fazla yakıt ekonomisi ve performans yönünde değişen müşteri istekleri, dizel motor teknolojisini piyasaya çıkışından 100 yıl sonra bile hala gelişmeye mahkum etmektedir. Her seferinde daha da azalan emisyon regülasyonlarını tutturmak için şüphesiz ki daha gelişmiş, yenilikçi çözümlerin piyasaya çıkarılması zorunludur.

Dizel motorlar için belki de bütün alt sistemler içinde yakıt enjeksiyon sistem son 20 yılda en çok gelişimin kaydedildiği sistem olmuştur. Eski mekanik dizaynlar, yerlerini yeni ve elektronik kontrollü sistemlere bırakıp dizel motorların son yıllarda kaydettiği büyük gelişimin kapısını aralamıştır. Kısa zaman öncesine kadar, dizel motorlar ağır, gürültülü, güçsüz ve siyah duman atan motorlar olarak görülürdü. Yalnızca uzun ömürlü ve daha ekonomik yakıt tüketim özellikleri sayesinde ağır iş ve ticari araçlarda kullanılırdı. Özellikle ortak hatlı yakıt enjeksiyon sistemi ve diğer sistemlerdeki gelişmeler sonucu (VGT turbolar ve çeşitli emisyon azaltıcı ek sistemler) günümüz dizel motorları en az benzinliler kadar çevre dostu, güçlü ve sessiz hale gelmiştir.

Yüksek yakıt basıncı dizel motorların üretebileceği maksimum gücü arttırırken VGT turbolar da özellikle düşük yük ve devir sayılarında görülen düşük maksimum tork üretimi ve turbo gecikmesi gibi problemleri büyük ölçüde elimine edip dizel motorlu taşıtları sürülmesi de bir o kadar zevkli hale getirmiştir.

Bu çalışmada, dört farklı özelliklere sahip enjektör setleri motor dinamometresi üzerinde sabit noktalarda test edilmiştir. Enjektörler, farklı debi, delik sayısı ve koni açılarına sahiptir. Enjektörleri karşılaştırmak için motor çıkış egzost gazının NOx ve duman karakteristikleri baz alınmıştır. Test noktaları Ford Transit model bir aracın koştuğu emisyon testi sonuçlarından seçilmiş, mümkün olduğunca geniş bir alanı kapsayabilmek için de seçilen nokta karakteristikleri birbirinden mümkün olduğunca farklı tutulmaya çalışılmıştır. Daha sonra seçilen bu iki test noktasının baz kalibrasyonu üzerinde 6 değişkenli iterasyonlar geliştirilmiştir. Bu altı değişken, ray ve emme manifoldu basıncı, EGR oranı, ana enjeksiyon başlama noktası, ön enjeksiyon ayrım uzunluğu ve miktarını kapsamaktadır.

Bütün test edilen enjektörler içinde ana enjektörler en başarılı sonuçları elde etmiştir. Daha düşük debili ve yüksek delik sayılı enjektörlerin daha iyi sonuçlar vermesi beklenirken yanma sisteminin bu enjektörlerin daha performaslı çalışması için yeterince uygun olmaması nedeniyle sonuçlar daha kötü çıkmıştır.

1. INTRODUCTION

Environmental pollution has ever become a concern for human health although significant actions to reduce pollution in every aspect have only recently been taken. After the industrial evolution, especially with the invention of the steam machine, the rate of using of natural resources increased very rapidly due to the competition between nations to leap in front of each other in the industrialization race. Of all the natural resources, oil and oil derivatives like diesel, gasoline or natural gas took the lead as main energy producer since they became the common source of energy for nearly all of the energy conversion machines humankind has created. Taking also faster than ever increase in population, demand in such machines also significantly increased over the years. Because of this, increased rate of pollution has started to threaten human life and one of the major sources of pollution is the vehicles using internal combustion engines as the primary source of energy conversion means.

Major byproducts of combustion in engines are NOx, HC, CO and soot, all of which are harmful for human health and stay in the atmosphere four tens of years. It is not thus a surprise that emission regulations have specific limits for especially those pollutants. After the invention of the internal combustion engines, only a hundred years was enough for pollution to reach dangerous levels. Terms like climatic changes and greenhouse effect all entered into daily life because of the mentioned air pollution problem. As a result of this, governments start to put mandatory regulations for vehicle exhaust gas emissions starting from 1980's. Within 25 years, now the regulation levels are nearly one tenth of what they were when first introduced, driving automotive industry to find new technologies to produce greener engines. Hybrid vehicles will be very common in near future and probably the next level will be fully electrically operated vehicles. However, at least until all the oil sources are depleted, internal combustion engines will still be in use.

Twenty years ago, diesel engines were more harmful to environment compared to spark ignition engines, mostly due to their high HC and particulate emissions. Nevertheless, big steps were taken afterwards and now the diesels are as clean as spark ignition engines. Those steps contain improvements in operation conditions as well as constructive advancements. In this study, one of the most important features of diesel engine, fuel injection system will be thoroughly investigated and effects of changes in constructive properties of fuel injectors will be tested. Tests will be conducted on a Ford DI diesel engine on engine dynamometer at Ford Otosan Gölcük Plant.

2. OVERVIEW OF FUEL INJECTION SYSTEMS

Stricter regulations on noise and especially exhaust gas emissions and further desire for more fuel consumption continually placed bigger demands on diesel fuel injection systems.

Mainly, fuel injection system is required to inject a precisely metered amount of fuel at a required pressure, depending on the equipment used, into the combustion chamber. With this, it has to be ensured that the fuel injected mixes effectively with the air inside the cylinder depending on the operating conditions of the engine. The power output and the speed of the diesel engine are controlled only by means of the injected fuel quantity inside the combustion chamber since the system does not include a throttle, as in gasoline engines, to control the air quantity. The most important criteria to control the diesel engine are:

- The timing and duration of the injected fuel
- The dispersal of this fuel within the combustion chamber
- The point at which the ignition is started
- The volume of the injected fuel relative to the engine speed and desired power requirement from the engine at that specific engine operating point

To realize the fuel injection, there are many parts and systems constituting the overall fuel injection phenomena.

In general, the low-pressure fuel supply system transfers the fuel from the fuel tank to the fuel injection pump at a specific pressure. However, nowadays, especially in light duty applications, fuel injection pump directly sucks fuel from the fuel tank without a necessity of using additional fuel supply systems. Then, the fuel injection pump generates the required fuel injection pressure. In most of the systems, fuel goes through a series of high pressure fuel lines during the transfer of the high pressure fuel from the fuel pump to fuel injectors, or as in the common rail systems, from fuel pump to common rail first and then from the common rail to injectors. The fuel injection pressure changes from system to system, but generally varies in the range of 200 to 2000 bar.

Engine power output, combustion noise and exhaust gas emission composition are mainly influenced by the mass of the injected fuel, timing, the rate of discharge and the overall combustion process itself.

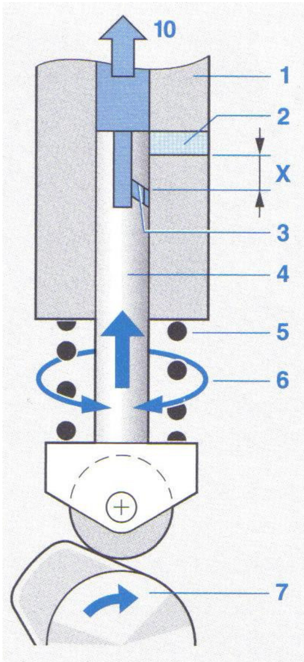
Until the introduction of electronic control systems into automotive engineering, injected fuel quantity and the start of injection were controlled by fully mechanical means. Nowadays, electronic engine control units manage the fuel injection process by involving many more parameters coming from the sensors such as geographical altitude, ambient and boost air temperature. Then, all the injection parameters are compensated accordingly, depending on the engine speed and load.

The main differences between fuel injection systems come from how the high fuel pressure is generated and how the duration and the start of injection are controlled. Below is a short summary of the most commonly used fuel injection systems in the recent diesel engine industry.

2.1 In-Line Fuel Injection Pump System

These fuel injection pumps have a separate pump element for each engine cylinder, arranged in-line. These elements consist of a barrel and a plunger moving the in the delivery direction by the camshaft. This camshaft is generally designed specific to the engine and its requirements, which is an integral element of the fuel pump driven mechanically by the engine itself. Plunger returns to its original position with the help of a spring.

The stroke of the plunger cannot be changed. Pressure generation starts when the top of the plunger closes off the inlet port on its delivery stroke. This point is called start of delivery. As the plunger continues to move upwards, pressure starts to build up and then the nozzle opens when the necessary pressure occurs, leading to fuel injection into the cylinder. When the helix on the plunger reaches the inlet port height, fuel finds a low pressure point to escape and the pressure at the nozzle entry starts to decrease. This marks the end of injection. Delivery quantity increases as the effective plunger stroke increases. Effective stroke can be adjusted by turning the plunger by a control rack depending on the engine load. This changes the position of the helix opening, leading to the change in the injected quantity.



1 Pump barrel
2 Inlet port
3 Helix
4 Pump plunger
5 Plunger spring
6 Adjustment range using control rack (injected-fuel quantity)
7 Camshaft

Figure 2.1 Standard in-line fuel injection pump [1]

2.2 Distributor Injection Pump Systems

These pumps have only one pump element serving to all cylinders. High pressure is generated by the axial piston or several radial pistons. Rotating distributor plunger opens and closes the ports, adjusting which cylinder will get the pressurized fuel.

2.2.1 Axial Piston Distributor Pump System

In this pump, there is a cam plate directly rotated by the engine. This cam plate rotates on the roller ring, thus also receiving a reciprocating motion. With the appropriately machined cam lobes on the bottom of the cam plate, lifting motion generates the pressure needed. The number of the lobes depends on how many cylinders the engine has.

In the port controlled axial piston distributor pump as shown on Figure 1.2, effective stroke is controlled by the control collar, which is actuated by a mechanical governor or an electronically controlled actuator mechanism.

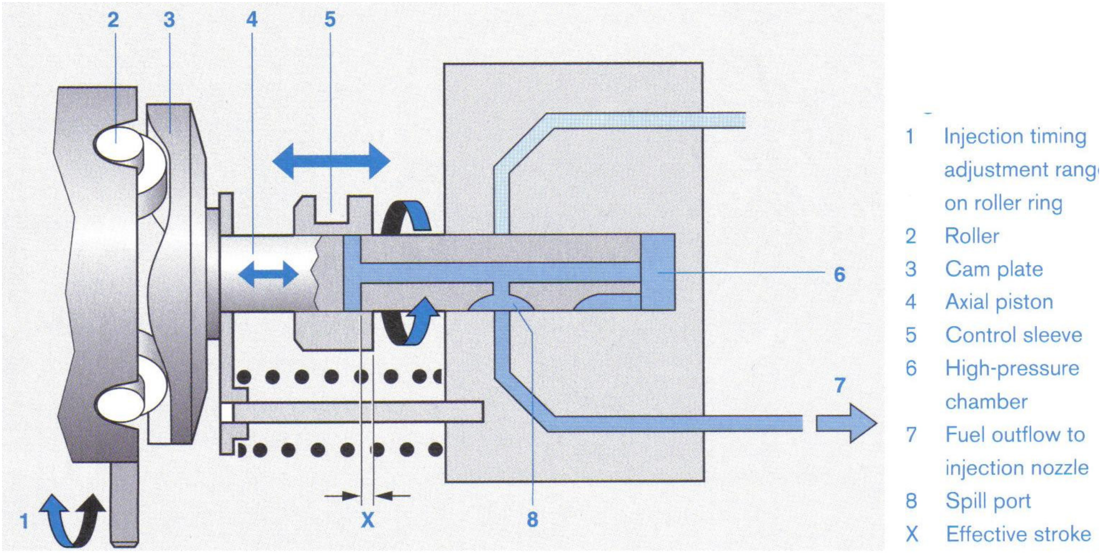


Figure 2.2 Port controlled axial piston distributor pump [1]

2.2.2 Solenoid Valve Controlled Radial Piston Distributor Pump System

A radial piston pump with a cam ring on the outside generates high pressure. As the radial piston rotates inside the cam ring, due to the shape of the cam lobes, the radial motion of the pistons inside the high-pressure chamber pressurizes fuel. The cam ring can be rotated by the timing device, thus changing the start of the delivery. With this system, an electronically controlled high-pressure solenoid valve meters the injection quantity and controls the start of injection. When the valve is closed, fuel pressure can build up in the high-pressure chamber, but when it opens, fuel can escape through the low-pressure passage, leading to end of injection.

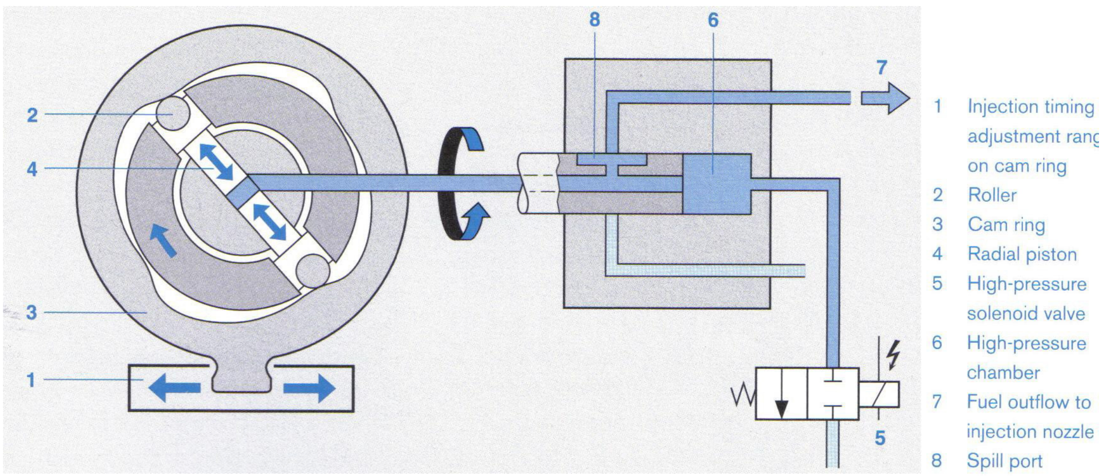


Figure 2.3 Solenoid valve controlled radial piston distributor pump [1]

2.3 Unit Injector System

In this system, fuel injection pump and nozzle are integrated into one single unit for each cylinder. The pump is actuated directly by a tappet or by a rocker arm driven by engine camshaft.

In this system, high pressure pipes are eliminated by integrating all the elements into one unit, thus reducing pressure drops during the fuel transfer. As a result, fuel injection is possible at higher pressures, reaching up to 2200 bar in some applications. This system is controlled electronically. A control unit actuates the solenoid valves to adjust start and duration of injection.

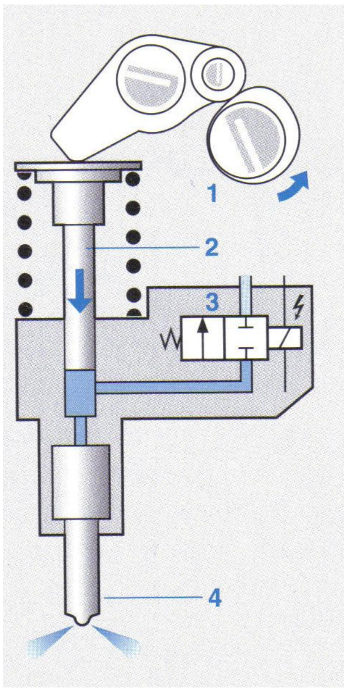


Figure 2.4 Unit injector system [1]

2.4 Common Rail System

In common rail systems, functions of pressure generation and fuel injection are separated. This is realized by adding a control volume to accumulate and pressurize the fuel. Injection pressure is independent of the engine speed and injected fuel quantity, thus giving high flexibility to the fuel injection regime.

In this system, the engine actuates high-pressure pump continuously, ensuring that the pressure inside the rail is kept at the desired value. Control unit calculates the start of injection and duration, and then actuates the injectors by applying appropriate current.

Solenoid valve injectors were used in previous common rail systems, but nowadays, piezo actuated systems start to take over due to its faster response and higher accuracy.

Fuel pressure is sampled by a sensor on the common rail and fed back to the control unit. In the earlier designs, there was a pressure control valve on one side of the common rail, regulating the pressure inside, but current designs employ an additional control valve on the low-pressure side of the system metering just the amount of fuel needed to be pumped into the rail to reach the desired pressure.

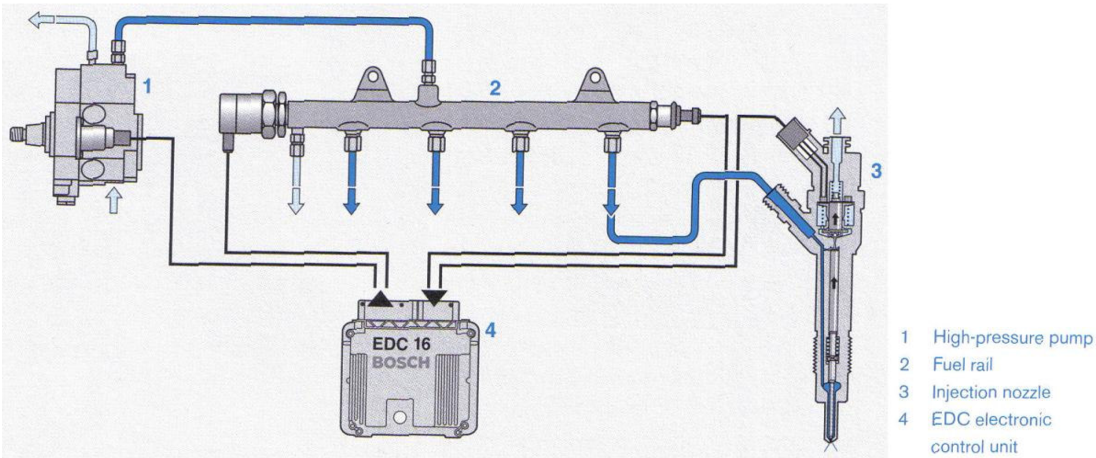


Figure 2.5 Common rail system [1]

3. DETAILS OF COMMON RAIL SYSTEMS AND THEIR COMPONENTS

The main advantage of common rail systems lies in its ability to change the injection pressure, start and duration of injection over a very broad scale and faster than any other system. This is largely due to the separation of pressure generation from the fuel injection event, by adding additional equipment known as pressure accumulator or common rail.

The main advantages of common rail system can be summarized as follows:

- High injection pressures are possible, up to 2000 bars nowadays
- Injection pressure is variable, depending on the engine state and condition from around 200 bar to up until 2000 bars in a very short time interval
- Start of injection can be varied in a very broad scale from injection to injection
- It is possible to make several injection events (up to 6 injections are possible in the current technology) per one combustion stroke

Due to its high flexibility and ability to reach very high injection pressures, these systems became the most commonly used system after its introduction in late 90's.

3.1 Working Principle

3.1.1 Pressure Generation

An accumulator volume, common rail, is used to separate pressure generation function from the injection function. A continuously operating fuel injection pump driven directly by the engine pumps the fuel into the common rail, generating the desired rail pressure. Pressure inside the rail is maintained irrespective of the engine speed or the injected fuel quantity since the system is fast enough to compensate them. Due to the reason that the injection pattern is almost uniform, high pressure pump can be designed much smaller and its drive torque requirements significantly decrease.

Most of the cases, high pressure pump is an electronically controlled radial piston pump.

3.1.2 Pressure Control

3.1.2.1. High Pressure Side Control

Especially on the earlier designs and still on some passenger car applications, a pressure control valve on the high-pressure side controls required rail pressure. Generally, this valve is mounted on the rail or in some applications on the fuel pump high-pressure side.

In this system, fuel that is not required for injection flows back to the low-pressure circuit to keep the rail pressure at the desired value. This type of control is very fast in reacting to sudden rail pressure change demands depending on the engine condition.

3.1.2.2. Suction (Low Pressure) Side Control

In this system, fuel is metered at the suction side to put just enough amount of fuel into the pistons to reach the desired rail pressure set point. The control unit calculates the amount of fuel to be put into the pistons and the metering device is actuated accordingly, generally by a PWM signal. In a fault situation, there is also a safety valve mounted on the rail. This is generally preset to a certain value, which is a bit higher than the maximum rail pressure capability of that specific system.

By putting exactly the right amount of fuel inside the pistons, the quantity of fuel under high pressure can be reduced greatly, leading to much less drive torque to rotate the pump. This advantage is actually returns back as improved fuel economy. In addition, by reducing the amount of high pressure fuel returned back, the temperature increase inside the fuel tank can also be kept lower.

In some applications, both systems are integrated into one for fastest response and accuracy.

3.1.3 Fuel Injection

Fuel is supplied to injectors by short high-pressure fuel pipes from the common rail. The engine control unit controls the switching of the valves integrated inside to injectors to open and close the injector nozzle. The fuel delivered in this system is controlled by the duration of the injector switching time at a given rail pressure, independent of the engine or pump speed. With fast switching mechanisms, either actuated by a solenoid or piezo system, it is possible to make up to six injection events per one combustion. This system brought many advantages in reducing exhaust emissions by making a pre injection for NOx reduction or post injection for PM reduction. In addition, combustion noise can be reduced significantly without a big penalty in fuel consumption. The closing action of the injector needle is hydraulically assisted for fast ending of injection.

3.2 High-Pressure Pump

The high-pressure pump is the interface between the low pressure and the high pressure stages. Its function is to make sure that there is always the demanded pressure inside the rail at all operating conditions. This pressure is generated constantly and independent of the injected fuel quantity. For this reason, fuel is not compressed during the injection event compared to the conventional fuel injection systems.

Generally, 2 or 3-plunger radial-piston pumps are used. Pump is driven by the engine via a coupling, gearwheel, chain or toothed belt depending on the application. As a result, there is a fixed engine-to-pump drive gear ratio.

On some of the passenger car systems, drive torque is just about 16 Nm, i.e. 1/9th of the drive torque required for a comparable distributor injection pump [2]. The power required to drive the high-pressure pump naturally increases as the injection quantity increases for any given injection pressure or increases as the pressure demand increases for any given engine speed.

Many of the applications use fuel-lubricated pumps, but in some heavy-duty applications oil lubricated pumps are still being used.

In radial piston pumps, drive shaft is mounted in a central bearing. The eccenter is fitted to the drive shaft and gives the plungers a reciprocating move to pressurize the fuel inside.

In the recent pump designs, fuel to be pressurized is metered by a solenoid control valve on the suction side, generally actuated by a PWM signal. This system not only drops the performance demand of the high-pressure pump, also reduces the maximum fuel temperature inside the system and the fuel tank.

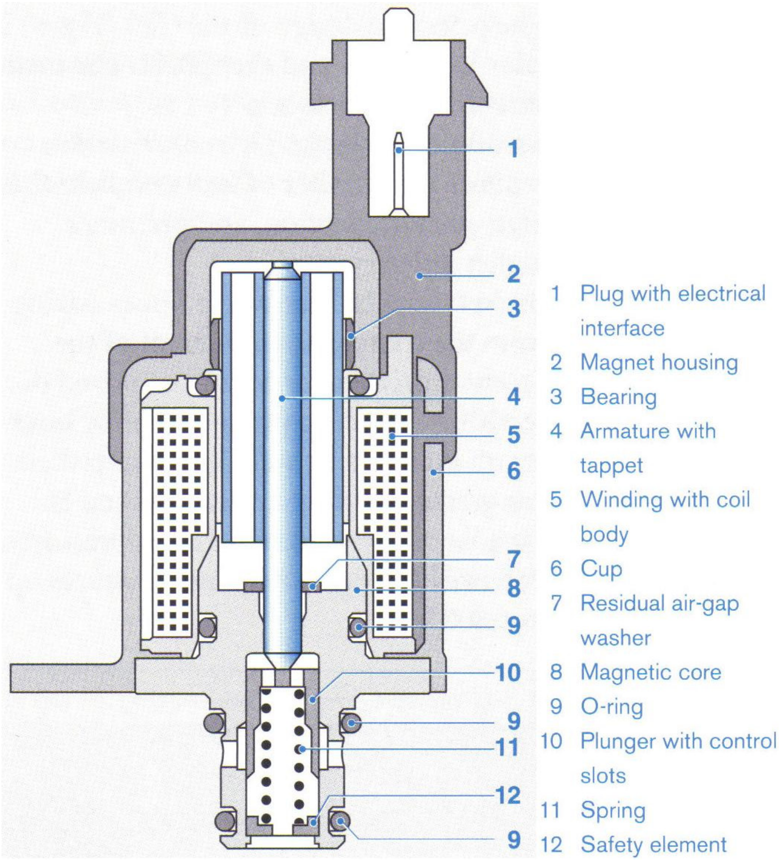


Figure 3.1 Fuel metering unit design [2]

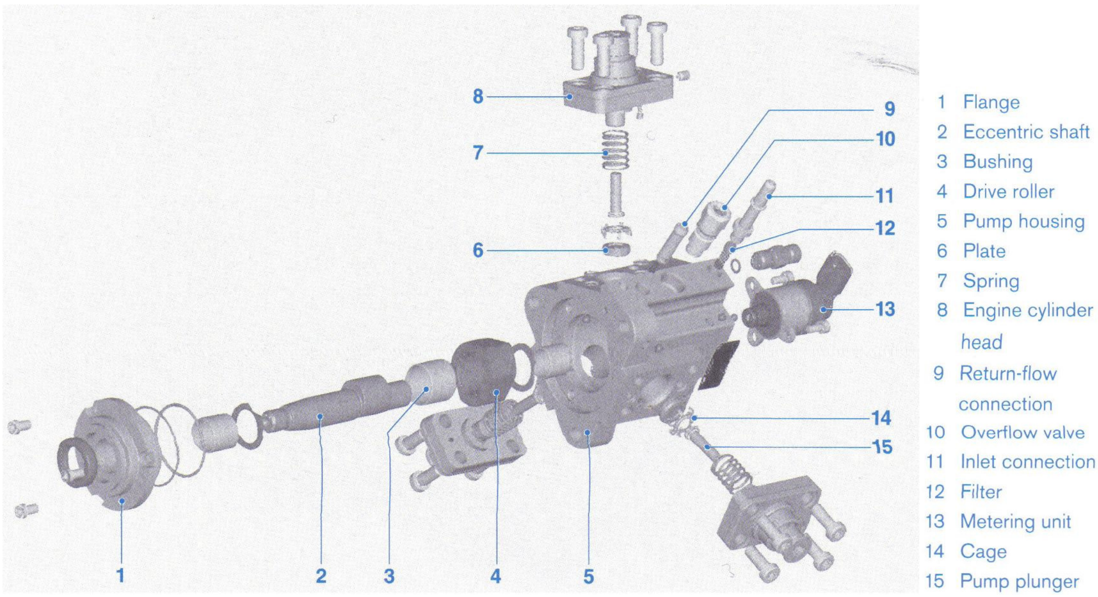


Figure 3.2 High pressure pump disassembled [2]

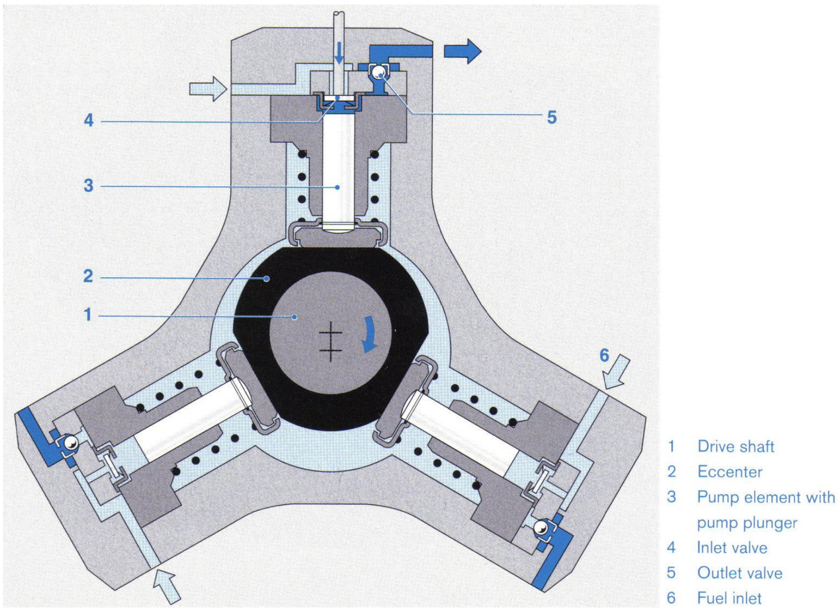


Figure 3.3 High pressure pump internal system [2]

3.3 High Pressure Accumulator (Fuel Rail)

The main purpose of the fuel rail is to contain fuel at the desired pressure. The accumulator volume should be large enough to ensure pressure inside remains almost constant, even during injection. It has to be designed stiff enough to withstand great forces due to the big fuel pressure inside. However, on the other hand, the material should be soft enough to dampen the pressure fluctuations due to pulsating nature of the fuel pump and mass loss due to injections. Also, the volume inside should not be more than necessary for fast enough pressure rise performance.

Fuel is transferred from the high-pressure pump to fuel rail by a high-pressure pipe. The pressurized fuel inside the rail is also transferred to the injectors via a series of short high-pressure pipes. Depending on the design, rail pressure regulator valve or pressure relief valve are also mounted on the rail. In addition, rail pressure has to be sampled fast enough to give feedback information to controller to calculate how much fuel should be metered and sent out to plungers of the high-pressure fuel pump to maintain the desired rail pressure. The compressibility of the fuel under high pressure is utilized to achieve the all important accumulator effect [2].

3.4 Injector

In a common rail system, injectors are sealed to the combustion chamber generally by copper gaskets, thickness varying from 1.4 mm up to 2.5 mm depending on the application. Injectors are fitted into the cylinder head by means of taper locks or special injector clamps fitted directly to cylinder head.

3.4.1 Solenoid Valve Injector

Operating principle is generally common whether it is a solenoid or a piezo type injector. Fuel is transferred to the injector via its connection to the high-pressure pipe from the common rail. There is an inlet orifice before the fuel goes into the control chamber. This control chamber is connected to the fuel return via another orifice, whose opening is controlled by the solenoid valve. The operating concept of these types of injectors is controlled by the balance of forces on the top and bottom of the nozzle needle and valve plunger assembly. When the engine is not running and there is no pressure inside of the rail, injector is closed by a spring.

When the engine is running, solenoid valve presses the valve ball onto the outlet orifice seat to ensure resting position of the injector. At this state, the valve control chamber and the chamber volume nearly have the same pressure, almost equal to the rail pressure. Nevertheless, the effective are on top of the valve control chamber is greater than the area on the chamber volume. Therefore, the force generated on top of the valve control chamber overcomes the force acting at the bottom, ensuring that injector stays at the closed position.

As soon as pickup current is applied to the solenoid valve by the ECU, the magnetic forces overcome the forces applied by the springs and the armature raises the valve ball from the valve seat. This opens the outlet orifice and fuel starts to flow to the cavity above and then to the return line up to the fuel tank. Inlet orifice prevents all the fuel flowing through the return line, thus, there is still almost the same pressure at the chamber volume, but greatly reduced rail pressure at the valve control chamber due to dynamic leakage. As a result, injector needle lifts up and injection begins. As long as the current is applied to the solenoid valve, injection continues. The rate of the needle movement depends on the inlet and outlet orifices and area differences between top and bottom of the needle. At any given rail pressure, injection amount is proportional to the current duration.

When the solenoid valve is no longer triggered by the engine control unit, the valve spring presses the armature down and the valve ball closes the outlet restrictor again. After this, the pressure at the valve control chamber starts to rise to the rail pressure and as soon as the force generated on the upper side of the needle overcomes the forces at the bottom, the needle starts to go down. The flow rate at the inlet orifice again determines the closing rate.

This indirect method of controlling the injection is required since the forces required to open the needle rapidly cannot be generated directly by a solenoid valve. Besides, there is always fuel leakage, which is necessary to lubricate the needle (static leakage). During the injection, the amount of leakage increases due to the indirect injection control method. This is called dynamic leakage.

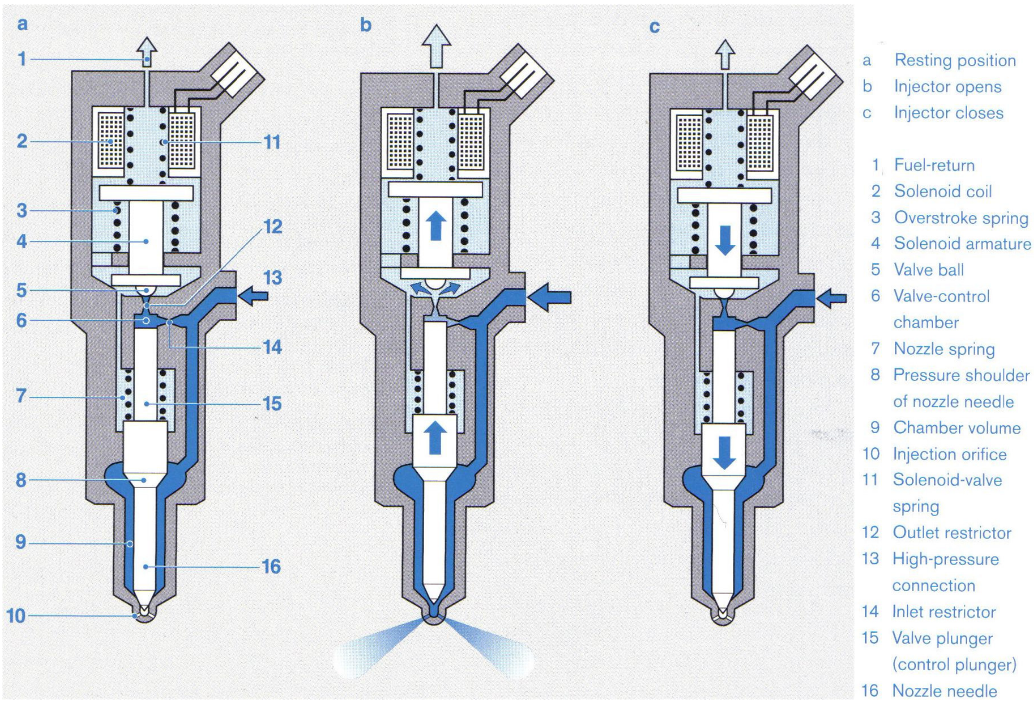


Figure 3.4 Solenoid valve injector working schematic [2]

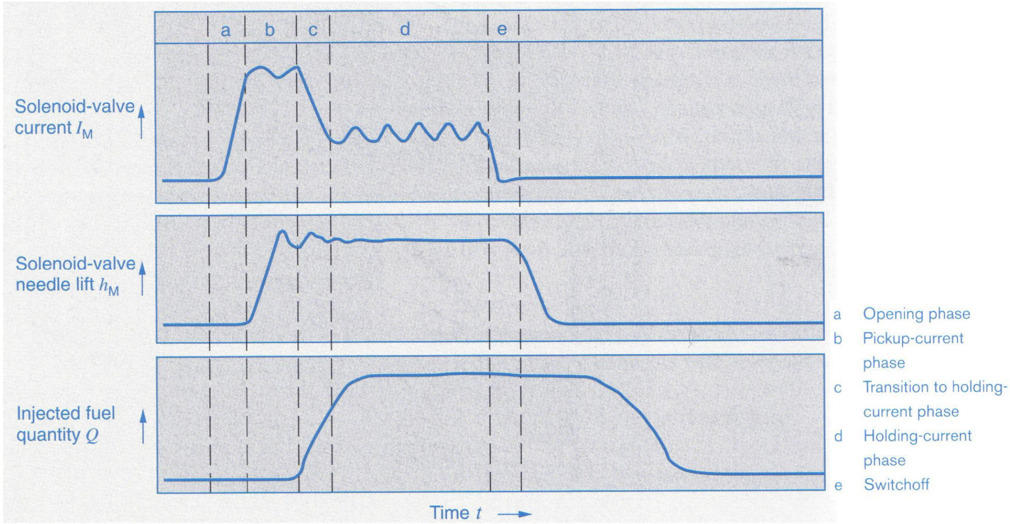


Figure 3.5 Triggering sequence of solenoid valve for single injection event [2]

3.4.2 Piezo Injector

In this design, moving masses and thus friction is reduced which leads to higher injector stability and less drift in time. In addition, this system allows very short intervals between injection events since the delay between electric start of triggering and hydraulic response of the nozzle needle is only about 150 microseconds [2].

In this system, a servo valve indirectly controls the nozzle needle. The valve triggering duration then controls the quantity of the injected fuel.

In the resting position, servo valve is closed and low pressure side is isolated from the high pressure side. The nozzle is kept closed by the rail pressure acting on the control chamber.

When the piezo actuator triggers the servo valve, bypass passage is closed and flow ratio between inlet and outlet orifices lowers the pressure in the control chamber. Due to dynamic leakage, nozzle starts to open and injection starts.

To close the injector, piezo actuator is discharged and servo valve releases the bypass passage. The control chamber then starts to be refilled by the reversing outlet restrictor by the fuel coming from the bypass passage. As soon as the required force is attained, nozzle needle starts to move down and injection ends.

Benefits of piezo injectors can be summarized as follows:

- More flexibility and lesser time intervals between each individual injection event

- Very small pre injection amounts possible (down to 0.5 mm^3/stroke)
- Smaller injector size
- Low noise and better fuel economy and exhaust emissions

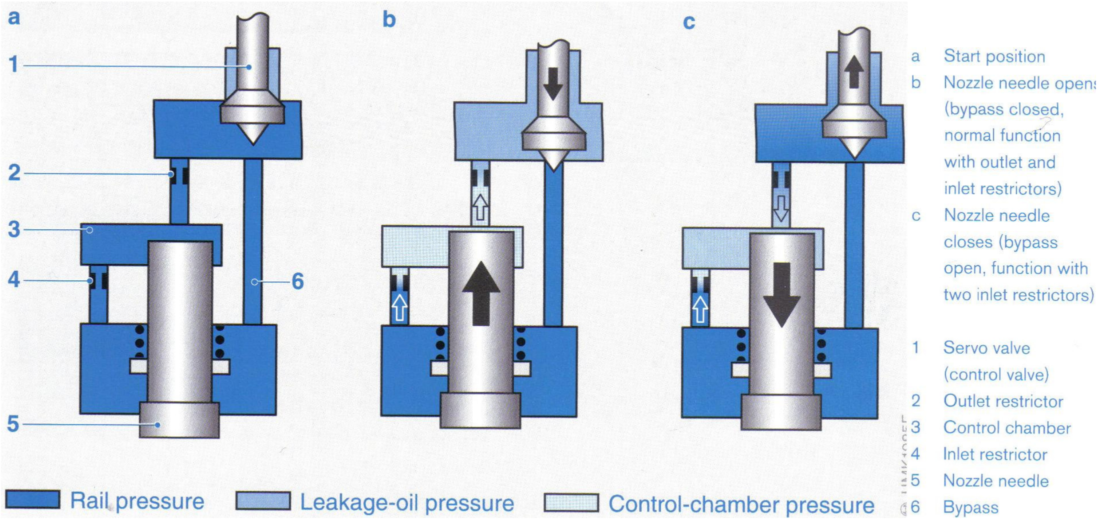


Figure 3.6 Function of the servo valve of piezo injector [2]

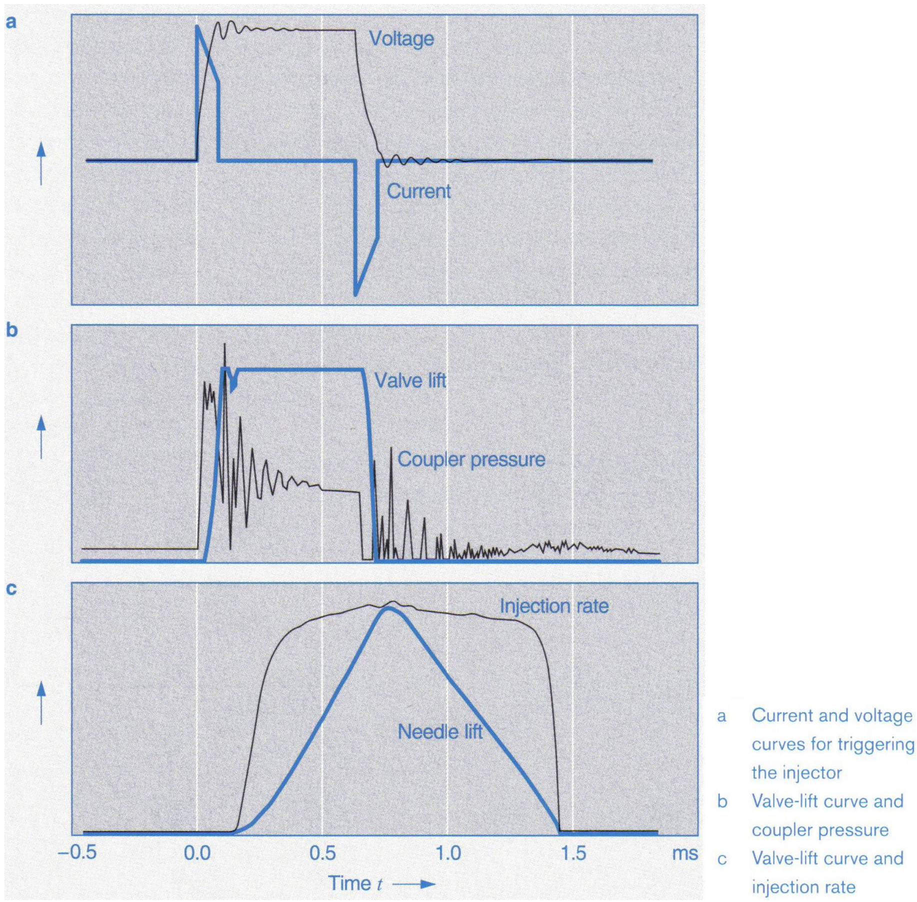


Figure 3.7 Triggering sequence of piezo injector for an injection event [2]

4. COMPRESSION IGNITION ENGINE COMBUSTION

Compression ignition or diesel engine combustion processes can be summarized as follows. First off all, fuel is injected into the cylinder towards the end of the compression stroke. Liquid fuel is injected at high velocity as one or more high velocity jets through small holes of the injector nozzles. Then, fuel atomizes into very tiny fuel droplets and after fast vaporization, mixes with high temperature and compressed air in the cylinder. Since the temperature and pressure inside the cylinder is well above the fuel ignition point, spontaneous auto-ignition occurs wherein places of already mixed fuel and air presents after some ignition delay. Cylinder pressure increases because of combustion and consequent compression of the unburned portion of the charge shortens the delay for the rest of the unburned fuel, which is mixed to the combustible limits. This second burning happens very rapidly.

Fuel injection continues until all of the desired injection amount entered into the cylinder. Above sequence of phases continue until all the fuel in the combustion chamber is burned, penetrating well into the expansion process.

Since injection starts just before the desired combustion point, there is no knocking as in the spark ignition engines. This enables using a higher compression ratio in diesel engines improving the fuel conversion efficiency.

In addition, due to combustion timing is controlled by the injection timing, the delay between the combustion and start of injection should be kept as short and stable as possible. Short delay is also needed to control maximum cylinder pressure within the mechanical design limits of the engine. Because of this, the auto-ignition characteristics of the fuel must be held within certain limits. This is controlled by cetane number, which is a universal measure of the ease of ignition of the diesel fuel.

Since the torque output of the engine is controlled by the amount of fuel injected with the airflow essentially unchanged, engine can be operated unthrottled leading to less pumping work and higher part load mechanical efficiency. Also the diesel engine operates only with lean mixture, leading to higher fuel conversion efficiency then spark ignition engines at a given expansion ratio.

4.1 Combustion in Direct Injection Systems

Figure 3.1 shows a heat release curve extracted from a single injection pattern.

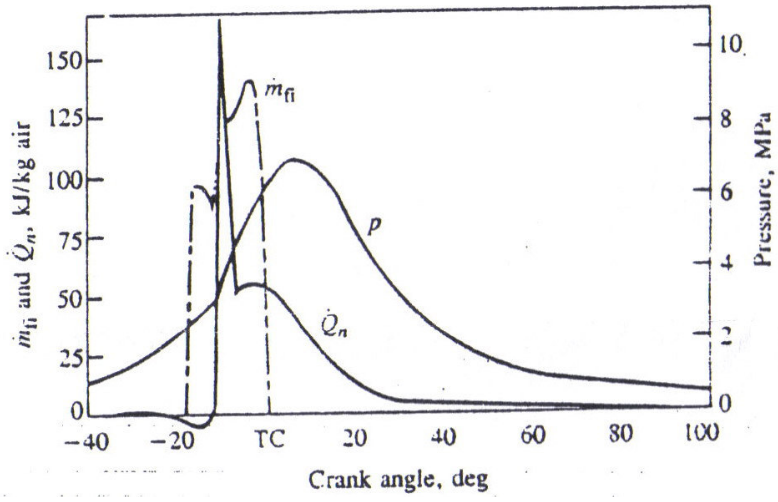


Figure 4.1 Cylinder pressure, heat release rate and rate of injection for a single injection [3]

During the combustion, burning proceeds in three distinguishable phases. In the first phase, cylinder pressure rises very rapidly since the initial rate of burning is very high. The second phase corresponds to a period of gradually decreasing heat release rate. This phase is actually the main heat release period and typically lasts about 30-40 crank degrees. Normally around 80% of the total heat release occurs during these two initial phases [3]. The final phase is the tail of the heat release curve in which a small rate of heat release remains through much of the expansion stroke.

Through the studies of combustion phenomenon, rate of injection and heat release, following model was derived for the overall combustion event over one cycle, which can clearly be identified on a typical heat release diagram.

Ignition Delay: This period is depicted between a-b on Figure 3.2. It is the period between the start of injection and the start of combustion. This is determined in the change of slope in the P-θ diagram.

Premixed Combustion: This phase is shown between b-c. In this phase, the combustion of the fuel, which has already mixed with air within the flammability limits during the ignition delay occurs. This combustion happens very rapidly and results the rapid build up of the cylinder pressure. Very high heat release rates within a couple of crank angles are the characteristic of this phase.

Mixing Controlled Combustion: This is between c-d in the diagram. Once the premixed combustion happens, the burning rate is then controlled by the rate at which the flammable mixture becomes available. The primary factor controlling the burning is the fuel vapor and air mixing process. The heat release rate decreases as this phase progresses and may or may not reach a second peak.

Late Combustion: This period is between d-e. Heat release in this phase continues at a lower rate but well into the expansion stroke. The main reason is some fraction of the fuel energy is still present in soot, which can be released.

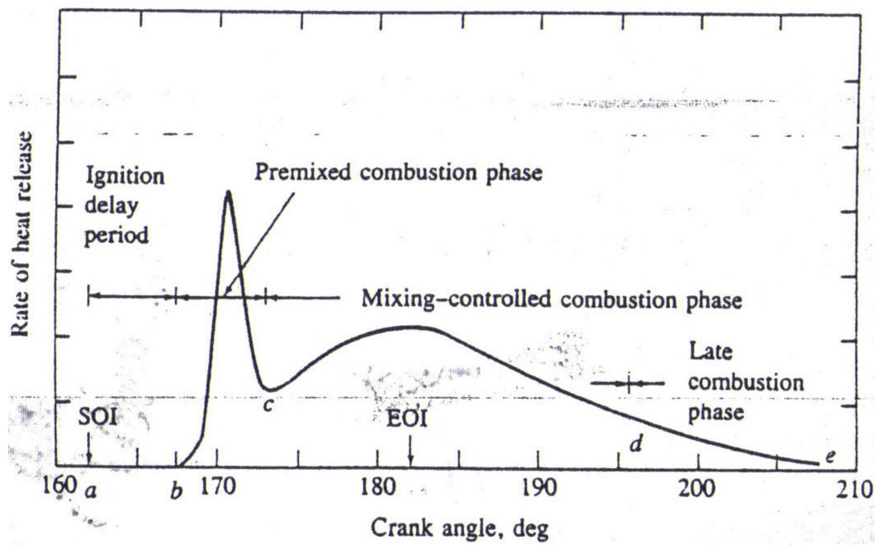


Figure 4.2 Heat release rate diagram [3]

4.2 Fuel Spray Characteristics

4.2.1 Spray Structure

Figure 3.3 is a typical structure of a DI diesel engine fuel spray. As the jet leaves the nozzle and enters into the combustion chamber, it suddenly becomes turbulent and spreads out wide. The initial jet velocity can be up to 100 m/s. The outer surface of the jet breaks up into drops of the order of 10μm in diameter. The breakup length is the portion of the jet where the liquid column disintegrates over some finite length into drops of very different sizes. Moving away from the nozzle, the amount of air within the spray structure increases and the spray starts to diverge increasing its width but the velocity decreases. As this air entrainment process continues fuel drops start to evaporate.

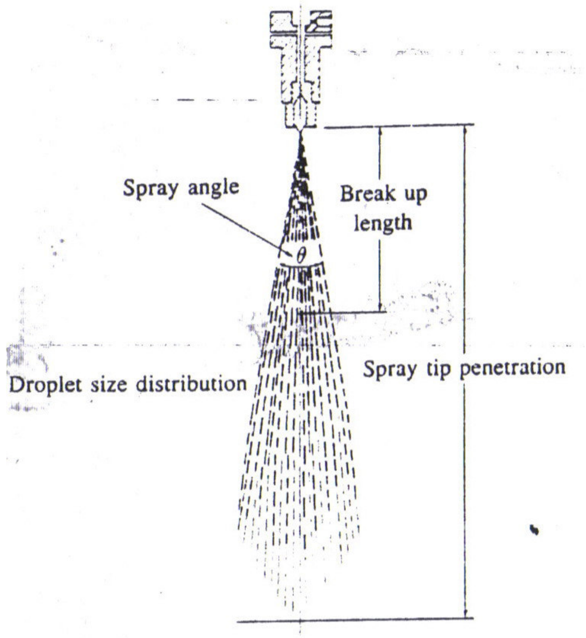


Figure 4.3 Diesel fuel spray schematic [3]

Most of the current high-speed diesel engines use air swirl inside the combustion chamber to increase fuel and air mixing rates. When the jet is injected radially outward into the swirling air, it becomes increasingly bent towards the swirling direction as it slows down and penetrates more. In this case, there occurs a large vapor-containing zone downstream of the liquid containing core.

4.2.2 Atomization

Fuel jet generally assumes a cone-shaped spray at the nozzle exit. This type of behavior is called atomization breakup regime and produces droplets with sizes very much less than the nozzle hole.

At low jet velocities, the behavior is defined by Rayleigh regime, which is due to unstable growth of surface waves caused by surface tension resulting in larger diameter droplets than the jet diameter. As jet velocity is increased, forces due to relative motion of the jet with respect to the air overcome the surface tension forces and results in droplet sizes about the jet diameter. This is called first wind-induced breakup regime. A further increase in the jet velocity results in breakup characterized by divergence downstream of the nozzle. Unstable growth of waves triggered by the relative motion produces droplets with sizes less than the jet diameter. This regime is called second wind-induced breakup regime. Further increase in jet velocity makes the outer surface breakup happens at the nozzle exit plane and results in droplets with average size much smaller than the nozzle diameter. Aerodynamic interactions at the air/fuel interface are the main mechanism in this regime.

Through controlled experiments and optical observations of the atomization regime, following behaviors can be summarized:

- Jet divergence angles increase with decreasing fuel viscosity
- Jet divergence angles decrease with increasing the nozzle hole length
- For the same length, rounded inlet nozzles produce less divergent jets than sharp edged nozzles.

4.2.3 Spray Penetration

The penetration length is one of the most influential factors for fuel air mixing rates. In many of the current high speed DI diesel engines, over-penetration gives impingement of liquid fuel on cool surfaces increasing emissions of unburned and partially burned species. On the other hand, under-penetration results in poor air utilization.

Data taken by Hiroyasu is shown on Figure 3.4. These data show that the initial tip penetration increases linearly with time and following the jet breakup, increase continues at $\sqrt{t}$. Injection pressure has a more significant effect on initial motion but ambient gas density start to become more of a factor on the motion after breakup.

In addition, the breakup length depends on nozzle geometry. Under high injection pressure and short nozzle hole length/diameter ratio, breakup can occur at the nozzle exit plane.

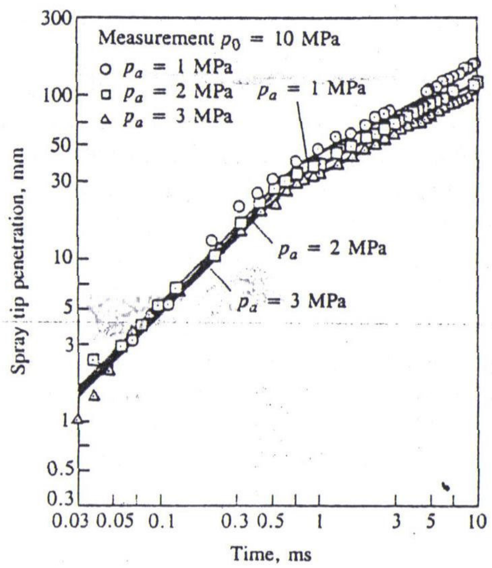


Figure 4.4 Spray tip penetration as a function of time for different ambient and injection pressures [3]

4.2.4 Size Distribution

A commonly used expression to characterize the mean diameter of a spray is Sauter Mean Diameter. This diameter represents the diameter of the droplet that has the same surface/volume ratio of the total fuel spray.

Effects of injection pressure, nozzle geometry and size, air and fuel conditions on Sauter Mean Diameter have been extensively studied over the years through various immersion, photographic and optical techniques.

Figure 3.5 show that nozzle size has significant effect on mean diameter. Nozzle L/d ratio is also seem to be important as ratio of 4 gives the minimum mean droplet size at low and intermediate injection pressures. Fuel viscosity and surface tension also was shown to be important especially at low injection pressures.

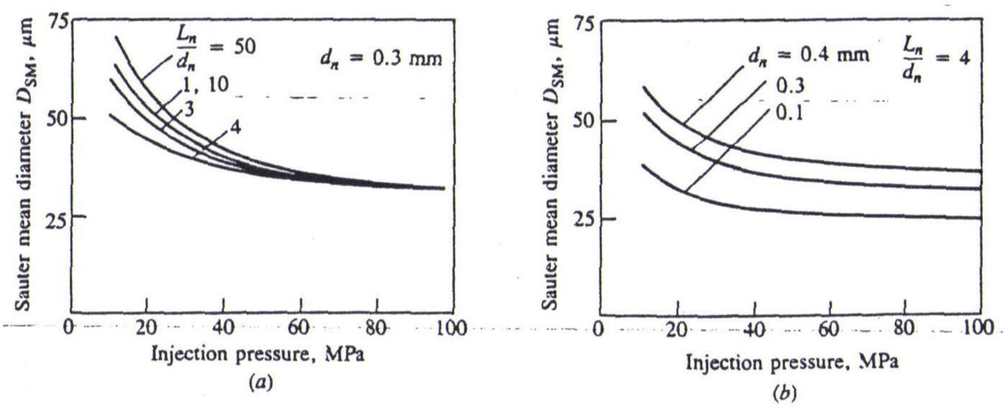


Figure 4.5 Effects of L/D ratio and nozzle hole diameter on SMD [3]

4.2.5 Spray Evaporation

Atomized liquid fuel injected into the cylinder has to evaporate to mix with air to become ready to burn.

The evaporation phenomenon actually depends on following sequence:

- Deceleration of the drop due to aerodynamic drag
- Heat transfer to the drop from the air
- Mass transfer of vaporized fuel away from the drop

As the droplet temperature increases due to heat transfer, because of increased fuel vapor pressure, evaporation rate also increases. As the mass transfer rate of vapor away from the droplet increases, the fraction of the heat transferred to the drop surface also increases. Due to velocity decrease of the fuel drop, the convective heat transfer coefficient between the air and the drop decreases. All these physical factors affecting the evaporation rate are summarized on Figure 3.6.

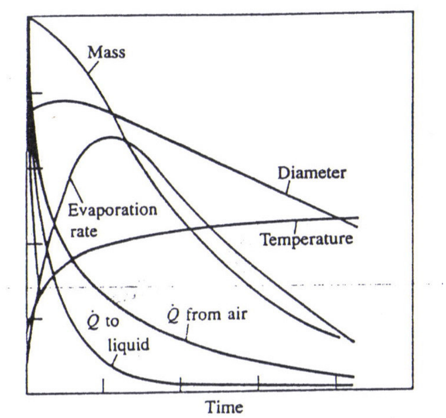


Figure 4.6 Change of physical properties with time during evaporation [3]

Above analysis also holds for the drops that are widely separated. In the spray core, the evaporation has a significant effect on the temperature and fuel vapor concentration in the air within the spray. As fuel vaporizes, the local air temperature decreases and the local fuel vapor pressure starts to increase. Finally, a thermodynamic equilibrium is reached.

4.3 Ignition Delay

Ignition delay is defined as the time between start of injection and start of combustion. Although determining start of injection is relatively easy by monitoring injector needle movement, it is harder to know when exactly the combustion starts. It is best identified from the change of slope of the heat release rate curve, derived from the cylinder pressure.

Ignition characteristics of the diesel fuel directly affect the combustion process. The ignition quality of the diesel fuel is defined by the cetane number. For low cetane number diesel, there is longer ignition delay resulting in most of the fuel being injected until the combustion starts. This results in very rapid burning rates with high rates of pressure rise and higher peak pressures. For higher cetane number diesel fuels, ignition delay is shorter meaning that ignition starts before most of the fuel is injected. As a result, heat release and pressure rise rates are smaller and more controllable. Combustion is mainly controlled by the rate of injection and fuel-air mixing, leading to smoother engine operation.

4.3.1 Factors Effecting Ignition Delay

Injection Timing: As the air temperature and pressure change significantly close to top dead centre, change in ignition delay is big with advanced or retarded injection. If injection commences early, initial temperature and pressure are lower so that ignition delay will increase until the combustible mixture forms. With the same process, if the injection starts later, initial pressure and temperature will become higher resulting in less ignition delay, but once the top dead centre is passed, temperature and pressure also start to decrease, meaning longer ignition delays again. An optimization should be found.

Intake Air Temperature and Pressure: Studies have shown that until around 1000 K cylinder air temperature, charge air temperature increase leads to a decrease in ignition delay. In parallel, increasing the charge air pressure also decreases the ignition delay but this effect starts to diminish as the charge temperature increases. As a result, higher compression will decrease ignition delay.

5. POLLUTANT FORMATION AND CONTROL IN COMPRESSION IGNITION ENGINES

When the air-fuel mixture is burned inside the cylinder, main combustion byproducts consist of NOx, CO, HC and soot. The amount of these pollutants in the untreated exhaust gas mainly depends on the engine operating conditions. Beside combustion chamber design and airflow path, fuel injection system also plays a very crucial role in minimizing emissions.

With the introduction of more and more stringent emission standards in the world, combustion process itself and its treatment obtain ever growing importance. To have the best possible trade off between the conflicting factors, pre and main injection must occur at precisely the right time and with the right quantity. With the latest developments in the electronic engine control systems, better control of the engine operating parameters is now possible leading to optimized combustion, reduced fuel consumption and lower pollutant emissions.

With the introduction of the Euro-5 emission regulations in Europe, it will no longer be possible to meet the emission requirements by only the internal modifications of the engine alone. There definitely will be necessities for additional exhaust gas after-treatment methods. With Euro-5, it will be necessary to fit particulate filter to comply with the very low particulate emission limits.

Following factors have big influences on the combustion process:

- Temperature and pressure inside the combustion chamber
- Composition, mass and movement of the charge
- Injection pressure process

These parameters are controlled by either engine specific fixed parameters or operating point specific variable parameters. Following properties can be said as fixed parameters:

- Compression ratio

- Stroke/bore ratio
- Shape of piston recess
- Intake port geometry
- Intake and exhaust valve timing
- Injector nozzle geometry

In addition, following factors are some of the operating point specific ones:

- Injection pressure
- Injection rate
- Injection strategy
- Exhaust gas recirculation rate
- Boost pressure

Regarding the air induction, mixture formation is mostly influenced by the movement of the charge inside the cylinder. This depends on intake duct geometry and combustion chamber shape. With the ever-increasing injection pressures, mixture formation process gradually shifted to fuel injection systems leading to development of low whirl combustion process.

On the fuel injection side, extremely small nozzle holes with flow-optimized geometries enhance good mixture formation. At the same time, this effect also shortens the ignition lag during which only small quantities of fuel is injected. During the diffusion combustion that follows, optimized atomization leads to high EGR compatibility, resulting in less NOx and soot formation.

Beside the fuel injection system, airflow side is also extremely important with ever more stringent NOx emission limits require very high EGR compatibility of the combustion process. This minimizes the formation of NOx so that particulate filter can then cope with the remaining particulates in the exhaust gas. Most important measures on the air side to minimize NOx emissions is exhaust gas recirculation. In this system, basically, exhaust gas is recirculated to the intake manifold, raising the proportion of the inert gas causing a drop in peak combustion temperature.

5.1 NOx Formation

Injection of fuel into the combustion chamber takes place just before the combustion starts and that non-uniform burned gas temperature and composition result from non-uniform fuel distribution during combustion. During the premixed combustion phase, just after the ignition lag, air-fuel mixture about stoichiometric ratio burns due to spontaneous ignition and flame propagation. During the mixing controlled combustion, mixture is closer to stoichiometric. However, throughout the combustion process, mixing between already burned gases, air and lean and rich unburned air-fuel mixture occurs, changing the composition of any gas elements that burned at a particular equivalence ratio. In addition, temperature changes due to compression and expansion occur as the cylinder pressure increases and decreases.

The critical time is when burned gas temperatures are at maximum, occurring between the start of combustion and shortly after the occurrence of the peak cylinder pressure. Mixture burning early in the combustion is important since it is compressed to a higher temperature, increasing the NO formation rate as cylinder pressure increases. After the peak pressure, burned gas temperature start to decrease as cylinder gases expand. Decreasing temperature due to expansion and mixing of high temperature gas with air freezes the NO chemistry. As a result, this freezing occurs in the diesel engine more rapidly than the spark ignition engine resulting in much less decomposition of the NO.

When the injection timing and amount are varied, all of the NO forms within the 20° crank angle after the start of combustion. As injection timing is retarded, combustion process is also retarded, resulting in late NO formation and concentration becomes lower since peak temperatures are now lower. At high load with higher peak pressures, temperatures and larger regions closer to stoichiometric, NO levels increase. However, the amount of the fuel injected decreases proportionally as the overall equivalence ratio decreases, much of the fuel still burns close to stoichiometric. Thus, NO emissions are roughly proportional to the mass of the fuel injected.

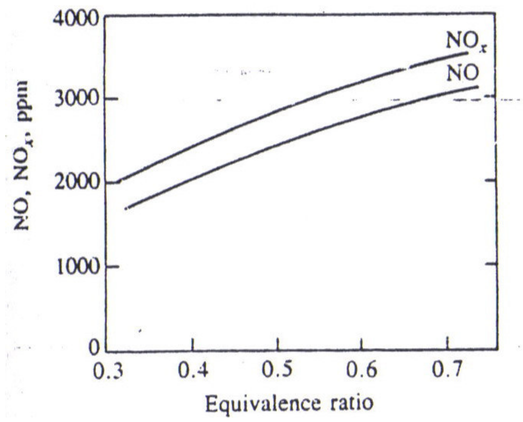


Figure 5.1 NO and NOx concentration as a function of equivalence ratio [3]

Added diluents to the intake air such as exhaust gas are effective in reducing NO formation rate. Figure 4.2 shows the effect of dilution of intake air with N_2, CO_2 and exhaust gas on NOx exhaust levels. This data show that the effect is primarily one of the reduced burned gas temperatures. The composition of the exhaust gas of a diesel engine varies with load. At idle, there is little CO_2 and H_2O and the composition is not so much different from the air. At high load, the heat capacity increases as the concentrations of CO_2 and H_2O are higher.

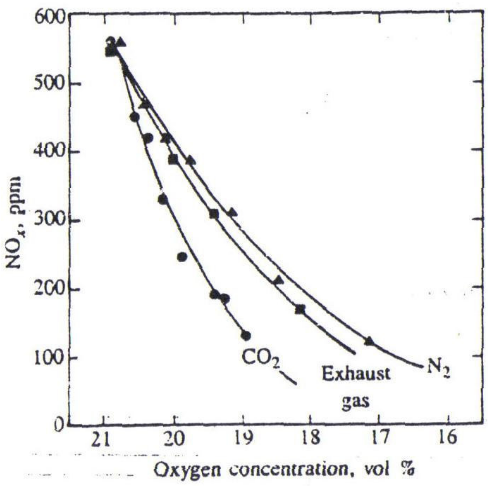


Figure 5.2 Effects of reduction in oxygen by different diluents on NOx emission [3]

5.2 CO Formation

Carbon monoxide is an odorless and tasteless gas. In humans, it inhibits the ability of blood to absorb oxygen, leading to asphyxiation. It is considered that CO concentrations of above 40% in blood carry a life risk.

Carbon monoxide results from incomplete combustion in rich air-fuel mixtures due to air deficiency. Although diesel engines always operate in lean mixtures, CO emissions still occur due to brief periods of rich operation or inconsistencies within the air-fuel mixture, yet in much less amounts than in spark ignition engines. Generally, fuel droplets that fail to vaporize form pockets of rich mixture areas that do not combust properly. Lower in-cylinder gas temperatures, lack of oxygen and less time left for CO to form CO_2 result in increase in CO emission. Especially in modern diesel engines capable of pumping high amounts of EGR, CO can be a major problem if EGR rate is more than enough especially when the engine is cold.

5.3 HC Formation

HC is a generic designation for the entire range of chemical compounds consisting of hydrogen H with carbons C. Hydrocarbons are the consequence of incomplete combustion of the hydrocarbon fuels. The combustion process also produces new hydrocarbon compounds that are not initially present in the original fuel. Again compared to the spark ignition engines, diesel engines emit much less HC emissions due to always-lean operation.

Due to the very complex nature of the diesel combustion, there are many ways that can contribute to diesel engine hydrocarbon emissions. Primarily, there are two ways by which fuel can escape normal combustion processes unburned. First, air-fuel mixture can become too lean to autoignite to propagate flame inside the combustion chamber, or second, during the premixed combustion process, the air-fuel mixture can become too rich to ignite to support a flame. This fuel can then be consumed only by slower thermal oxidation reactions occurring later in the expansion process after mixing with additional air.

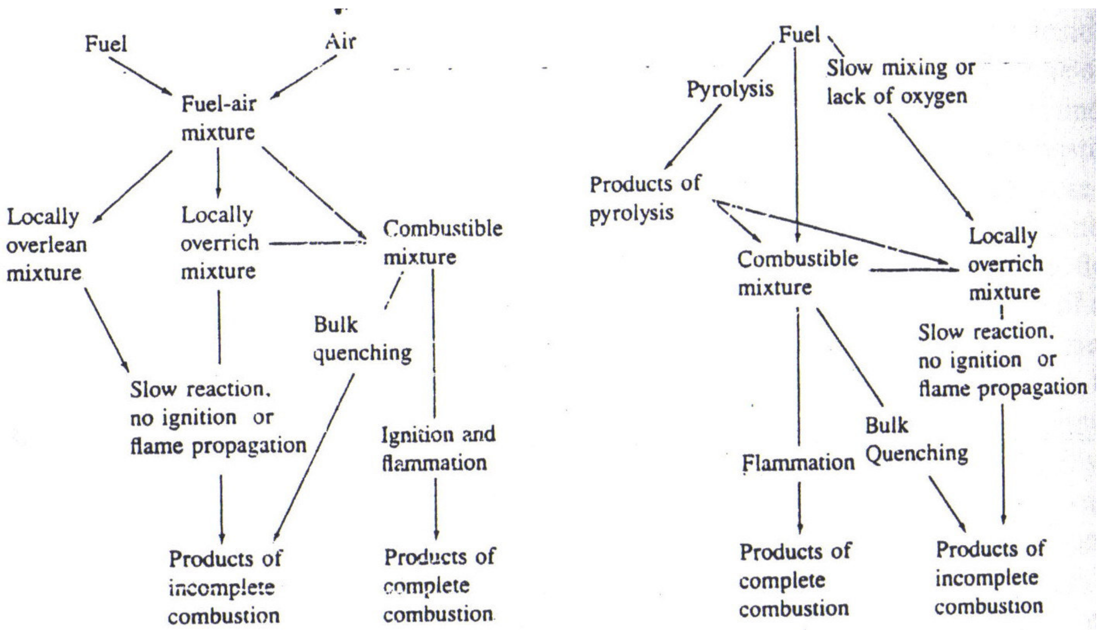


Figure 5.3 HC formation phases [3]

Hydrocarbon emission levels from diesel engines vary widely with operating conditions. Engine idling and light-load operation produce significantly higher HC emissions than full load operation. On the other hand, when the engine is overfuelled, HC emissions start to increase dramatically. Beside overleaning and undermixing, wall temperature also effect HC emissions, suggesting that cyclic variability in the combustion process can cause an increase in HC emissions due to partial burning and misfiring cycles.

5.3.1 Overleaning

As the fuel is injected into the cylinder, a distribution in the air-fuel equivalence ratio across the fuel sprays develops. Figure 4.3 shows this equivalence ratio distribution at the time of ignition. In a swirling flow, ignition occurs in the slightly lean of stoichiometric region downstream of the spray core where fuel, which has spent most of the time within the combustible limits is present.

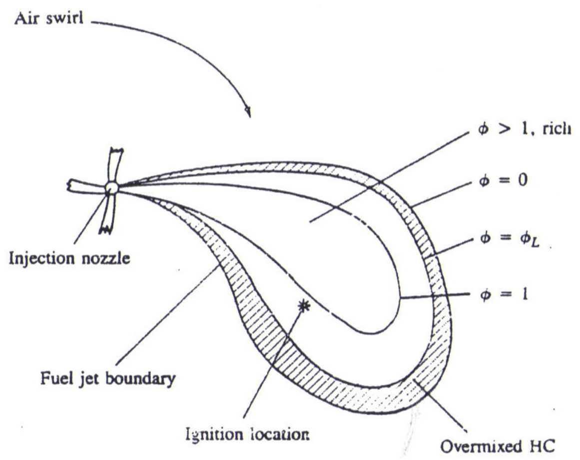


Figure 5.4 Diesel spray equivalence ratio schematic [3]

On the other hand, fuel close to the spray boundary has already mixed beyond the lean limit of combustion and will not autoignite or sustain a fast reaction front. This mixture can then only oxidize by relatively slow thermal oxidation reactions, which in the end will be incomplete. The magnitude of the unburned HC from these overlean regions will depend on the amount fuel injected during the ignition delay and the mixing rate during this period. As the delay period increases, HC emissions increase at an increasing rate. Under the conditions where ignition lag is long, overleaning of fuel is a major source of HC emissions.

5.3.2 Undermixing

Two sources can be singled out resulting in HC emissions during combustion due to slow or undermixing with air. One is fuel leaving the injector nozzle at low velocity during late in the combustion. This is mostly due to the nozzle sac volume. Second source is the excess fuel injection under overfuelling conditions.

At the end of fuel injection, nozzle sac volume (small volume left in the tip of the injector nozzle after the needle seats) is left filled with fuel. As the combustion proceeds, this leftover fuel is heated and starts to vaporize, entering the cylinder at very low velocity. This vapor will then mix very slowly with air and can easily escape the combustion process. Figure 4.5 shows HC emissions at the minimum ignition delay as a function of sac volume.

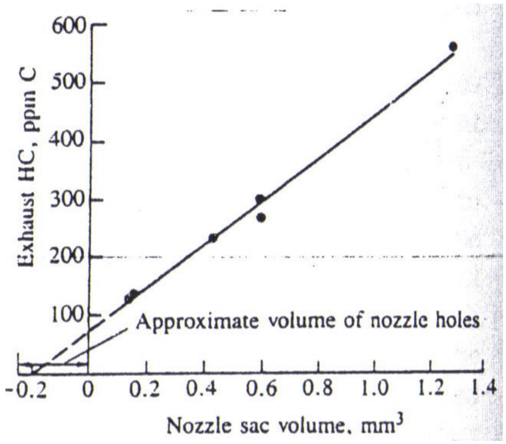


Figure 5.5 Effect of nozzle sac volume on HC emission [3]

Under transient conditions, especially as the engine goes through acceleration process, overfuelling can occur. Even though the overall equivalence ratio can remain lean, locally overrich conditions can exist through the expansion stroke and into the exhaust process. Figure 4.5 shows the effect of increasing the amount of fuel injected at constant speed while injection timing is adjusted to keep ignition delay at minimum value. HC emissions are unaffected by an increase in equivalence ratio until a critical value of about 0.9, after which a dramatically increase can be seen. This is not so significant under normal operating conditions, but can contribute to high amounts of HC emissions under acceleration conditions if overfuelling occurs.

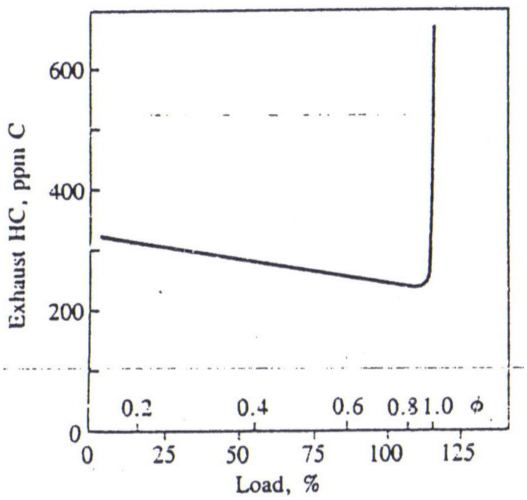


Figure 5.6 Effect of overfuelling on HC emission [3]

5.4 Particulate Formation

Diesel particulates consist of mainly combustion generated carbonaceous material, called soot, on which some organic compounds were absorbed. Particulate material results from incomplete combustion of fuel hydrocarbons, some of which is also contributed by the lubricant oil. The composition of the particulate material depends on the engine exhaust conditions. At temperatures above 500˚C, the individual particles are principally clusters of many spheres of carbon. The diameter range is about 15 to 30 nm. As temperature decreases below 500˚C, the particles become coated with adsorbed and condensed high molecular weight organic compounds, which include unburned hydrocarbons, oxygenated hydrocarbons and polynuclear aromatic hydrocarbons.

In diesel engines, the highest particulate concentrations are present in the core region of each fuel spray where local average equivalence ratios are very rich. Soon, concentrations rise rapidly after combustion starts. These very high local soot concentrations which decrease rapidly once fuel injection ends and rich core mixes to leaner equivalence ratios. Soot concentrations in the spray close to the piston bowl outer radius and at the cylinder wall rise later, but lesser in overall magnitude decaying more slowly. Away from the spray core, soot concentrations decrease rapidly with increasing distance from the centerline.

The soot formation process starts with a fuel molecule containing 12 to 22 carbon atoms and an H/C ratio of about 2, and ends up with particles a few hundred nanometers in diameter. Soot formation takes place in the diesel combustion environment at temperatures between about 1000 K and 2800 K at pressures of 50 atm to 100 atm. The time available for the formation of solid soot particles is around a few milliseconds.

First condensed phase material arises from the fuel molecules by oxidation. These products typically include various unsaturated hydrocarbons. The condensation reactions of gas phase species lead to the appearance of the first recognizable soot particles, often called nuclei. These first particles are very small, less than 2 nm in diameter.

Surface growth, by which the bulk of the solid-phase material is generated, involves the attachment of gas-phase species to the surface of particles and their incorporation into the particulate phase. Surface growth reactions lead to an increase in the amount of soot, but the number of particles remains unchanged. The opposite is true for growth by coagulation, where the particles collide and coalesce, which decreases number of particles with amount of soot is constant. By the time the surface growth stops, continued aggregation of particles into chains and clusters begin to occur.

The emitted soot is then subject to a further mass addition process as the exhaust gases cool and are diluted with air. Adsorption into the soot particle surface and condensation to form new particles of hydrocarbon species in the exhaust gases occurs in the exhaust system and continues after exhaust gases meet with the ambient air. Figure 4.7 shows the relationship between these processes.

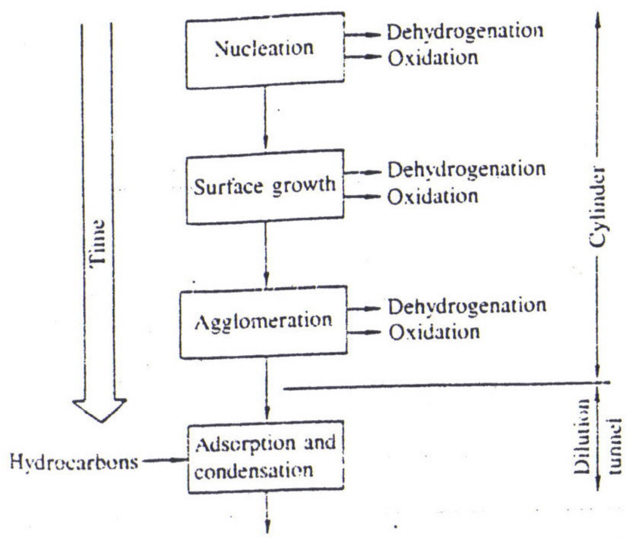


Figure 5.7 Diesel particulate formation processes [3]

5.5 Diesel Engine Emissions Control

5.5.1 Exhaust Gas Recirculation (EGR)

Exhaust gas recirculation (EGR) is a very effective way of internal engine measures to reduce NOx emission of an engine. There actually are two main types of EGR systems, namely internal and external EGR. In internal EGR systems, exhaust gas is kept inside the cylinder during the exhaust stroke, which is controlled by valve timing. In external EGR systems, exhaust gas is routed to the combustion chamber through additional external lines and a control system metering the amount. The NOx reduction effect of these systems is mainly due to the following causes:

- Decrease in rate of combustion and as a result, reduced local peak temperatures due to an increase in inert gas content in the combustion chamber.
- Reduction in partial oxygen pressure
- Reduction in exhaust gas mass flow

Since high local temperatures and high amount of partial oxygen pressure are required to form NOx, above measures make a drastic reduction in NOx emission as the amount of EGR increases. Reducing the reactive components in the combustion chamber also increase the black smoke, which limits the maximum rate of EGR inside the cylinder.

According to the operating concepts, external EGR systems can be grouped in two:

5.5.1.1. High Pressure EGR systems

In this system, exhaust gas is tapped upstream of the turbocharger turbine and routed through a mixer to the engine, upstream of the intake manifold. EGR rate strictly depends on the pressure difference between the pressure upstream of the turbine and the intake manifold pressure. A valve controls rate metering. On light duty or passenger car engines, throughout the emission region, this system is capable of flowing enough amount of EGR gas. Nevertheless, in the low load points, it is frequently necessary to restrict the gas flow on the intake side to maintain the pressure difference in a certain range to achieve sufficient amount of EGR gas inside the cylinder. This system is shown on Figure 4.8.

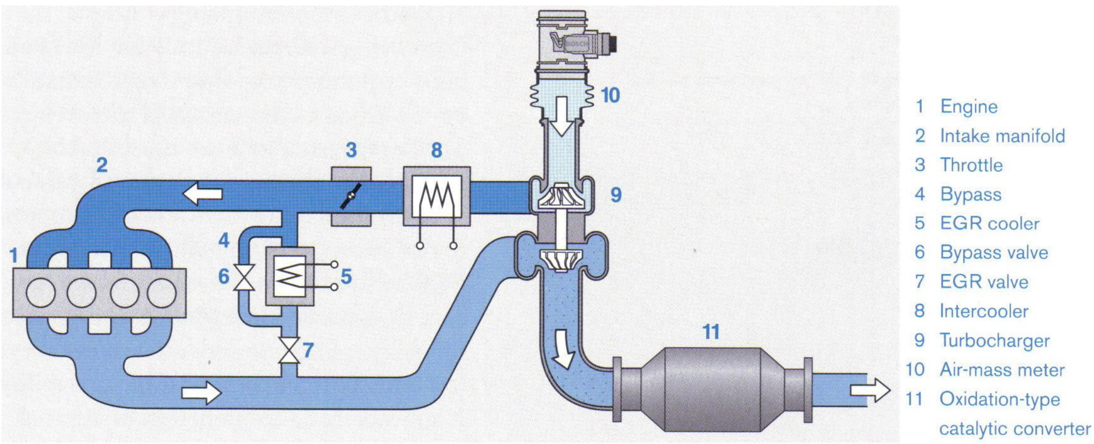


Figure 5.8 High pressure EGR system [1]

5.5.1.2. Low Pressure EGR systems

In low pressure EGR systems, exhaust gas is routed downstream of the turbine, tapped from the exhaust gas treatment system and injected on the air side upstream of the turbocharger compressor. Although this system is not so common nowadays, it is highly possible that in future, it will come into use more frequently due to below advantages:

- Uniform EGR distribution between the cylinders since exhaust gas is injected upstream of the compressor
- More cooling of the mixture after passing through the compressor and intercooler
- Increase and complete decoupling of possible charge-air pressure from the EGR rate since full exhaust gas mass flow is routed through the turbine

On the negative side, this system has larger volume contaminated with exhaust gas in operation. System schematic is shown on Figure 4.9.

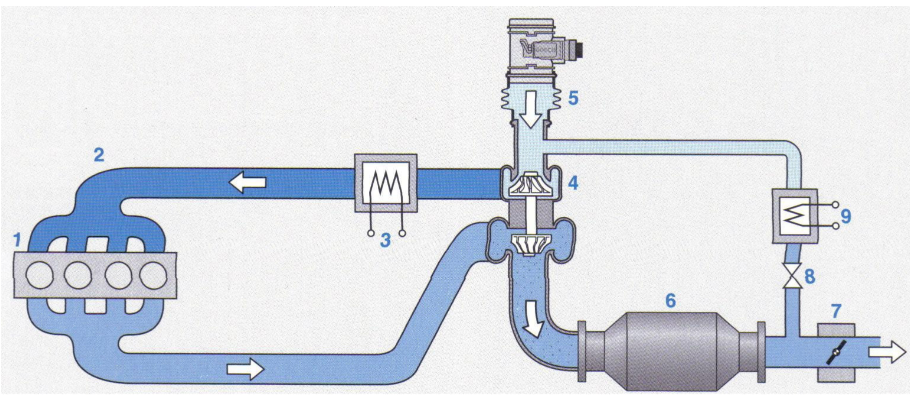


Figure 5.9 Low pressure EGR system [1]

In order to enhance the effects of EGR, recirculated exhaust gas is cooled in a heat exchanger by the engine coolant to raise gas density in the combustion chamber. Since diesel exhaust gas has already a low temperature at low load points, further cooling the EGR gas may lead to unstable combustion especially when the engine is cold. This then results in a significant rise in HC and CO emissions. To overcome this problem, a switchable EGR cooler is incorporated in some systems to increase combustion chamber pressure, stabilize combustion and reduce HC and CO emissions at such conditions. This also helps oxidation type catalytic converter to reach its operating temperature window much faster. Figure 4.10 shows how the EGR rate affects engine out HC, CO and NOx emissions as well as particulates. As can be seen, up until some certain EGR rate, NOx reduction is very significant without much penalty in HC, CO and particulate emissions. However, after this point, HC and CO increase dramatically, due to very inefficient combustion.

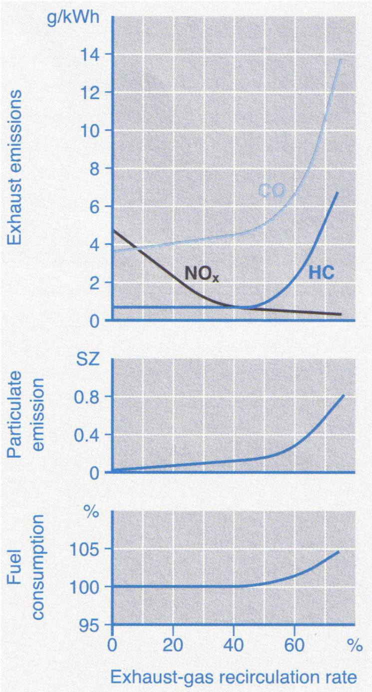


Figure 5.10 Influence of EGR on emissions and fuel consumption [1]

5.5.2 Diesel Oxidation Catalyst (DOC)

Apart from mainly oxidizing HC and CO in the engine exhaust gas, this aftertreatment device also reduces particle mass to some extent and can be used as catalytic burner.

Carbon monoxide (CO) and hydrocarbons (HC) are oxidized to form water vapor (H_2O) and carbon dioxide (CO_2). Oxidation in the DOC is almost complete, starting from a specific temperature, which is called light-off temperature. Depending on the exhaust gas composition, flow velocity and catalyst composition, light-off temperature varies between 170 ˚C to 220 ˚C. After reaching this temperature, catalyst conversion efficiency quickly rises up to over 90% within a short temperature range. Figure 4.11 shows the typical conversion efficiencies for HC and CO with respect to catalyst temperature.

The particles emitted by a diesel engine partly consist of hydrocarbons, which desorb from the particle core as temperature rises. Particle mass can be reduced up to 15% - 25% by oxidation in the DOC.

Catalytic burners can be used to raise the exhaust gas temperature during regeneration of the particle filter. The reason is the reaction heat is released when CO and HC are oxidized which increases the exhaust gas temperature. As an approximation for the heat released during oxidation, the temperature of the exhaust gas rises by about 90 ˚C for every 1% volume of CO [1].

Oxidation catalysts consist of a carrier structure made of ceramics or metal, an oxide mixture (washcoat) composed of aluminum oxide (Al_2O_3), cerium oxide (CeO_2), zirconium oxide (ZrO_2) and active catalytic noble metals, such as platinum (Pt), palladium (Pd) and rhodium (Rh) [1].

The main function of the washcoat is to provide larger surface for the noble metal and to slow down catalyst sintering occurring at high temperatures, leading to an irreversible drop in catalyst activity. The quantity of the noble metals used for the coating is referred to as catalyst loading which is specified in g/ft^3. This generally varies between 50 to 90 g/ft^3, depending on the application. Since only surface atoms are chemically active, it is aimed to keep the noble metals as small as possible in order to minimize their usage due to cost reduction purposes. Structural differences change major properties such as light-off temperature, conversion efficiency, temperature stability and manufacturing costs. For these reasons, DOC design should be perfectly matched to the application requirements to obtain best possible cost/performance ratio.

Beside the use of the correct catalyst, the main factors governing the efficiency of the exhaust gas treatment are the correct operating conditions. If the operating temperatures are excessively high, sintering process will occur (sintering happens when several noble metal particles clump together to form a larger particle with a smaller surface area, leading to reduced activity). For this reason, excessive exhaust gas temperatures should be avoided at any operating point of the engine.

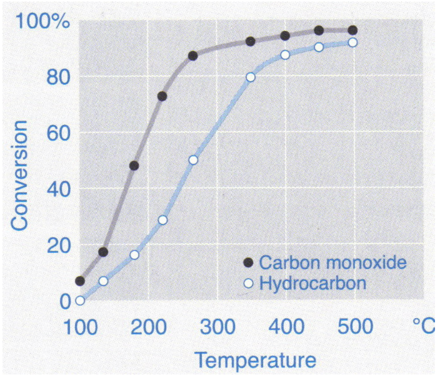


Figure 5.11 CO and HC conversion efficiency of catalytic converter as a function of temperature [1]

5.5.3 Selective Catalytic Reduction of Nitrogen Oxides (SCR)

Selective catalytic reduction (SCR) process operates continuously and does not intervene in engine operation. It offers the possibility of minimizing NOx emissions and reducing fuel consumption at the same time. It is based on reducing certain nitrogen oxides in the presence of oxygen using selected reducing agents. Selective means the reducing agent prefers to oxidize selectively with the oxygen contained in the nitrogen oxides instead of molecular oxygen, which is present in much larger quantities in the exhaust gas. Ammonia (NH_3) has proven to be a highly selective reducing agent, but presence of this substance can raise safety issues since it is toxic. However, ammonia can be produced from non-toxic carrier substances such as urea. Urea is also highly soluble in water and can easily be metered and injected to the exhaust gas as water-urea mixture. At a mass concentration of 32.5% urea in water, the freezing point has a localized minimum of -11 ˚C, but does not separate when frozen. Therefore, in vehicle operation, main components are heated to ensure metering function can start shortly after a cold start [1].

Urea first forms ammonia before the actual SCR reaction starts. This happens in two steps, called as hydrolysis reaction. First, NH_3 and isocyanic acid forms:

$(NH_2)_2CO \rightarrow NH_3 + HNCO$

Then isocyanic acid is converted with water to obtain ammonia and carbon dioxide.

$HNCO + H_2O \rightarrow NH_3 + CO_2$

To prevent precipitation of solids, second reaction must take place rapidly by selecting suitable catalysts and temperatures that are sufficiently high. Then, ammonia produced starts to react in the SCR catalytic converter according to the following series of equations:

$4NO + 4NH_3 + O_2 \rightarrow 4N_2 + 6H_2O$

$NO + NO_2 + 2NH_3 \rightarrow 2N_2 + 3H_2O$

$6NO_2 + 8NH_3 \rightarrow 7N_2 + 12H_2O$

At low temperatures (<300 ˚C), conversion mainly takes place through the second reaction above. For this reason, it is necessary to set a 1 to 1 ratio for NO_2: NO to achieve good conversion. Oxidizing NO to form NOx occurs in an upstream oxidation catalyst and this is necessary to achieve optimized efficiency.

If more reducing agent is dispensed than is converted in reduction with NOx, this can result in NH_3 leakage. However, this problem can be eliminated by placing an additional oxidation catalyst downstream of the SCR catalytic converter. This blocking converter oxidizes any ammonia to form N_2 and H_2O.

The reducing agent requirement depends on the specific NOx emission of the engine. Modern SCR catalytic converters achieve a NOx conversion rate of above 50% at temperatures above 250 ˚C. Optimized conversion rates are attained within a temperature range of 250 to 450 ˚C. Below schematic shows a modern SCR system layout.

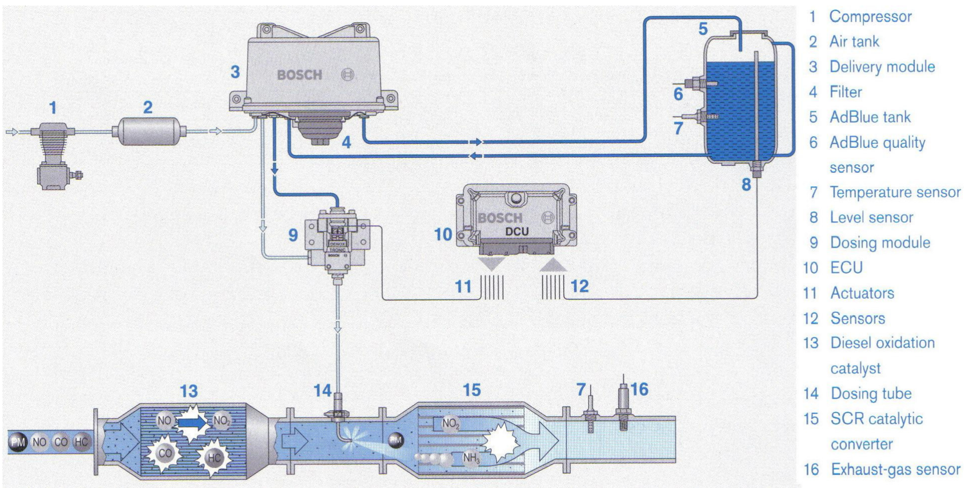


Figure 5.12 Schematic of SCR system [1]

5.5.4 Diesel Particulate Filter

Soot particles emitted from a diesel engine can be effectively removed from the exhaust gas by diesel particulate filters (DPF). Most of the current DPFs consist of porous ceramics.

Ceramic particulate filters consist of honeycomb structure made of silicon carbide or cordierite, having large number of parallel, generally square channels. The thickness generally varies between 300 to 400 μm. Adjacent channels are closed off at each end by ceramic plugs to force the exhaust gas to penetrate through the porous ceramic walls. As soot particles pass through the walls they are transported into the pore walls by diffusion where they adhere. As the filter becomes increasingly saturated with soot, a layer of soot forms on the surface of the channel walls. However, excessive saturation must be prevented due to build up in the exhaust backpressure, hindering vehicle performance and component durability.

Many filters have symmetrical arrangements of square inlet and outlet channels. However, some types incorporate octosquare substrates, having larger octagonal inlet channels and smaller square outlet channels, increasing the efficiency and storability of the system.

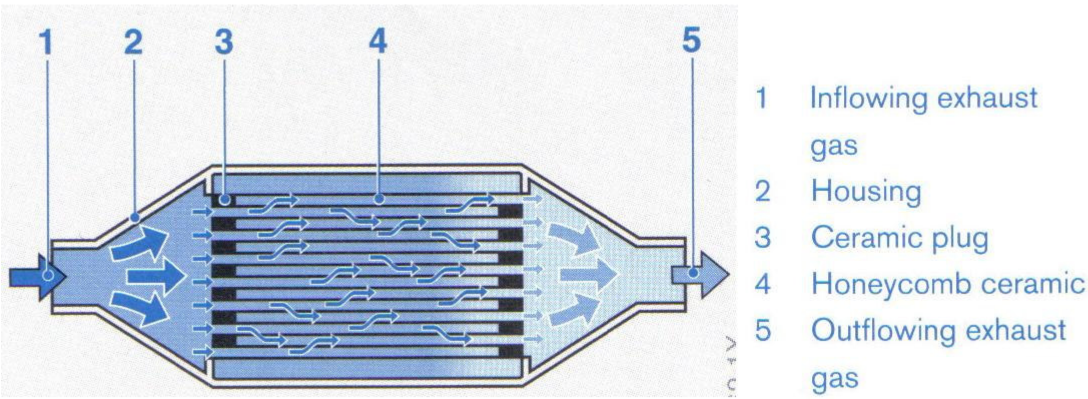


Figure 5.13 Ceramic particulate filter [1]

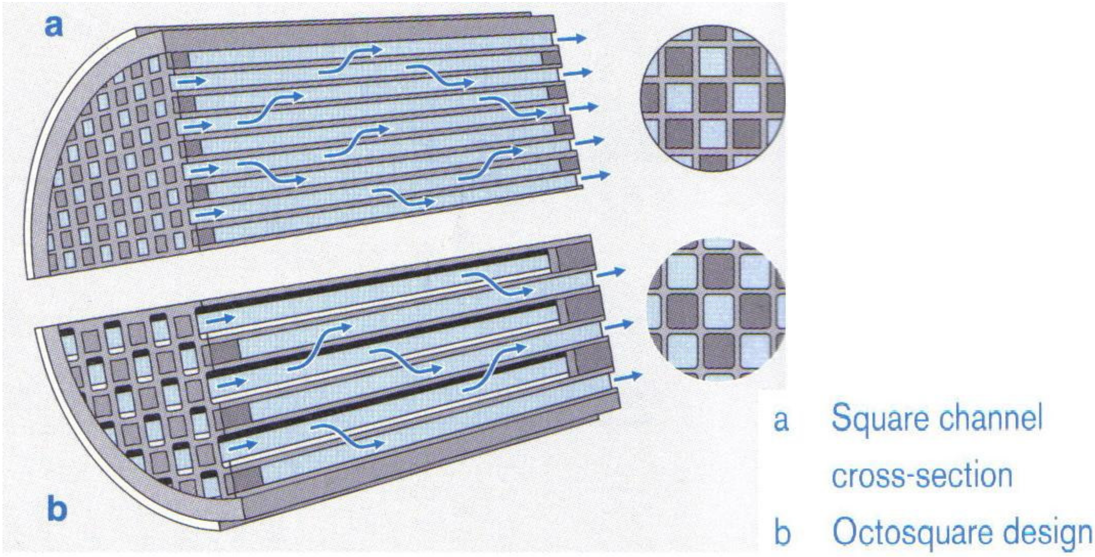


Figure 5.14 Different designs of ceramic particulate filters [1]

The particulates stored in the filter must be freed from time to time with a process called regeneration. Depending on the DPF size, vehicle and driving conditions, regenerations take place every 500 to 1200 km to prevent the filter from clogging. This process usually takes about 10 to 30 minutes. The filter is regenerated by burning off the soot collected inside the DPF. The particle carbon component can be burned using the oxygen present in the exhaust gas above a temperature of approximately 600 ˚C to form CO_2.

Soot particle burn off can also be improved by coating the filter with noble metals like platinum. The catalyzed coating has the functions of oxidizing CO and HC and NO to form NO_2. At high HC and CO emissions, system acts like oxidation catalytic converter. The energy released results in temperature hikes acting directly on the point where high temperatures are required to ignite the soot. This way, any heat loss can be eliminated by using a separate upstream oxidation converter.

NO is oxidized on the catalyzed coating to form NO_2. NO_2 is a more active oxidizer than O2, leading to oxidizing of soot at lower temperatures. In this reaction, NO_2 is again reduced to NO. Since exhaust gas flow velocity through the filter wall is slow, any NO occurring can diffuse contrary to the flow direction and oxidize soot in further oxidation reducing cycles.

For regeneration, control system senses the state of the filter, defines the regeneration strategy and monitors the filter as well as controlling the fuel injection and air intake systems during the process. To perform this operation, DPF system consists of following components:

Diesel Oxidation Catalyst: Its main function is to minimize HC and CO emissions. It also acts as catalytic burner helping to increase exhaust gas temperature for regeneration.

Differential Pressure Sensor: This sensor measures the pressure drop across the particulate filter to calculate filter saturation. This can also be used to calculate exhaust gas backpressure to be limited to a maximum value.

Temperature Sensor Upstream of DPF: During regeneration, temperature upstream of DPF is an important parameter to determine soot burn-off in the filter.

5.5.4.1. Regeneration Process

As the quantity of the soot mass rises in the filter, regeneration must be triggered in the appropriate time. As the filter saturation increases, the heat released during regeneration also increases, as well as peak temperatures in the filter. This carries the risk of damaging the filter, so, regeneration must be triggered before a critical saturation state is reached.

The strategy also has to bring forward the regeneration if the conditions are favorable like driving on the highway, or delay regeneration when the conditions are far from ideal.

When a regeneration request is received, the fuel injection and air intake systems are switched over to other setpoint parameters, but the driver cannot sense any change in torque output or noise. The new setpoints to achieve the required regeneration temperature in the exhaust gas depend on the operating point. The exhaust gas temperature is monitored and controlled continuously during regeneration to avoid any reliability problems or possible failures in the system.

The exhaust gas temperature of around 550 to 650 ˚C required during regeneration is hardly reached during a normal operation of a diesel engine except to areas close to full load. The overcome this problem, during regeneration, certain measures are taken to ensure the required exhaust gas temperature is reached. These are retarding main injection, adding a post or double post injection, which do not contribute to torque output but supplies exotherm to heat up the exhaust gas or intake throttling to reduce the amount of intake air. Depending on the engine operating point, one or several of these measures are triggered during regeneration. By retarding the injection or adding post injections, exhaust gas temperature is increased due to oxidation of fuel in the DOC. In some applications, a vaporizer system is incorporated which injects fuel vapor directly into the exhaust gas before the DOC to increase the amount of HC even more. Aside from these measures, there also has to be at least 5% of residual oxygen to realize regeneration. In addition, exhaust gas recirculation is shut down to avoid high components of unburned hydrocarbons in the combustion air. At the same time, this provides a more stable air mass control.

6. EXPERIMENTAL WORK

6.1 Aim of the Experiments

The aim of the thesis work is to investigate the effects of four different diesel injector nozzles on engine-out exhaust gas NOx – Soot emissions trade-off. The injector nozzles differ in three of the most influential parameters on combustion, namely, injector hydraulic flow (or hole size), cone angle and number of holes. Injectors are supplied from Denso Corporation. The engine used during the tests is a Ford DI diesel engine from the PUMA engine family fitted on Ford Transit commercial vehicles. The 5-cylinder 200 PS engine features latest generation technologies including high-pressure EGR system, VGT turbocharger and 1800 bar common rail injection technology. Engine is Euro-4 emission level compliant and is in serial production since end of 2007.

The so called 'Injector Screening' work is generally a common practice in automotive industry, especially during early phases of an engine development project to select the best possible nozzle to satisfy program targets like power and torque requirements as well as emission regulation targets. Selecting the best injector nozzle minimizes the calibration work and program risks that may arise later on in the project.

6.2 Effect of Nozzle Design Parameters on Emissions

Considering emission regulations, which are becoming even stricter over the years, smaller droplet sizes, better and faster mixture formation and more control over the combustion are now a must for engine designers to satisfy the regulation targets. Thanks to the advance of the new manufacturing technologies, it is now possible to control injector nozzle hole sizes to a much greater extent and produce injectors having even up to 9 holes having very close dimensional tolerances to each other. This, of course enabled injector manufacturers to build very small nozzle holes and combining this with the very high injection pressures, previously mentioned 'better control over combustion' has now become a reality. At the beginning of a new project, now the engineers have to select the injectors from a large production portfolio with many different design parameters. Of all of them, the main parameters can be counted as injector hydraulic flow, number of holes, cone angle, hole type (straight, tapered, etc…), hole fuel entrance edge type (rounded or sharp-edged), hole L/D ratio and even hydro-grinding rate. CAE analyses show very satisfying results to reduce the options down to three or four, but still real-life testing is mandatory to make a final and successful decision.

Nishida, Zhang and Manabe have shown in their experiments that micro-hole nozzles have much better mixture properties than conventional nozzles. At the same pressures, the spray angle is increased and the Sauter mean diameter is reduced obviously. The hole diameters of 0.10mm and 0.08mm have similar trend, but the 0.08mm has much smaller droplet size. This indicates that the hole diameter of 0.08mm has a great influence on the generation of high level turbulence within the liquid phase and the implosion of cavitation-bubbles which contribute to the disintegration of the spray. In addition, they have written that micro-hole nozzle can help to increase the fuel vaporization rate and then decrease the duration of uniform mixture preparation. The ultra-high injection pressure of 300MPa and micro-hole diameter of 0.08mm are very effective to increasing the turbulent mixing rate and reducing the droplet size, which are very important to shorten the duration of mixture preparation and to form the premixed lean mixture [4].

For the nozzle hole entrance configurations, Su, Farrell and Nagarajan tested two different nozzle configurations having sharp-edged and rounded nozzle hole inlets. They concluded that higher injection pressure is required for the SEI (Sharp-edged inlet) tip to maintain the same injection rate under the same injection duration, which produces wider spray angle, smaller overall average Sauter Mean Diameter (SMD), and smaller value of particulate emission according to the results. Higher injection pressure results in longer spray tip penetration, narrower spray angles, and smaller droplet sizes for both SEI and RI nozzle tips. The difference in SMD becomes smaller as the injection pressure increases. Under the same high injection pressure (160 MPa) condition, the SEI tip takes longer injection duration to deliver the mass and produces larger overall average SMD due to the lower rate-of-injection meaning lower injection velocity [5].

Siewert studied the effects of wider cone angles on combustion and emissions and he showed that wider spray angles carry a risk of missing the piston bowl for advanced injection timings resulting in high HC, CO and smoke emissions and significant reduction in thermal efficiency. He emphasizes cone angle is important for setting up a good rotation inside the piston bowl for better mixture. Injection sprays missing the bowl or perpendicular to the bowl surface create uncertain rotations hampering the combustion efficiency and results in smoke increase [6]. Kook [7] and Opat [8] each also have shown deterioration in emissions for advanced injection timings in smaller-bore diesel engines. Use of the spray model for those applications has also shown the same strong correlations between the calculated amounts of fuel that misses the bowl and the deterioration in emissions. Finally, Su and Farrell shown for the sprays contained inside the piston bowl that nozzle injection angle has a significant effect on the spray characteristics. Comparing the spray characteristics results of different injection angle nozzles, the larger injection angle nozzle produces a narrower spray angle and a larger overall average SMD. Also, the 140° nozzle has a slightly higher level of NOx than the 125° nozzle at higher injection pressures; however, the effect of nozzle injection angle on particulate emission was small in this case [9].

6.3 Properties of the Test Injectors

The properties of the injectors used in the experiments are shown on Table 6.1.

Table 6.1 Specifications of test injectors

Ford Injector Part Number	Hydraulic Flow	Hole Diameter [mm]	Number of Holes	Cone Angle [deg]
AB [Base]	370	0.129	7	153
AA1	340	0.124	7	153
AA4	370	0.129	7	158
AA9	370	0.121	8	153
Ford Injector Part Number	**Discharge Coefficient**	**Sac Volume [mm3]**	**Hole Shape**	**Hydro-Grinding Rate [%]**
AB [Base]	0.88	0.15	Straight	70
AA1	0.88	0.15	Straight	70
AA4	0.88	0.15	Straight	70
AA9	0.88	0.18	Straight	70

Hydraulic flow is defined as cc/30 sec at 100 MPa. This unit shows that if the injector is held open for 30 seconds at 100 MPa fuel pressure, the resulting flow will be equal to the defined value.

6.4 Test Engine Specifications

During the dynamometer tests, Ford PUMA I5 3.2 liter DI diesel engine was used, whose specifications are given on Table 6.2.

Table 6.2 Specifications of the test engine

PUMA I5 Euro 4 DI Diesel Engine	
Displacement	3202 cc
No of Cylinders	5 / Inline
No of Valves / Cylinder	4
Bore	89.9 mm
Stroke	100.9 mm
Compression Ratio	17.5
Injection System	1800 bar Common Rail
Rated Power	200 PS
Max. Torque	470 Nm
Max. Engine Speed	4650 rpm
Firing Order	1 - 2 - 4 – 5 – 3
Oil Gallery Volume	9.5 liter

The engine is capable of sustaining its maximum torque of 470 Nm constant between engine speeds of 1700 to 2500 rpm and rated power of 200 PS is delivered at 3500 rpm. Maximum engine speed is electronically restricted to 4650 rpm. Denso Corporation supplies the fuel injection system having the capability of 1800 bar common-rail pressure and ECU is supplied by Visteon. Test engine also features VGT (Variable Geometry Turbine) turbocharger and high-pressure EGR system to meet the strict demands of the Euro-4 emission regulation.

6.5 Testing and Measuring Equipment

Ford Otosan dynamometer facility was used for the tests, which is located in company’s Gölcük Plant. The details of the testing systems are presented below:

6.5.1 Engine Dynamometer

AVL Elin Highly Dynamic Engine Dynamometer was used for testing. This dynamometer is capable of running vehicle simulations on engine due to its very low inertia and fast response, but test point measurements were taken at steady-state engine conditions. Table 6.3 summarizes the technical specifications of the dynamometer:

Table 6.3 Dynamometer specifications

AVL Elin Dynamometer	
Max. Torque Capacity	600 Nm
Max. Power	250 kW
Overall Accuracy	0.10%
Torque Response (0-Max Torque)	20 ms
Speed Response	5000 rpm/s
Inertia	0.3 kg.m2

6.5.2 Smoke Meter

AVL 415S Variable Sampling Smoke Meter was used to measure engine out soot levels. Device output indicating the amount of soot level is FSN (Filter Smoke Number).

Working Principle: The Smoke meter probe draws a volume of exhaust gas from the exhaust line through a clean filter paper. Then, it measures the exhaust gas volume that passes through the filter paper in a metering pipe and by calculating the effective length; it records the extent to which soot has blackened the filter paper with an optical reflectometer head. After this, it determines the soot content in the exhaust gas from the paper blackening and effective length and outputs the soot content as FSN. The output range is 0 to 10. 0 FSN indicates pure white paper (no soot) and 10 FSN is pure black.

The sampling volume per unit of time is approximately constant. The measured values therefore correspond to the mean value of the soot content over the sampling time.

Table 6.4 Smoke meter specifications

AVL 415S Variable Sampling Smoke Meter	
Measurement Range for Smoke Value	0 to 10 FSN
Resolution	0.001 FSN
Repeatability	σ ≤ ± (0.005 FSN + 3 % of the measured value)
Reproducibility	σ ≤ ± (0.005 FSN + 6 % of the measured value)
Ambient Temperature Functional Range	5 to 55 °C
Humidity	95 % max. with no moisture condensation
Exhaust Gas Backpressure	-100 to +400 mbar
Permissible Exhaust Gas Temperature	600 °C

6.5.3 Fuel Mass Flow Meter

AVL Fuel Mass Flow Meter 735S was used to measure fuel consumption during the tests. With a high accuracy mass flow sensor, fuel consumption can be determined continuously via direct mass flow measurement in kg/h. The density does not have to be determined in addition to the direct determination of the gravimetric or specific fuel consumption like in volumetric measurement methods. The fuel consumption can thus be determined to an accuracy of 0.12% for the whole system under real test bed conditions.

Table 6.5 Fuel mass flow meter specifications

AVL Fuel Mass Flow Meter 735S	
Measuring Range	0 to 125 kg/h
Density Measurement Uncertainty	≤ 0.0005 g/cm3
Maximum Measurement Frequency	20 Hz
Ambient Temperature	0 to 50° C
Fuel Supply Pressure	0.1 to 0.8 bar
Fuel Supply Temperature	- 10 to + 40° C
Outlet Pressure	adjustable from 0.05 to 0.5 bar

6.5.4 Fuel Temperature Control

AVL Fuel Temperature Control 753C which works in tandem with AVL Fuel Mass Control was used to held fuel temperature constant at 40 °C throughout the tests to minimize any possible variance on combustion entry condition fluctuations. System uses chilled water to cool the fuel, which is constantly recirculated.

Table 6.6 Fuel temperature control system specifications

AVL Fuel Temperature Control 753C	
Max. Temperature Control Range	10 to 80° C
Temperature Stability	+ / - 0.02° C
Cooling Power	1.6 kW at 10° C spread
Ambient Temperature	5 to 40° C
Fuel Circulation Capacity	250 l/h
Fuel Set Temperature Range:	–30° C to 80° C
Heating Power	1.6 kW
Feed Pressure	0 to 6 bar
Return Pressure	0 to 0.8 bar

6.5.5 NOx Analyzer

AVL CLD 4000 HC NOx analyzer was used to measure engine out NOx level. Device output unit is in ppm, but this is later on normalized to g/kWh to form an independent basis to compare different injectors. System uses a photo multiplier to detect NO_2*, which is at higher energy level after the chemical reaction shown below, taking place inside the ozonator.

$2NO + 4/3O_3 \rightarrow O_2 + NO_2 + NO_2^*$

To get the correct NOx level within the exhaust gas, all the NOx components are converted to NO inside the converter before the chemical reactions in the ozonator. The final ppm results are then corrected back to reflect the original NOx level before the conversion takes place. The operation range of the analyzer is between 10 to 10000 ppm.

The engine itself, ECU, instrumentation and the surrounding equipment can be seen in the following figure.

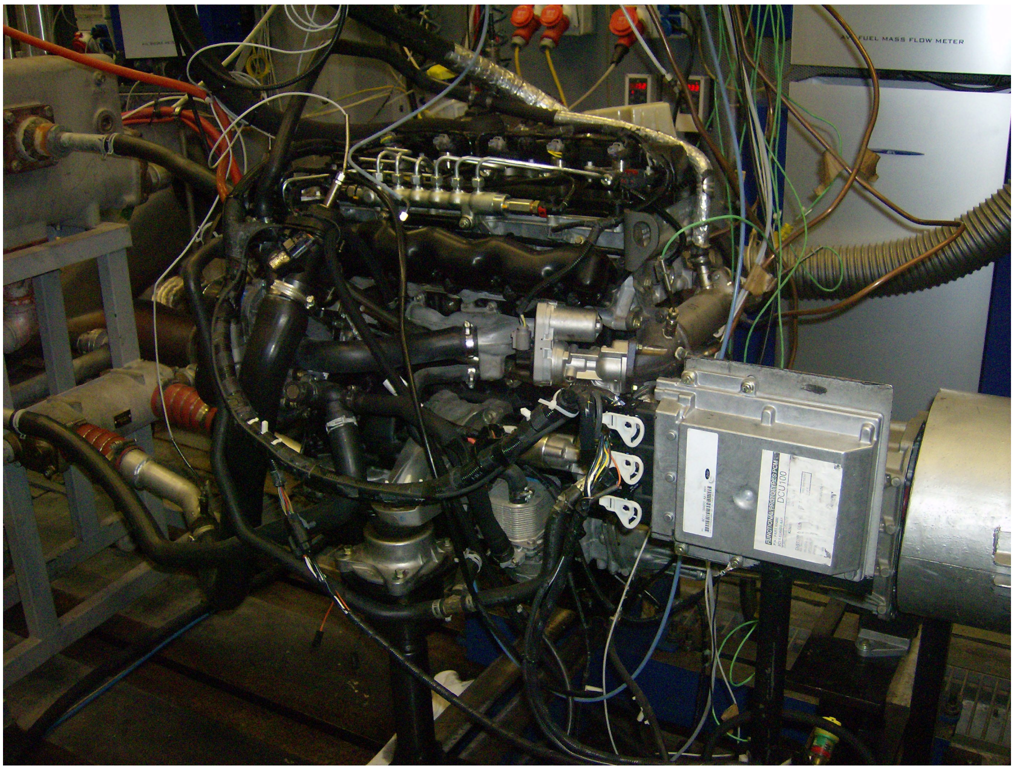


Figure 6.1 Test engine and test cell

6.6 Test Points and Experimentation

Two different steady-state test points were selected to test four different types of injector nozzles. NEDC (New European Driving Cycle) was used as the basis of test point selection, which is the drive cycle applied for European Emission Regulation certification. Currently, Euro-4 is the valid regulation across Europe. The reason of using NEDC for test point selection is that it covers a wide range of engine speed and load points, reflecting both city and highway driving conditions. Secondly, it would be illogical to select a point outside the torque & speed window of the NEDC cycle since it will be irrelevant to compare engine out emission characteristics of injectors at points where emissions is not a priority any longer.

6.6.1 Brief Information on European Emission Regulations

The so-called 'New European Driving Cycle' is used which is performed on a chassis dynamometer to certify light duty vehicles in Europe. The cycle consists of four consecutive ECE segments to start with and then followed by a EUDC segment. Before the test, the vehicle has to be soaked at least 6 hours at an ambient temperature of between 20 to 30 ˚C, and the cycle starts immediately without allowing a prior warming-up of the engine. Emission sampling also starts as soon as the engine starts. This modified cold-start procedure has been added after year 2000. Exhaust gas is diluted with ambient air before being collected in separate bags for each segment. After the test, these bag samples are analyzed back and results are expressed in g/km for each pollutant. If the average of the results over the three tests is within the limits, vehicle is then certified as ‘Emissions Compliant’.

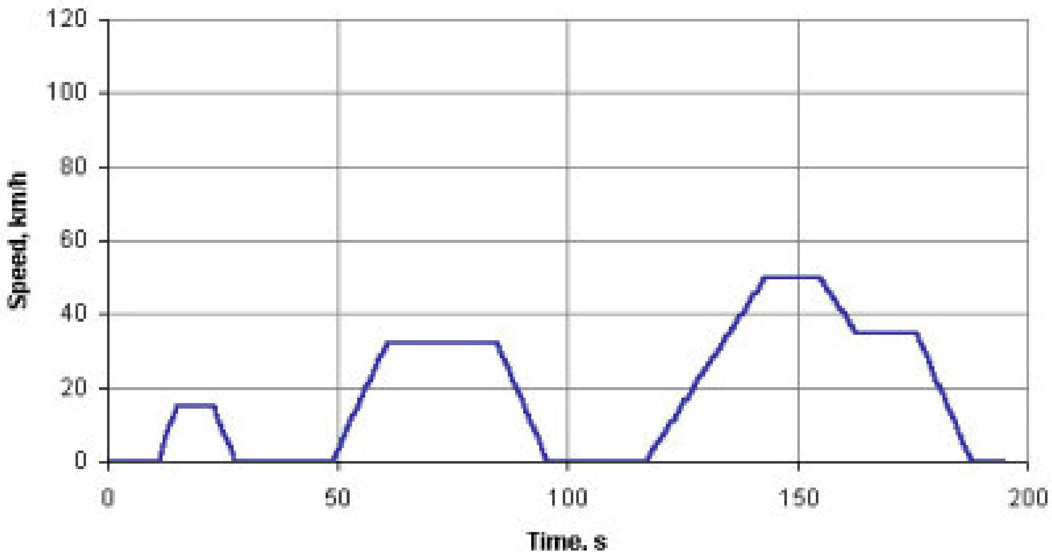


Figure 6.2 ECE 15 cycle [12]

ECE is urban drive cycle devised to reflect city driving conditions. It is characterized by low vehicle speed, engine load and exhaust gas temperatures.

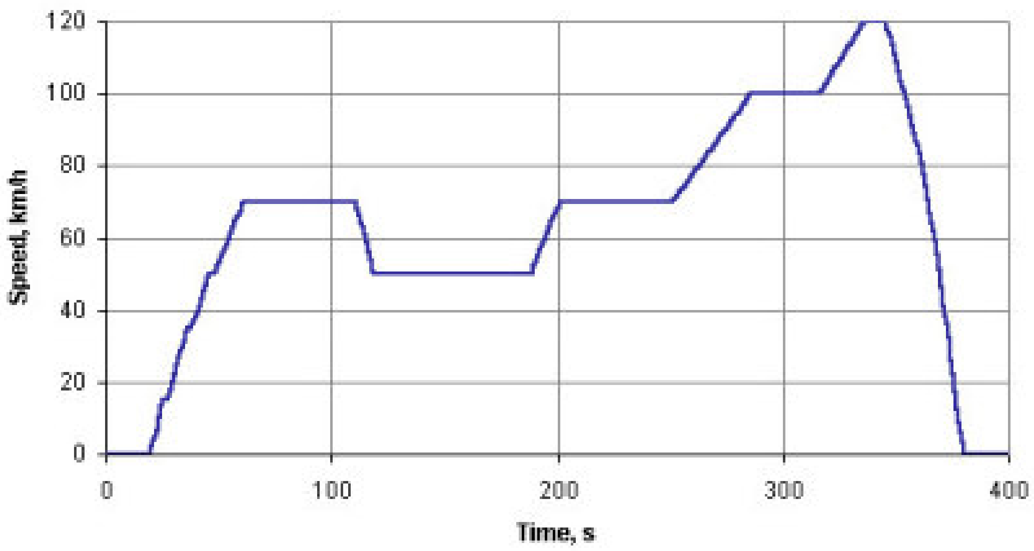


Figure 6.3 EUDC cycle [12]

EUDC cycle is added to represent more aggressive, high-speed driving conditions. There is a second alternative EUDC cycle with reduced maximum speed (90 km/h versus 120 km/h) for low powered vehicles [12].

Euro-4 emission regulation has been in place in Europe since year 2005, but after 09.2009, automotive companies have to satisfy Euro-5 norms for new types and 01.2011 is the deadline to switch to Euro-5 for all the productions. With Euro-5, there are many changes imposed on the manufacturers. Table 6.7 shows Euro-4 and Euro-5 norms in a comparative basis for M and N1 Class-I categories where usually passenger cars fall in.

Table 6.7 Euro-4 and Euro-5 emission norms [13]

Applicable categories: M, N1 class I		EURO 4		EURO 5^(0)	
		Limit	OBD	Limit EURO5/5+	OBD EURO5/5+
mandatory Application Date	new type	01/2005	01/2003	09/2009	
	new vehicle	01/2006	01/2004	01/2011	
CO (mg/km)		500	3200	500	1900
NMHC (mg/km)		-	400	-	**320**
NOx (mg/km)		250	1200	180	540
THC + NOx (mg/km)		300	-	230	-
PM (mg/km)		25	180	5/**4.5**	50/80
PN (#/km)		-	-	-/**6** E+11	-

The biggest change coming with Euro-5 is the substantial reduction in Particulate Matter (PM) emissions. The reason behind this change is to force manufacturers to use DPF systems since achieving those limits only with measures inside the engine will be impossible. Additionally, Particulate Number Index is now introduced which also has a limit to eliminate emitting very small particulates to the atmosphere. The logic being this change is that the smallest particulates are the most harmful ones for human health. Finally, OBD limits are much more tightened that if a failure, which can deteriorate emissions happens, the failure action should also satisfy OBD emission limits in any case, necessitating more cautious actions to be taken as Euro-4 OBD limits were much more far away.

6.6.2 Determination of Test Points

Test points were selected from an actual NEDC drive cycle run at Ford Motor Company Dunton Emissions Laboratory on a Ford Transit Light Duty vehicle with the Puma I5 DI Diesel Engine fitted on. During the drive, emission traces were recorded by the facility and data from the vehicle ECU were recorded by a special data acquisition device and software supplied by ATI Lab. Inc. With the help of a post-processing code developed in MatLab environment, both sets of data were synchronized and below graph showing HC, CO and NOx emissions with respect to vehicle speed and time was obtained.

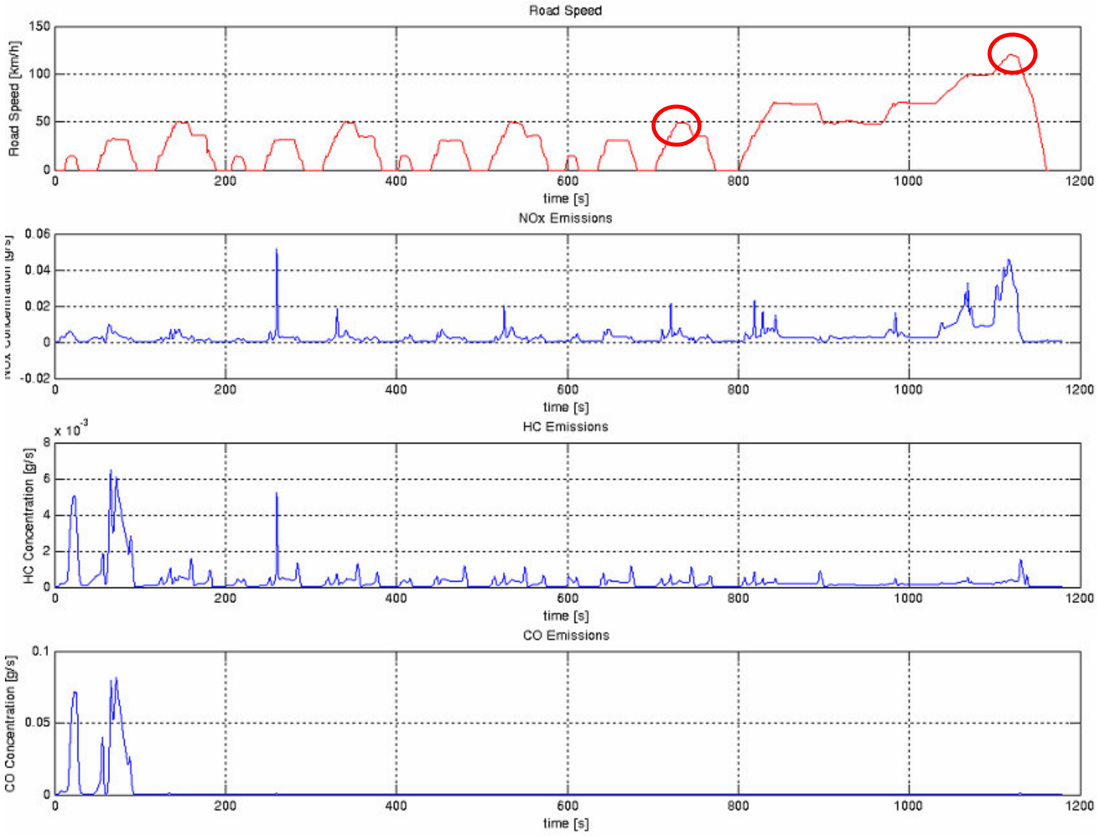


Figure 6.4 NEDC cycle and tailpipe emissions run on Ford Transit

As shown in the data, CO emission reduced to nearly zero around 100 seconds after the test started because oxidation catalyst reached its light-off temperature by then. In a similar manner, HC emission was also reduced substantially at the same instance although it was not as low as CO emissions throughout the rest of the cycle due to comparatively lower conversion efficiency of the oxidation catalyst for HC. There seems a spike around 230th second in HC and CO emissions but this is later on confirmed by the Emissions Lab that a sensor problem happened at that instant.

Looking at the NOx emission trace, highest emission occurred when the vehicle was cruising at 120 km/h, which is the top speed of the NEDC cycle. So, definitely this point should be selected to compare the injectors' NOx – PM trade-off. Second point selected is the highest speed cruise point of the ECE cycle, which is taken at 50 km/h. Second test point could again be selected from the EUDC part where both speed and engine load are higher than ECE part, but in that case, point 1 and 2 would be similar in characteristics. By choosing a point from the ECE part, the engine speed and load is now much lower compared to point 1 since it represents city driving conditions. Moreover, this 50 km/h cruise point is repeated four times throughout the cycle, so in terms of overall contribution, point 2 is as important as point 1 for the total NOx contribution.

After detecting the test points, same steady-state conditions were run on the engine dynamometer to see the real brake torque at those operating points. Since the test engine features VGT and EGR technologies as well as common-rail injection system, there are six parameters that can be independently and electronically controlled by the ECU, which are controlling the combustion inside the cylinders and thus emission characteristics. Those are boost pressure, EGR valve lift, start of main injection, pre injection separation duration, pre injection amount and rail pressure. During the roll test, all above parameters were recorded and values for each of them in the base calibration were determined for the selected test points. The base calibration was already a Euro-4 certified calibration, so testing the injectors around those setpoints was a good practice. After that, a design of experiments was set up for the test points around the base calibration values with two steps for each calibrateable, one on the negative and one on the positive side. The design of experiment values are shown on Tables 6.8 and 6.9.

Table 6.8 Design of experiments for test point-1

POINT 1:	**120 km/h Cruise**
Engine Speed [rpm]:	2934
Brake Torque [Nm]:	150
Percent Load:	33
MAP (Manifold Air Pressure) [kPa]:	150 – **165** - 180
Main Injection Start of Injection [degBTDC]:	2.5 - **(-1)** - (-4.5)
Pre Injection Separation [us]:	1100 - **800** - 500
Pre Injection Volume [mm3/str]:	1 - **1.5** - 2.5
Rail Pressure [MPa]:	100 – **125** - 150

Table 6.9 Design of experiments for test point-2

POINT 2:	50 km/h Cruise
Engine Speed [rpm]:	2591
Brake Torque [Nm]:	75
Percent Load:	7
MAP (Manifold Air Pressure) [kPa]:	105 - **115** - 125
Main Injection Start of Injection [degBTDC]:	2 - **(-1.8)** - (-6)
Pre Injection Separation [us]:	850 - **1075** - 1300
Pre Injection Volume [mm3/str]:	1 - **1.5** - 2.5
Rail Pressure [MPa]:	30 - **60** - 90

The middle values shown in bold cases are base calibration values. Step sizes are determined as far away from the base values as possible to test in a wide operating window, but also selected as close as possible for not to test unrealistic points which may never occur at that specific point when calibrating according to the NEDC cycle. After determining the design of experiment points, only one parameter was changed at a time while the rest were held constant at base calibration values during the test. To obtain the NOx – PM trade-off curve, the sixth variable, which is the EGR valve lift, was swept between 0 mm valve lift (no EGR condition) to maximum lift value of 5.25 mm with two additional points in between to divide the total segment into four almost equal EGR rate points. Maximum possible EGR rate is mechanically limited by the relative pressure difference across the exhaust and intake manifolds, which is the only driving force to pump EGR into the cylinders. Due to the dynamic nature of the system, maximum EGR capability of the engine highly depends on the operating conditions. Figure 6.5 shows a sketch of the test engine's air path.

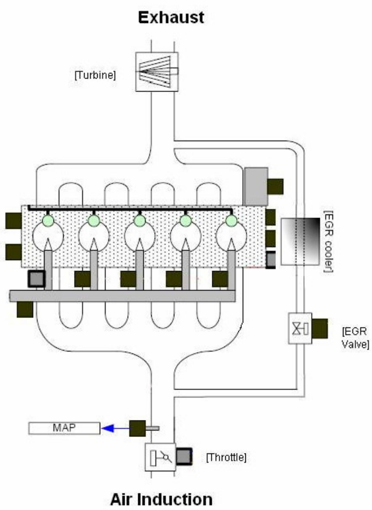


Figure 6.5 Test engine air path

6.7 Test Procedure

Before testing each set of injectors, NOx analyzer had to be warmed-up for about half an hour and calibrated with calibration gases of known NO concentrations. After this, test room ambient air was conditioned at 25 ˚C constant as well as engine clean air temperature, which was kept at 18 ˚C. Fuel temperature was also kept at 40 ˚C at engine inlet point to keep the variance as small as possible.

During the tests, an emulated ECU was used to enable changing of the engine control variables in real time. With a standard ECU, new calibration have to be flashed every time when a variable was intended to be modified which generally takes around 2 minutes and necessitates engine-off state. After every change of variable, engine is allowed to stabilize for around two minutes before taking a 30-second recording. Emissions were sampled downstream of the turbocharger turbine and upstream of the oxidation catalyst through heated lines to eliminate condensation of water vapor before entering the analyzers. All test point iterations were repeated for each set of injectors after replacing them according to the engine assembly procedure supplied by Ford Otosan.

6.8 Test Data

6.8.1 Test Point-1

6.8.1.1. NOx – PM Sensitivity for Boost Pressure Changes

Boost pressure sensitivity was tested at two different boost pressures of 150 and 180 kPa as well as the base calibration value of 165 kPa. Resulting figures are shown on Figures A.1, A.2 and A.3.

Due to the dynamics of the EGR system explained before, maximum possible EGR rates were 25%, 27% and 30 % for boost pressures of 150 kPa, 165 kPa and 180 kPa respectively. When the boost pressure setpoint is increased, turbine blades start to close more to achieve that setpoint which in effect increases the backpressure on the exhaust side. Boost pressure also increases, but as a general trend, with higher boost pressures, the pressure difference across the intake and exhaust manifolds tends to favor more EGR flow. The usual NOx – Soot hyperbolic curve is common for all the injectors and setpoints, with higher cone angle injector exhibiting the worst performance on all conditions. Base and 340 flow injectors are extremely close, but one would normally expect lower flow injectors to be better due to lower droplet sizes they can produce and better mixing properties, but this is not the case here. In addition, the 8-hole injector, having the smallest hole diameter was worse than AA1 and base injectors at boost pressures of 150 and 165 kPa, only coming close to them with 180 kPa boost pressure. Again, this contradicts with the assumption above.

6.8.1.2. NOx – PM Sensitivity for Start of Main Injection Changes

Start of main injection influence was tested at setpoints of 2.5° BTDC, 1° ATDC (Base Calibration) and 4.5° ATDC. Resulting data are shown on Figures A.4, A.2 and A.5 respectively.

As expected, for all the injectors, NOx emissions increase and soot levels decrease as start of injection is advanced, due to more fuel injected inside the combustion chamber until the premixed combustion starts, leading to higher internal temperatures and more efficient combustion. 158° cone angle injector is again worst of all. This time, other three injectors are very close with advanced injection, but as start of injection is retarded, base injectors seem to have the best trade-off curve where 340 flow and 8-hole injectors are close in second.

6.8.1.3. NOx – PM Sensitivity for Pre Injection Separation Changes

Pre injection separation duration from the start of main injection was tested at three different points, 500, 800 (base calibration) and 1100 μs, whose graphs are given on Figures A.7, A.2 and A.6 respectively.

Pre injection separation seems not to be so effective for NOx – Soot trade-off at this specific point. SFC values also stayed the same around 260 to 265 g/kWh across all the injectors with little variance. Similar to previous points, 158° cone angle injector showed the worst trade-off whilst AA1 and base injectors are very close again. 8-hole injector stood in between.

6.8.1.4. NOx – PM Sensitivity for Pre Injection Amount Changes

Pre injection amount was varied between 1, 1.5 (base calibration) and 2.5 mm^3/stroke values. Figures A.8, A.2 and A.9 show the respective data.

Small amount of pre injection generally tends as a NOx reducer and helps the combustion noise, since it reduces the ignition lag and thus, amount of fuel burned at premixed combustion phase. On the other hand, if the amount becomes more than necessary, it acts like an additional torque producer before the main injection and it starts to act the opposite way for emissions.

1.5 mm^3/stroke of pre-injection seems to be the best option as the trade-off curves for AA1, AA9 and base injectors are the most compacted at both the ends. As explained above, increasing the amount to 2.5 mm^3/stroke acts the opposite way and increased both NOx and soot emissions at both ends. As usual, AA9 is the worst option while base and AA1 are evenly matched, base injector having a slight advantage.

6.8.1.5. NOx – PM Sensitivity for Rail Pressure Changes

When the rail pressure increases, droplet sizes tend to be smaller and spray penetration as well as spray angle increase, leading to better mixing properties and combustion that is more efficient. This in effect increases the NOx levels at the expense of soot emission. In the tests, three rail pressure setpoints were tested, 100, 125 (base calibration) and 150 MPa. The resulting data are shown on Figures A.10, A.2 and A.11 respectively.

As expected, with increasing rail pressure, NOx increases while the opposite is true for soot. AA4 again proved to be a wrong option while AA9 is again very close to other two but slightly worse. At low EGR points, AA1 exhibits better soot emissions while at high EGR amounts, base injectors produce less soot.

6.8.2 Test Point-2

In the second test point iterations, maximum possible EGR rate increased to levels of 55% due to changes in turbocharger turbine working regime. Because of the low load and fuelling amounts, the exhaust gas energy turbine receives is now much more reduced. Therefore, to achieve the desired boost pressure, the turbine blades have to close more. The percent closure of turbine blades was in the range of 45 to 60% at point 1, but in point 2, it increased up to the range of 75 to 95% depending on the iteration. Taking also lower levels of boost pressure into consideration, relative pressure difference between the exhaust and intake manifolds now increased, resulting in higher EGR rate.

6.8.2.1. NOx – PM Sensitivity for Boost Pressure Changes

Boost pressures of 105, 115 (Base Calibration) and 125 kPa were tested. Figures A.12, A.13 and A.14 show the respective data. This time, the 8-hole injectors exhibit better trade-off characteristics compared to the 340 flow injectors compared to point

1. Again, base injectors remain the best and AA4 are still the worst. With increasing boost pressure, trade-off curves assume the usual hyperbolic shape, but tend to get vertical as boos pressure decreases, being less sensitive to EGR rate changes in terms of soot emissions. At this stage, especially for base injectors, once the EGR rate is at maximum, there is a simultaneous reduction both in NOx and soot emissions. This is although not an expected behavior, as usually increasing EGR decreases NOx and increases soot emissions, this contradictory phenomenon is still present in literature. Wagner and Green [10] showed that with excessive cooled EGR at low to moderate load conditions, same phenomenon of both reductions in NOx and soot is happening. The first approach they used to flow more EGR beyond the maximum rate was to use an intake throttle to resist clean airflow while the EGR valve lift was at maximum. This enabled them to increase the rate more than 50% while the standard hardware capability was at 44%. Figure 6.6 shows NOx and PM traces for this study.

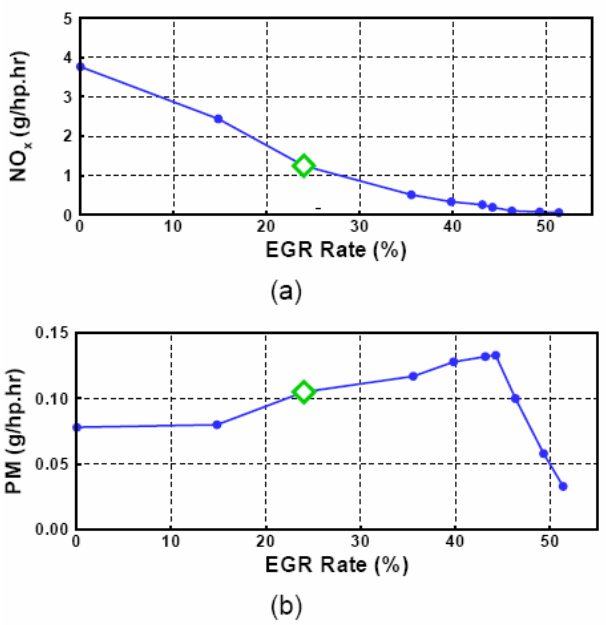


Figure 6.6 NOx and PM trace for 1500 rpm 2.6 bar BMEP condition [10]

Around 44% EGR, PM starts to decrease with increasing EGR and reaches the lowest at the maximum EGR rate point engine was tested. As expected, NOx also employs a decreasing trend with increasing EGR following the exhibiting the same pattern seen on Figures A.12 and A.13 for base injectors.

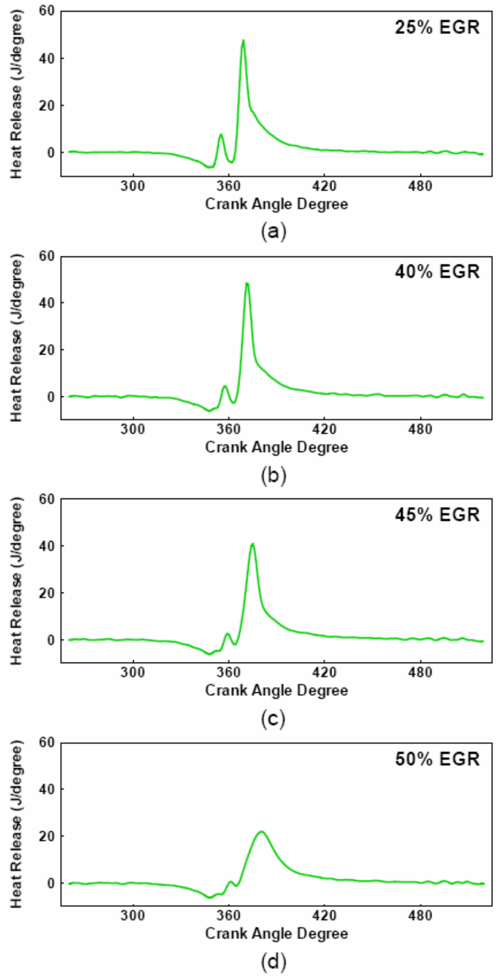


Figure 6.7 Average heat release rate with varying EGR rates at 1500 rpm and 2.6 bar BMEP [10]

Significant change occurs between EGR rates of 45% and 50%, which is also, where the dramatic PM reduction is seen. Over this range, the pilot becomes very weak, peak heat release rate decreases and the combustion duration increases. General shape of the heat release profile indicates premixed dominated combustion as compared to the classical premixed-diffusion dominated combustion, typical in diesel engines.

Akihama, Takatori and Inagaki [11] also tried to understand this so called ‘smokeless combustion’ phenomenon through combustion modeling and post processing. The way specific processes they followed can be summarized as follows.

- “Soot - Φ - T map", in which the amount of soot formed is represented as contours, was created using 0-D calculations with a detailed chemical kinetic model including PAH (polycyclic aromatic hydrocarbons) formation, soot particle nucleation, growth and surface oxidation.

- The combustion processes of the smokeless and other operating condition were simulated by 3D-CFD KIVA2 code.
- In-cylinder conditions of smokeless and another combustion predicted by 3D-CFD were plotted on the Φ - T map to investigate their behaviors and differences on the map.

Figure 6.8 shows the outcome of their studies showing in-cylinder condition simulations during combustion on a comparative basis.

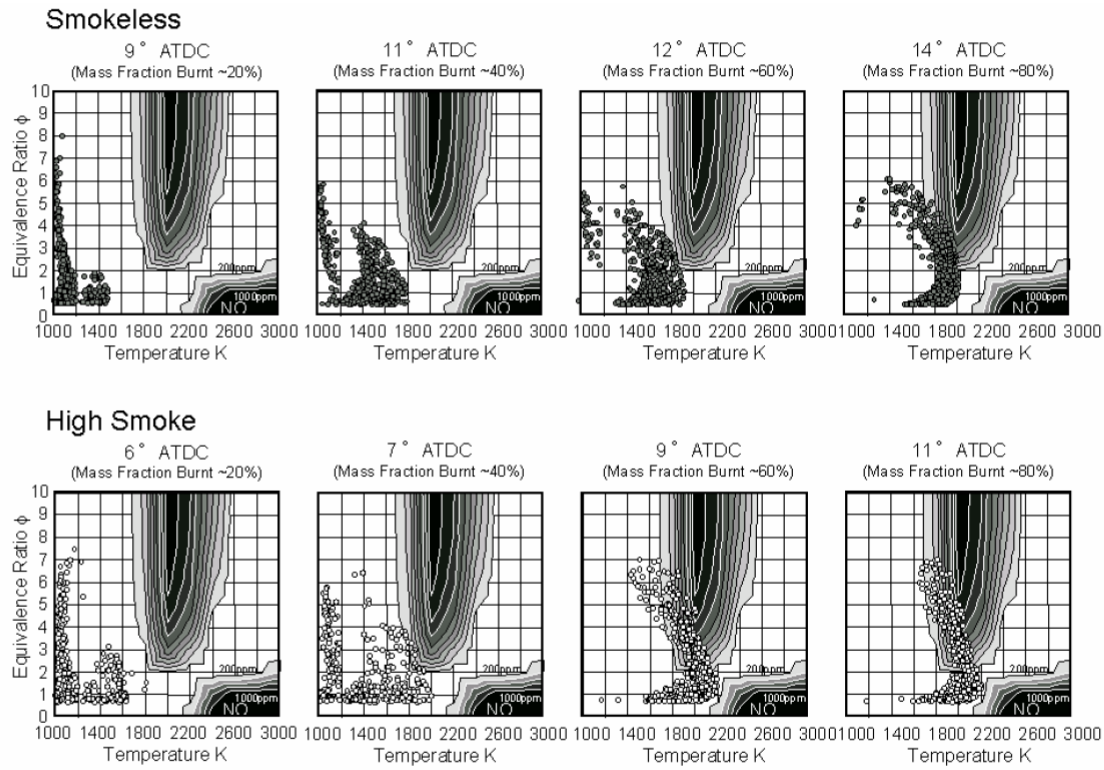


Figure 6.8 Time histories of Φ – T map for 'smokeless' and 'high smoke' combustion cases [11]

As a supplement to Figure 6.8, Figure 6.9 shows time histories of the fuel quantity existing in the soot formation region for smokeless and high smoke conditions. This shows that the combustion of the smokeless case occurs in the foot of the soot formation peninsula on the map even if the combustion condition enters the soot formation region.

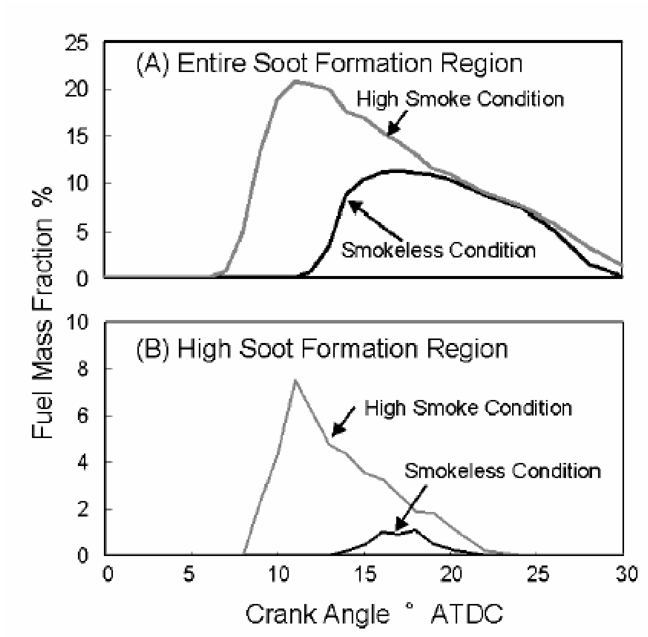


Figure 6.9 Time histories of fuel quantity existing in the soot formation region [11]

According to the authors, the simulations show the mechanism of the smokeless combustion as follows.

- The temperature of smokeless rich combustion is reduced by large amounts of EGR. This temperature reduction is sufficient to allow the combustion to avoid the soot formation region on the Φ - T map. This is the major reason of the soot suppression mechanism in the smokeless combustion.
- There exists a local rich region in the mixture of the smokeless rich combustion similar to conventional diesel combustion. However, the soot formation can be suppressed because the soot formation on the lower temperature side of soot formation peninsula is suppressed even if the rich combustion occurs. This is the major reason why the smokeless combustion is achieved with no improvement of the mixture formation by changing fuel spray system, combustion chamber geometry etc., under rich operating condition.
- The combustion temperature of smokeless rich combustion is also sufficiently low to avoid the NO formation region.

6.8.2.2. NOx – PM Sensitivity for Start of Main Injection Changes

Start of main injection values of 2˚ BTDC, 1.8˚ ATDC (Base Calibration) and 6˚ ATDC were tested. Results are shown on Figures A.15, A.13 and A.16 respectively.

With advanced start of main injection, 8-hole nozzle exhibits the best behavior while AA1 and Base are very close. However, as the injection is retarded, again the similar behavior of trade-off lines becoming vertical happens for all the injectors and base injectors seems to be giving the best trade-off again. As an outcome from the results, as the combustion starts to be less efficient, either by reduced boost pressure or very retarded start of injection, both NOx and soot emissions seem to reduce at the same time, which is a bit strange to general behavior of the systems. With point-2 iterations, 158˚ cone angle injector is still the worst one, but now 8-hole injector becomes the second best to base injectors, at some points even better.

6.8.2.3. NOx – PM Sensitivity for Pre Injection Separation Changes

Pre injection separation between main injection was tested at three different points, 850, 1075 (base calibration) and 1300 us. Figures A.17, A.13 and A.18 depict the results respectively.

Although pre injection separation proved not to be so influential at point-1, this is not the case for point-2. As can be seen from the graphs, as the pre injection comes closer to the main injection, soot amount starts to increase although NOx emission seems not to be affected too much. As a common behavior for point 2, as the combustion starts to be less efficient, a simultaneous NOx and soot reduction is still the case here, especially when the pre injection separation is 1300 us. Also, depending on the worsening combustion, SFC degraded around 30 to 40 g/kWh for 1300 us separation compared to the other two, whose SFC values are closer. Base injectors are still giving the optimum trade-off curve while AA9 is the second best. As usual, higher cone angle injector shows significantly worse results.

6.8.2.4. NOx – PM Sensitivity for Pre Injection Amount Changes

Pre injection amount was varied between 1, 1.5 (base calibration) and 2.5 mm^3/stroke values, as shown on Figures A.19, A.13 and A.20.

Similar behavior again can be seen here. With increasing pre injection amount, SFC starts to decrease and trade-off curve shifts to the right side of the graph. As also is the case with highest boost pressure and most advanced injection, where again combustion is more efficient, 8-hole injector exhibited the best trade-off curve. On the other hand, as the EGR ratio and other parameters were changed towards a less efficient combustion, Base injectors gave the better NOx and soot values.

6.8.2.5. NOx – PM Sensitivity for Rail Pressure Changes

The rail pressures tested were 30, 60 (Base Calibration) and 90 MPa. The graphs are shown on Figures A.21, A.13 and A.22 respectively.

As a general trend, with a less efficient combustion, EGR ratio does not change the soot emissions at all. This time, NOx behavior is in correlation with rail pressure changes. Higher the rail pressure, higher the NOx emissions. Base injectors again seem to have the best trade-off curve.

CONCLUSIONS AND RECOMMENDATIONS

Following the data obtained through a series of tests on four different types of injectors tested on an engine dynamometer, following conclusions and recommendations can be drawn:

- Although today's computers are very powerful and CAE analyses can lead to many conclusive results, it is seen that real world testing is still imperative to make a final decision on component alternatives. Only in this way, many practical problems and cases specific to that particular system or test part can be identified.
- Selecting two very different engine speed and load points proved to be the right decision since injectors exhibited different behaviors at each test point. Increasing the number of test points can increase this variety of different behaviors as well.
- Most of the time, effects of setpoint changes resulted in the expected way on engine out emissions, except at some iterations at point-2, where excessive EGR in a less efficient combustion led to both NOx and soot reduction at the same time at the expense of fuel economy.
- 158° cone angle injector (AA4) always yielded the worst results. Probably the cone angle is wider than the piston bowl so that some of the fuel misses the bowl, increasing soot content as well as NOx, as also shown in the studies of Su [9] and Siewert [6] respectively.
- Before the tests, one would expect the 340-flow (AA1) and 8-hole (AA9) injectors to have better trade-off curves compared to the base injectors since their respective hole sizes are smaller, 8-hole having the smallest of all. However, this again proved wrong as base injectors most of the time gave the best NOx – soot trade-off and proved to be the right choice for the program.
- 8-hole injectors being not so effective can be attributed to the air swirl movements inside the cylinder. Swirl rate is higher for this engine to enable

using 8-hole injectors, as probably the sprays are colliding with each other forming bigger droplets during the course of the injection due to smaller angle in between the sprays compared to the 7-hole injectors. However, this effect became smaller at test point-2, where torque output and engine intake air are less compared to point-1. This means less air movement inside the cylinder and due to this, less risk of colliding of sprays. In that case, 8-hole injectors were second best to base injectors. Even at some cases exhibited the best results.

- 340-flow injectors have the second smallest hole size of all, but again proved not as effective as the base injectors, especially at point-2 where 340-flow injectors exhibited higher smoke values. This can be attributed to longer injection time necessary to inject the same amount of fuel at the same rail pressure. This in effect lead to longer and less efficient combustion.

- Although utmost attention was paid to reduce variability between the tests as much as possible, due to time constraints, tests could be completed in a 2-week time span at different days. Ideal is to complete the tests back-to-back in the shortest time possible.

- In addition, FSN was used to reflect particulate contents of the exhaust gas at every iteration point. In fact, directly measuring the particulate matter content would be the best practice, but this would increase the total test time to at least ten times the original since filters had to be soaked at special conditions for hours before being ready for weighing. As the airflow remained nearly constant for the same test point across all the injectors, FSN still remains a good approach for comparison purposes.

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[12] **Url-1** <*http://www.dieselnet.com*>.

[13] **Url-2** <*http://www.siemens.com*>.

APPENDIX

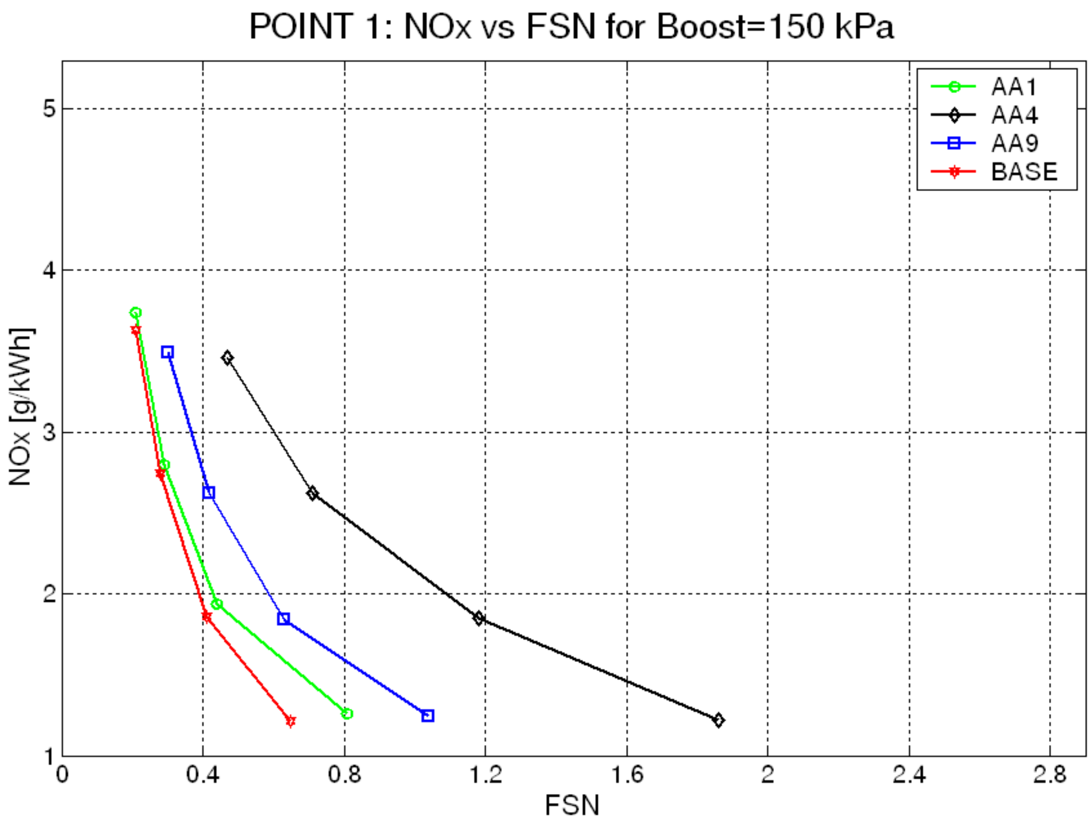

Figure A.1 Test data for point-1 boost pressure = 150 kPa

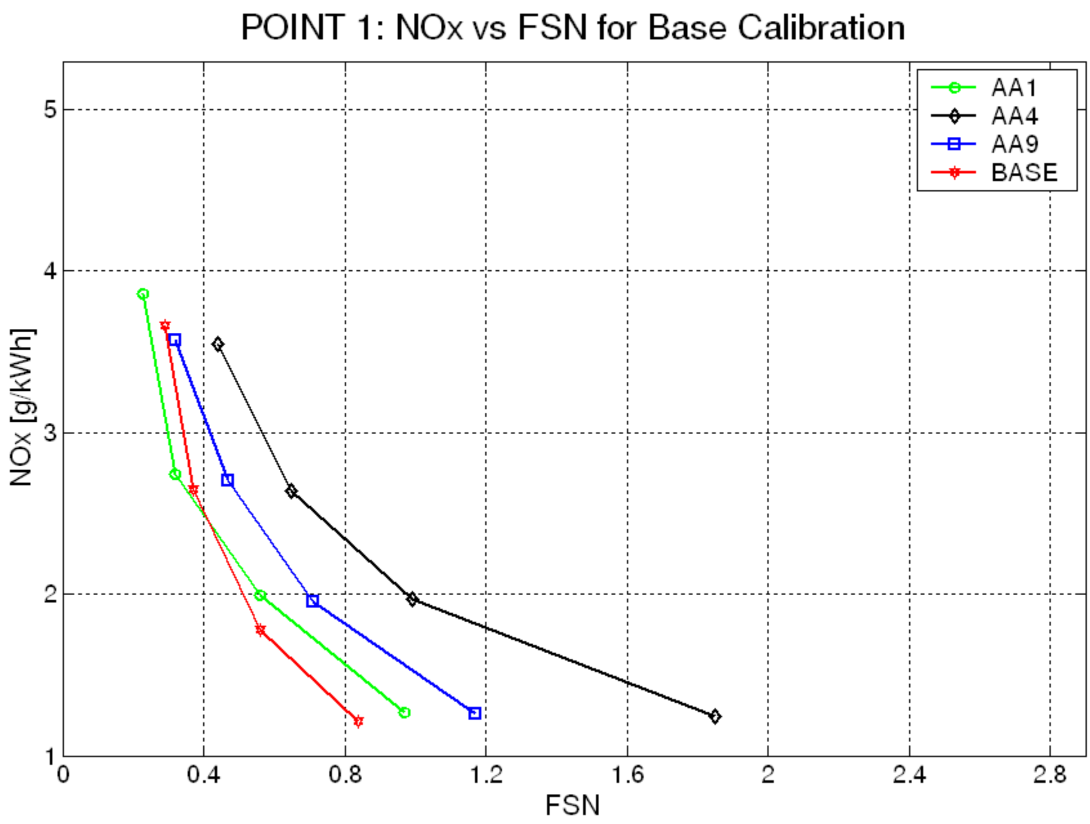

Figure A.2 Test data for point-1 base calibration

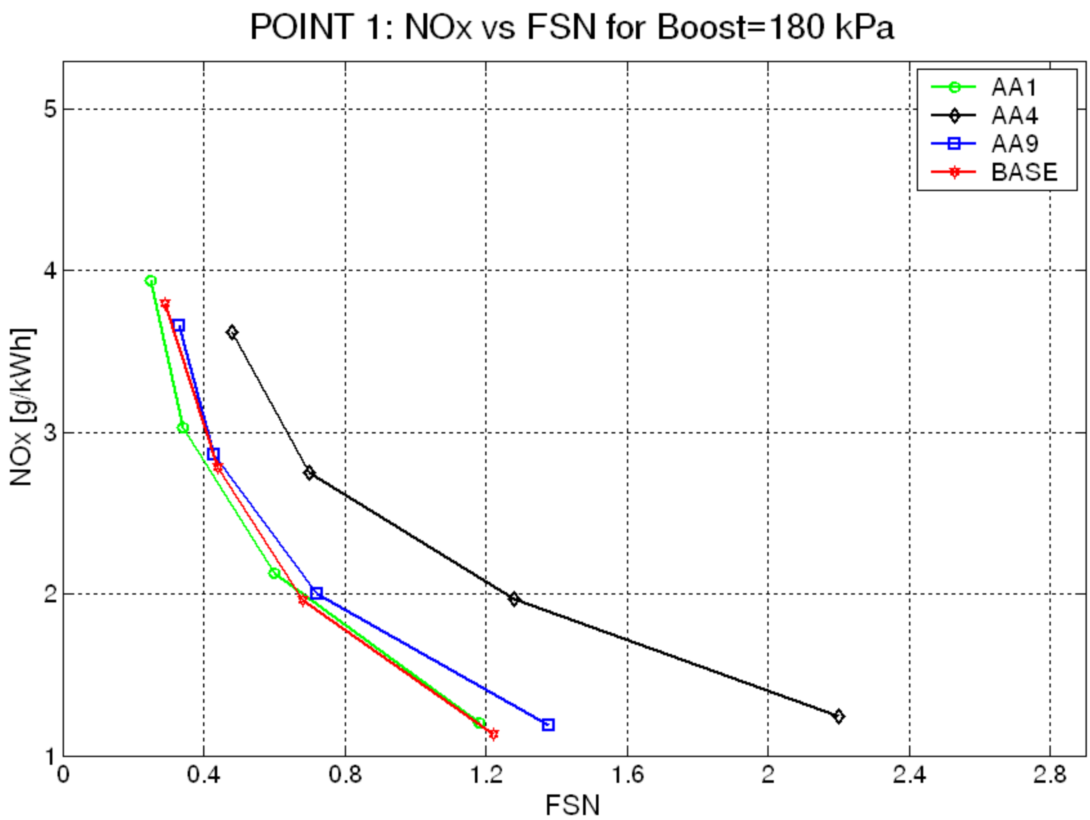

Figure A.3 Test data for point-1 boost pressure = 180 kPa

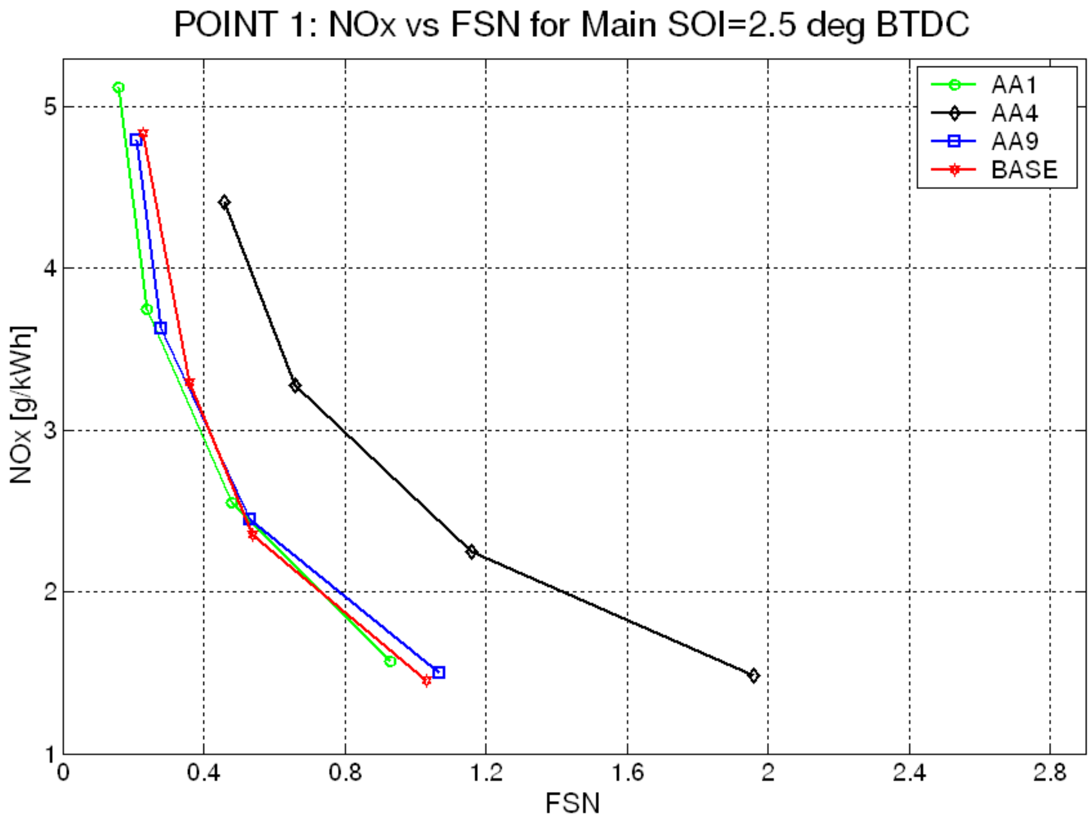

Figure A.4 Test data for point-1 start of main injection = 2.5˚ BTDC

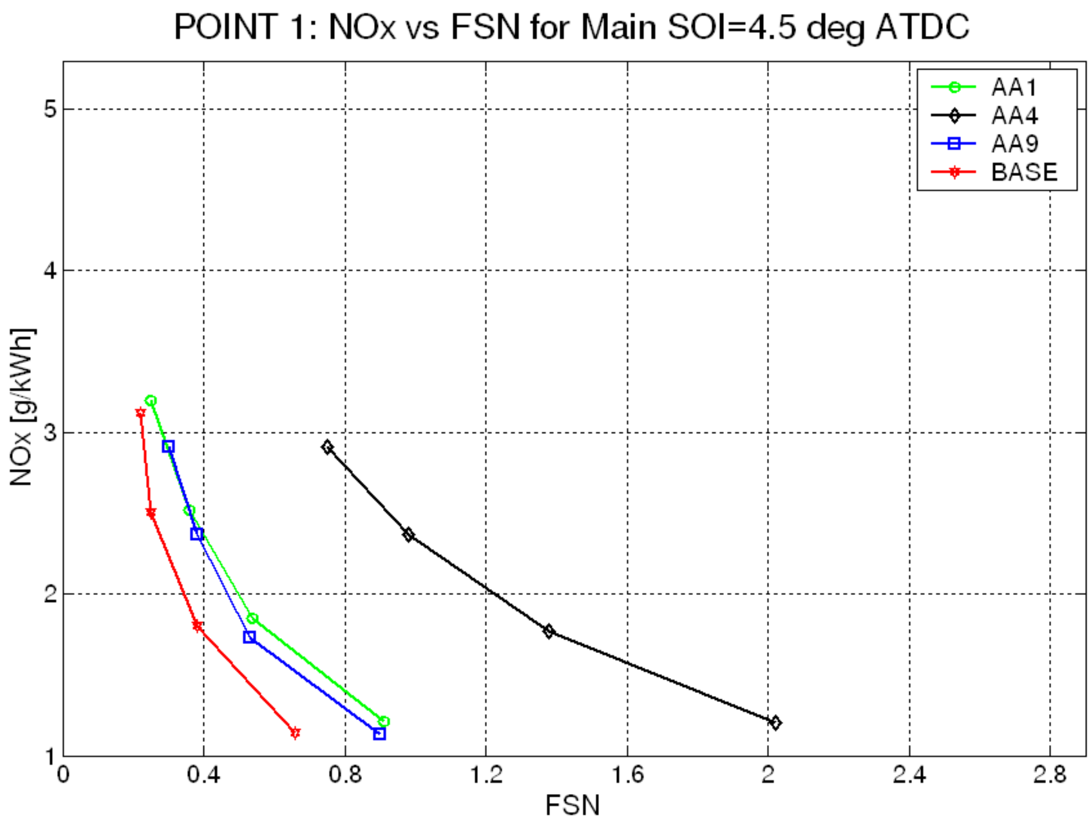

Figure A.5 Test data for point-1 start of main injection = 4.5˚ ATDC

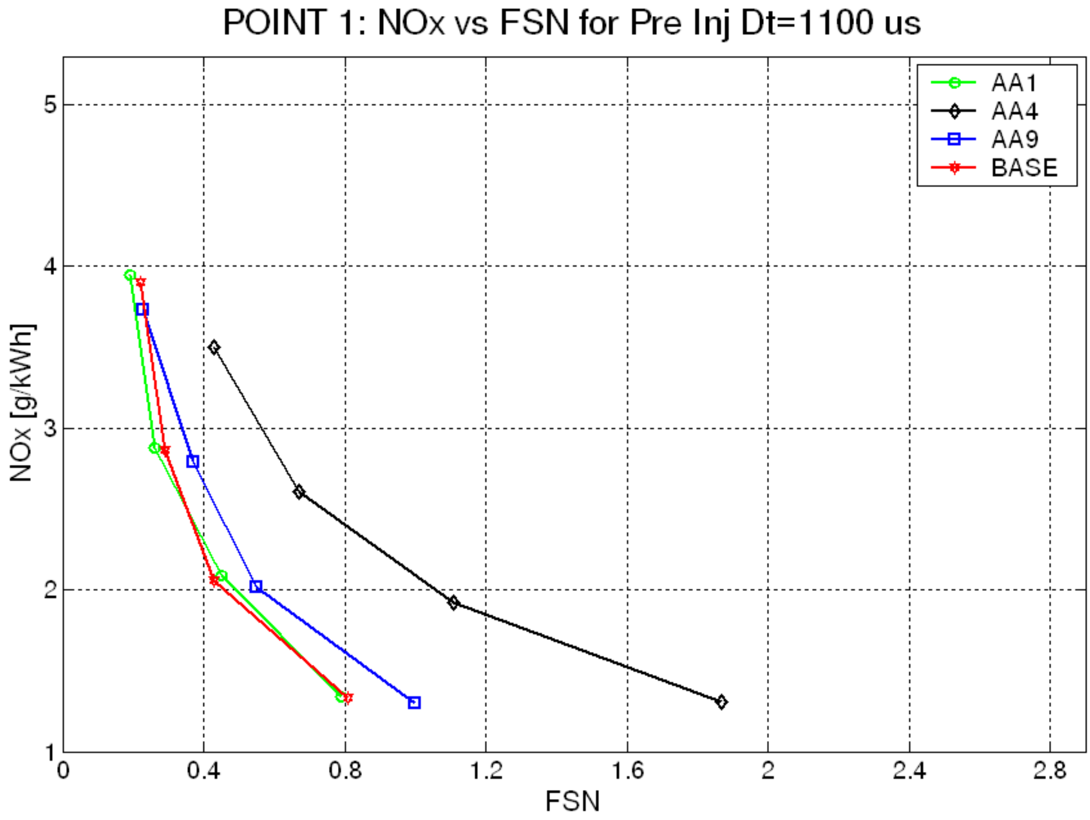

Figure A.6 Test data for point-1 pre injection separation = 1100 us

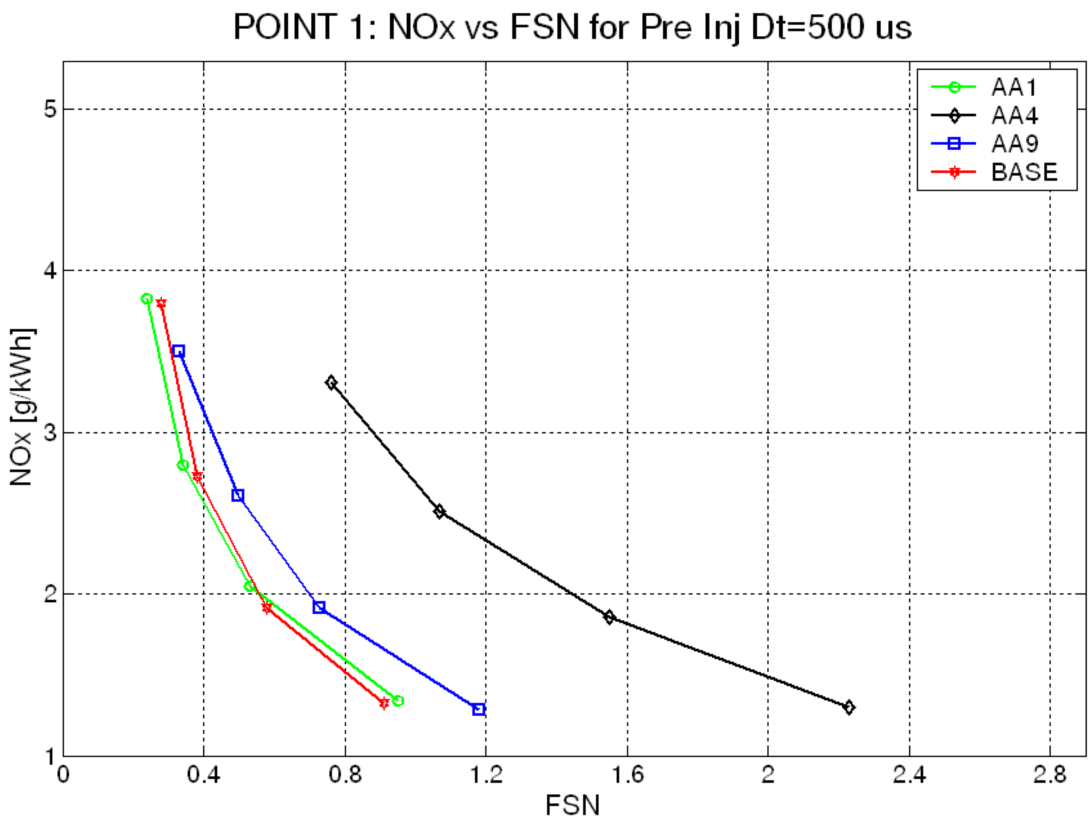

Figure A.7 Test data for point-1 pre injection separation = 500 us

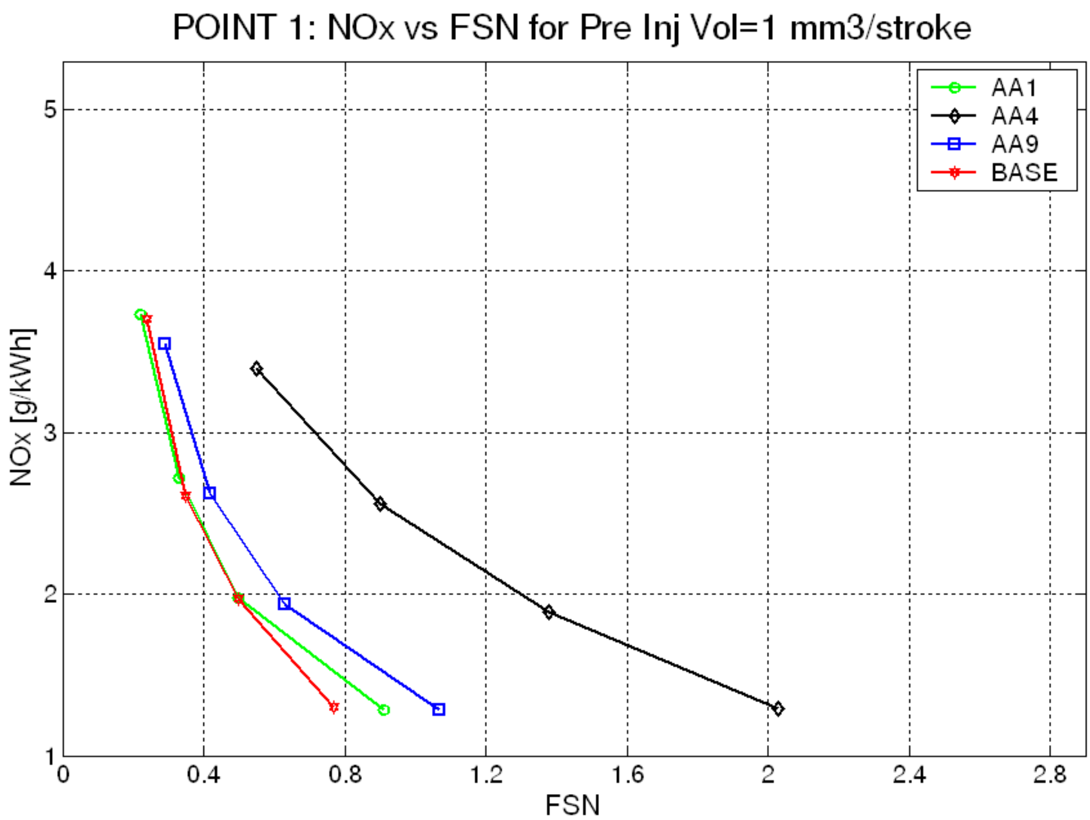

Figure A.8 Test data for point-1 pre injection amount = 1 mm^3/stroke

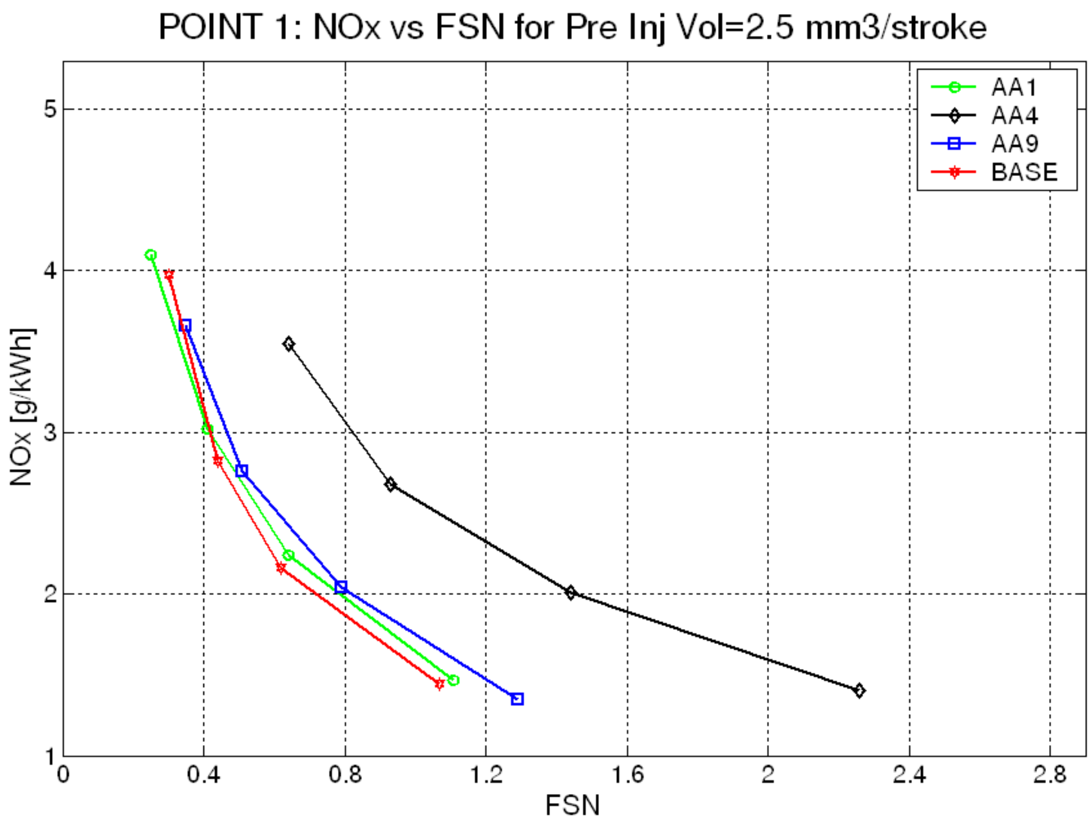

Figure A.9 Test data for point-1 pre injection amount = 2.5 mm^3/stroke

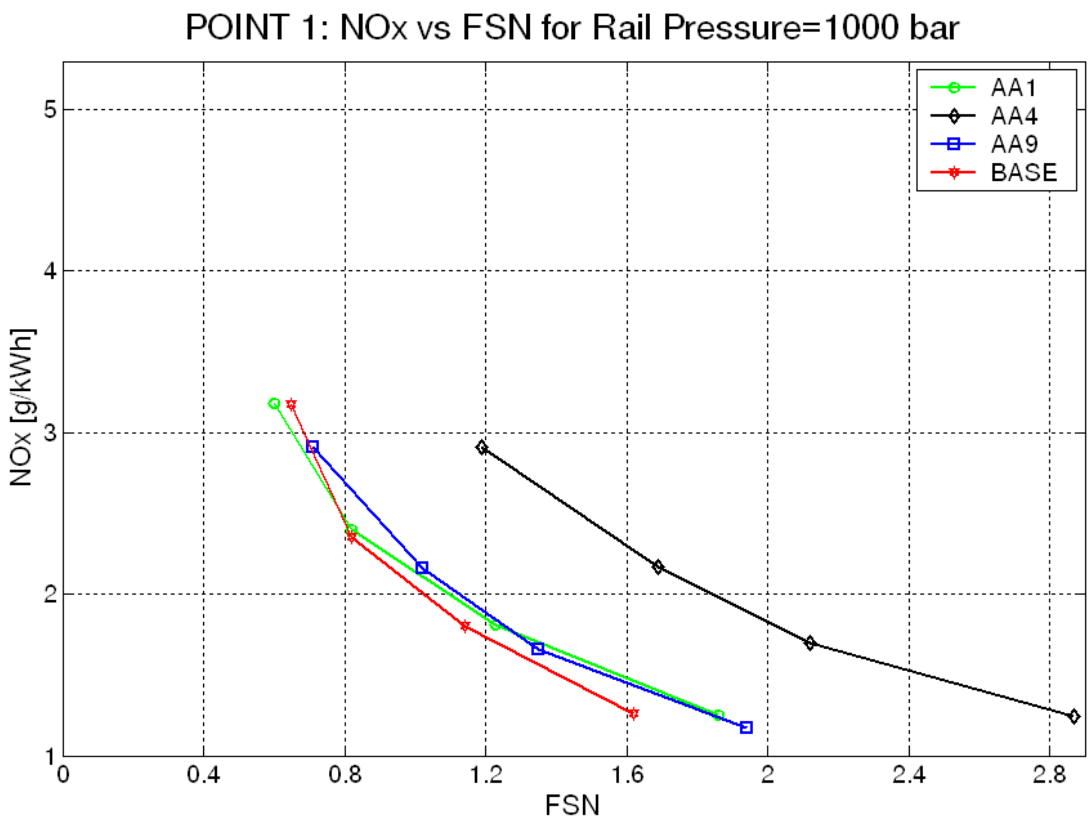

Figure A.10 Test data for point-1 rail pressure = 1000 bar

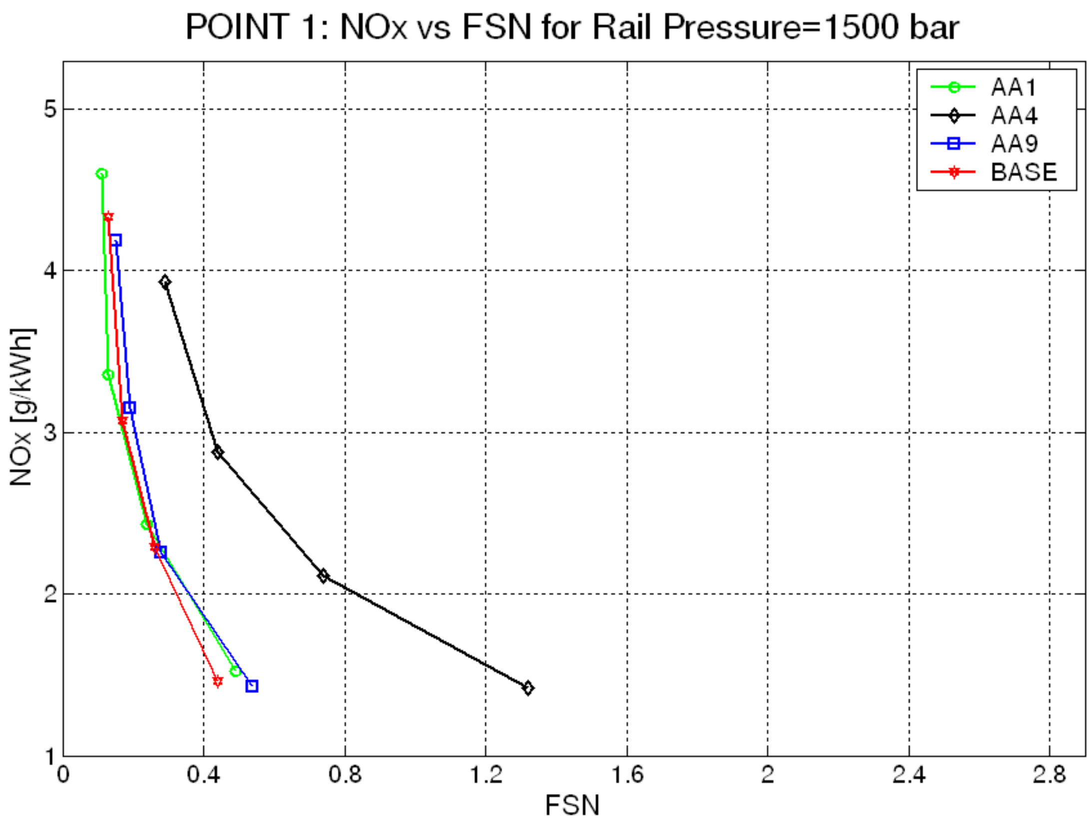

Figure A.11 Test data for point-1 rail pressure = 1500 bar

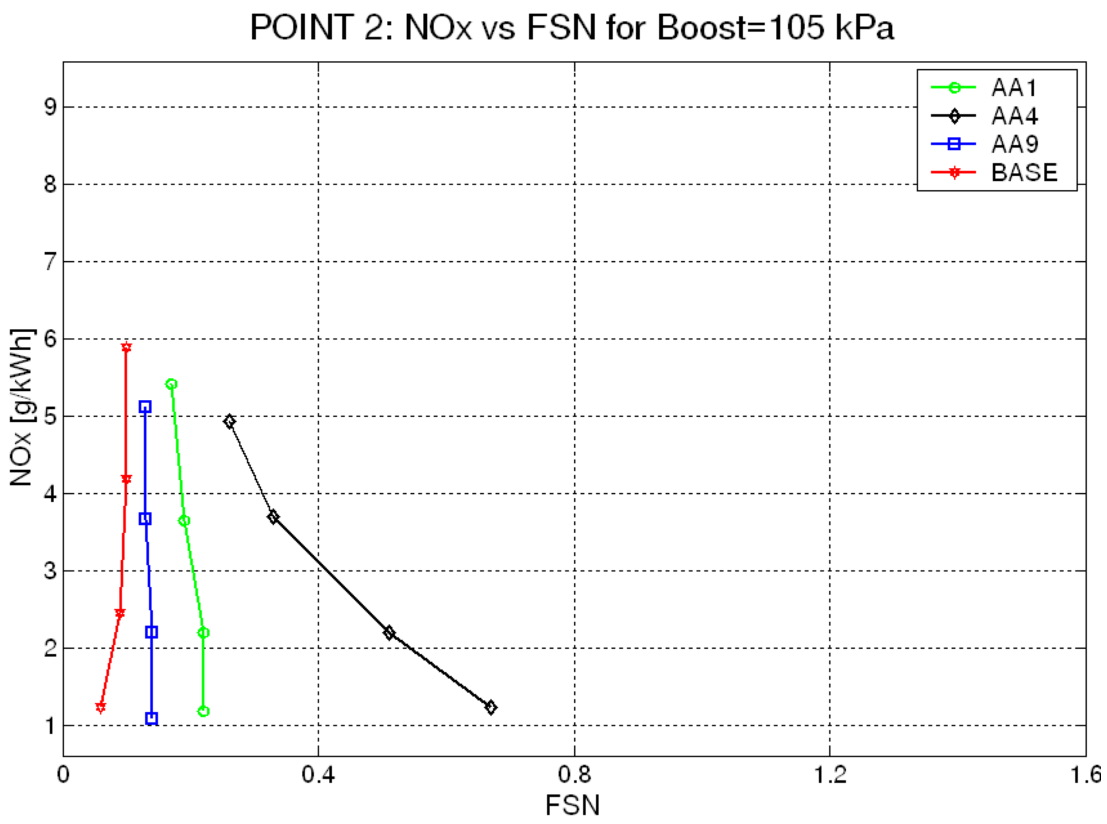

Figure A.12 Test data for point-2 boost pressure = 105 kPa

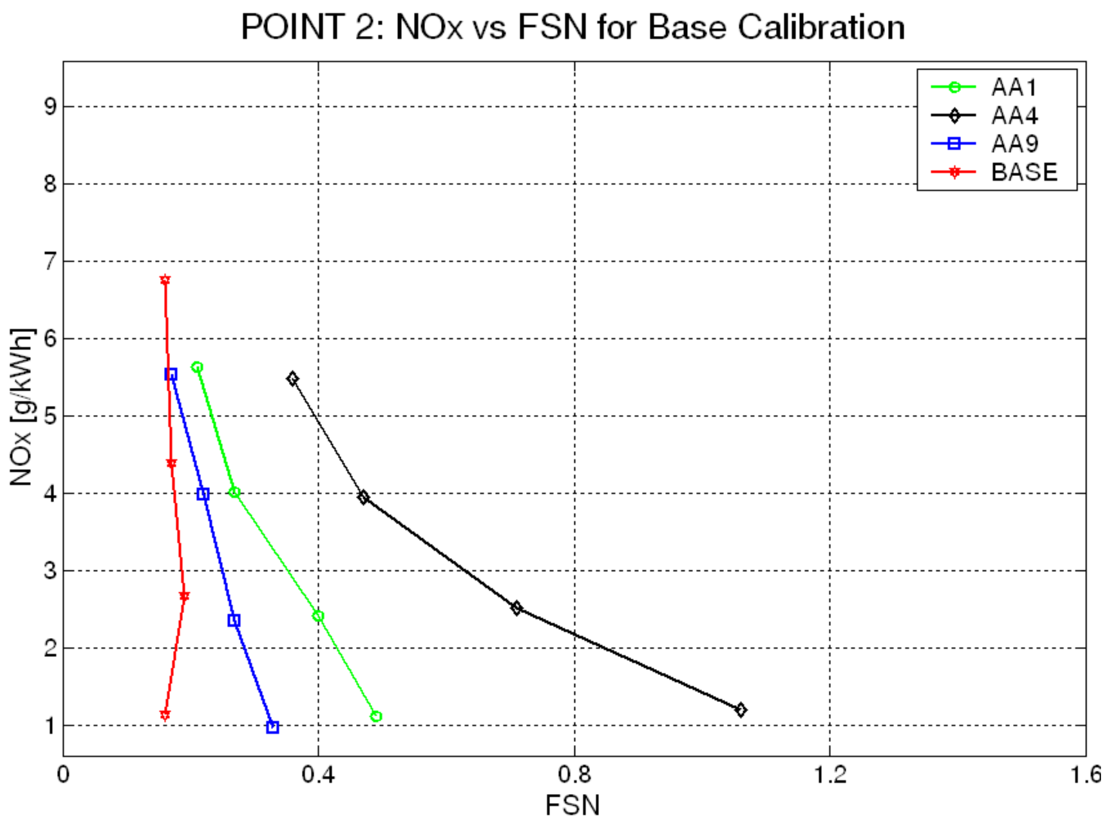

Figure A.13 Test data for point-2 base calibration

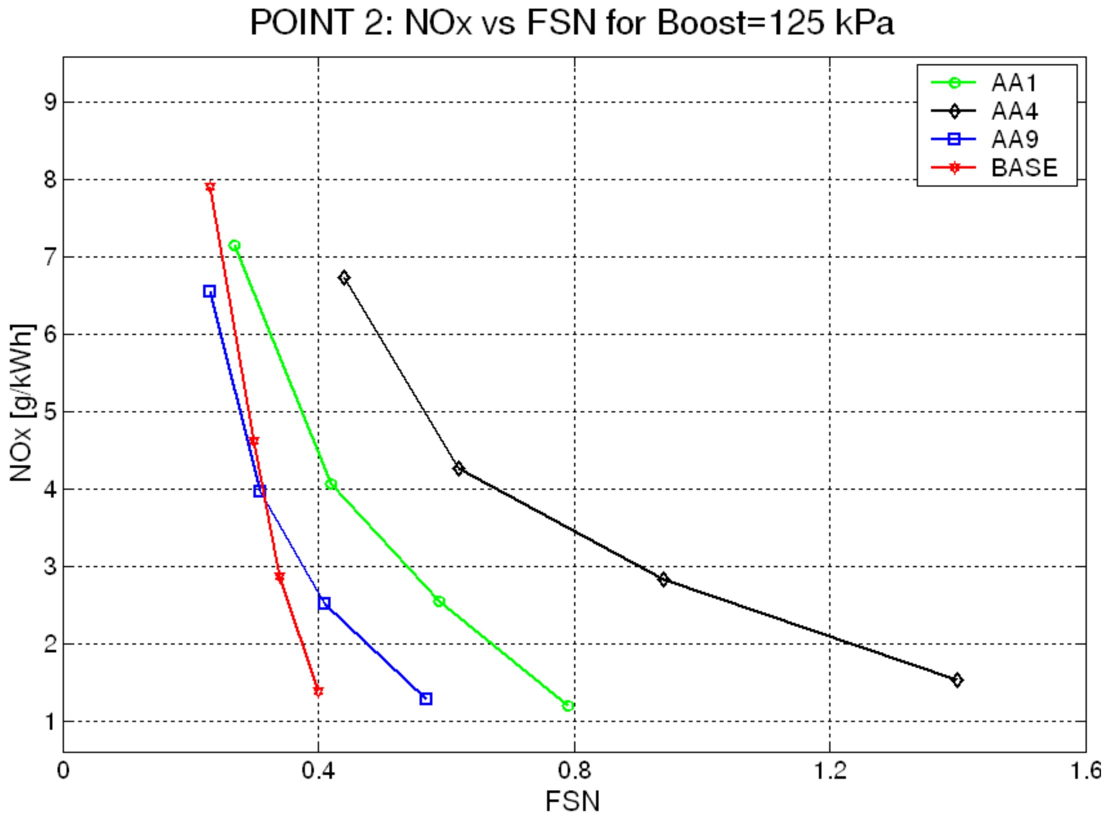

Figure A.14 Test data for point-2 boost pressure = 125 kPa

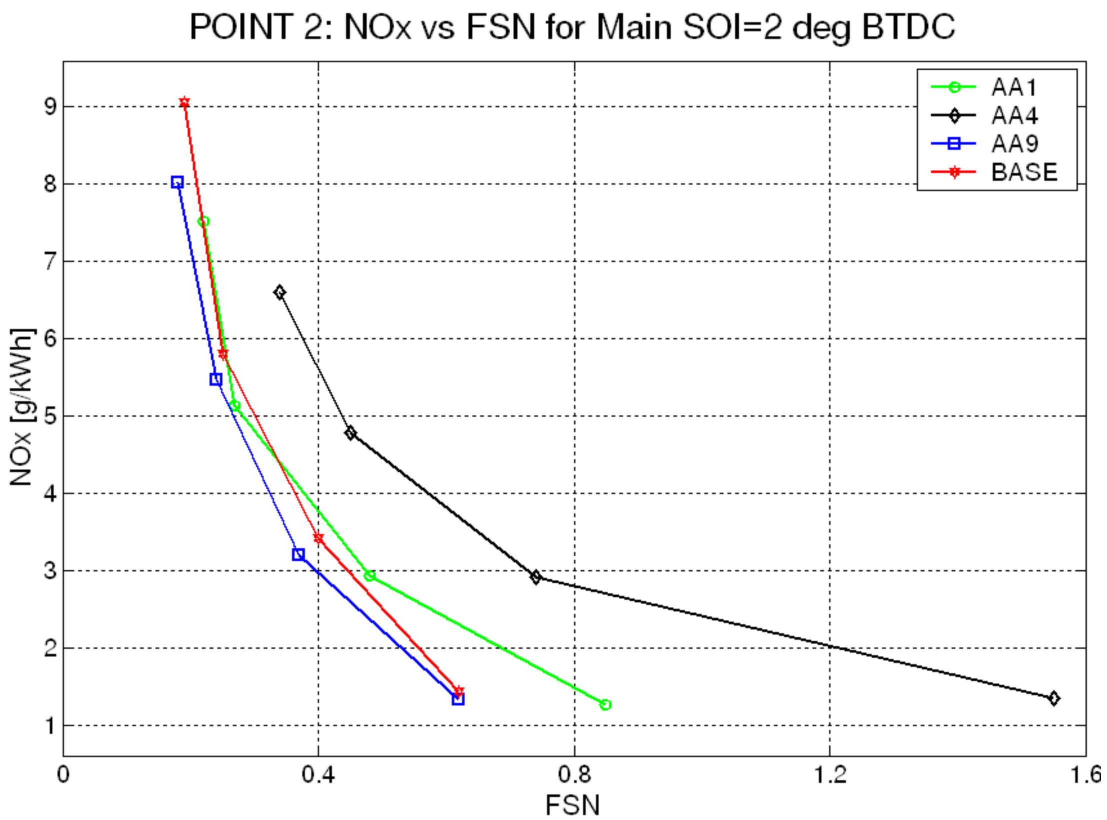

Figure A.15 Test data for point-2 start of main injection = 2° BTDC

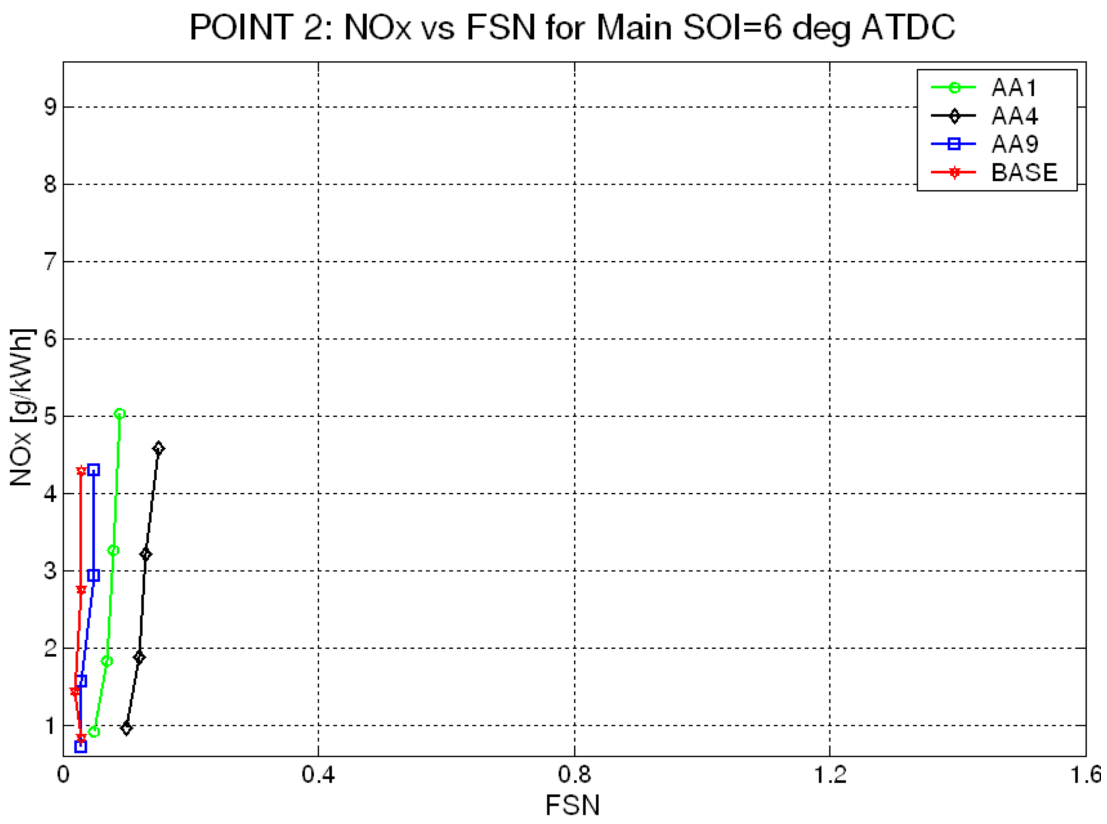

Figure A.16 Test data for point-2 start of main injection = 6° ATDC

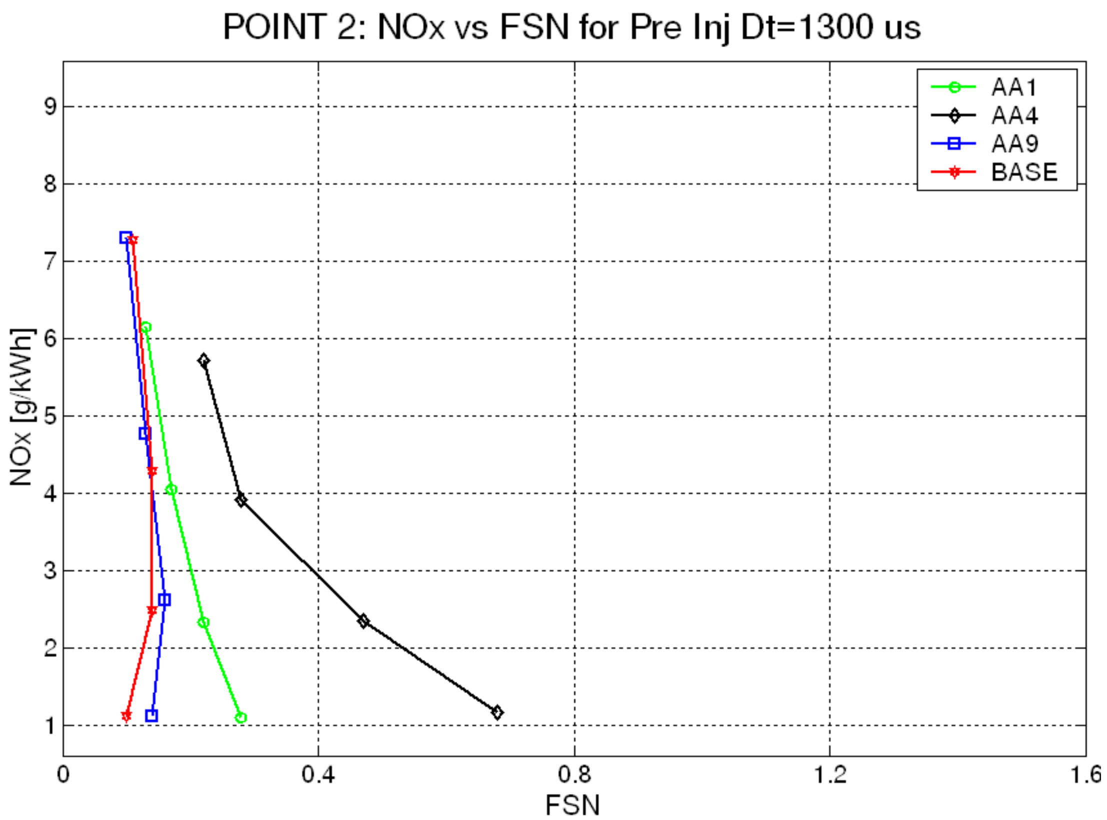

Figure A.17 Test data for point-2 pre injection separation = 1300 us

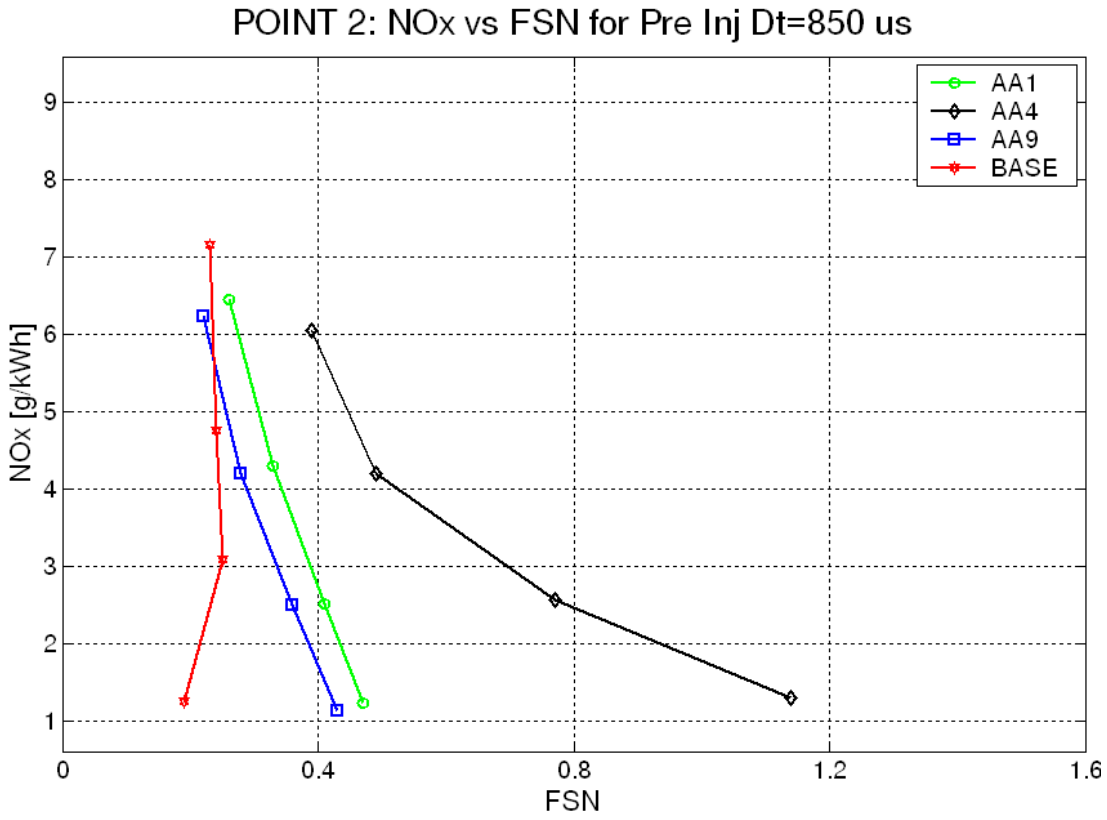

Figure A.18 Test data for point-2 pre injection separation = 850 us

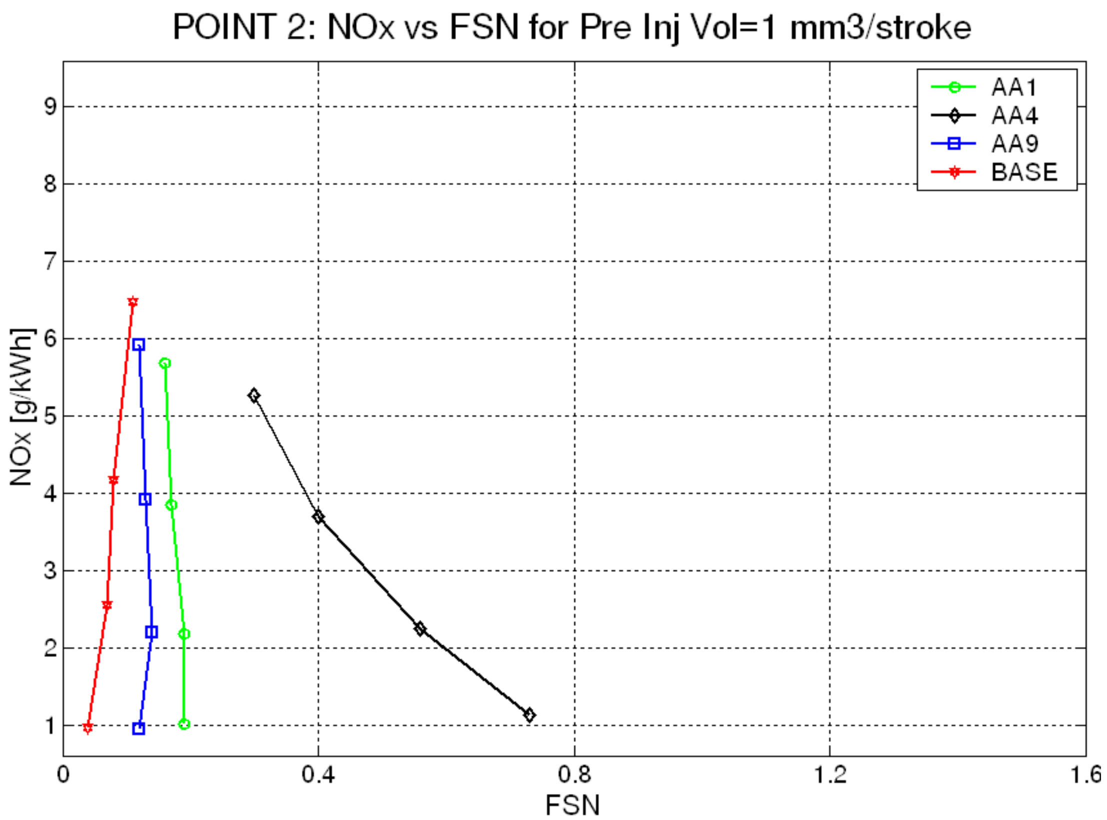

Figure A.19 Test data for point-2 pre injection amount = 1 mm^3/stroke

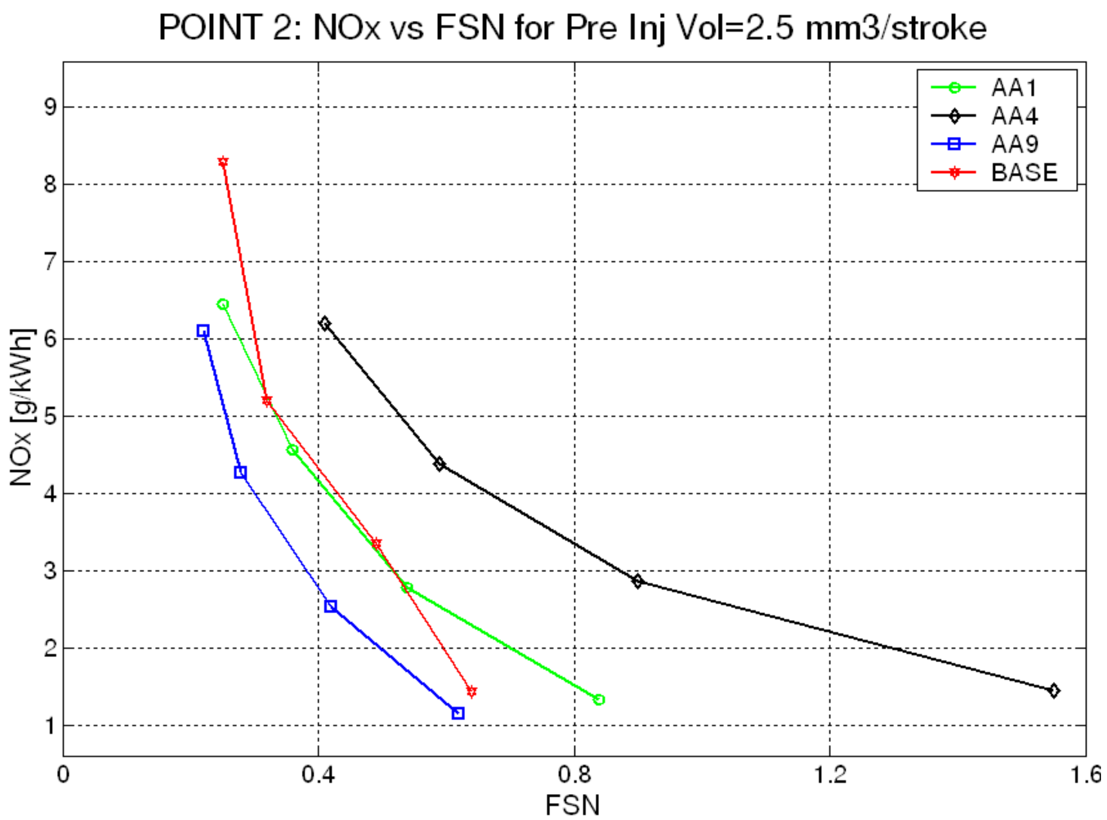

Figure A.20 Test data for point-2 pre injection amount = 2.5 mm^3/stroke

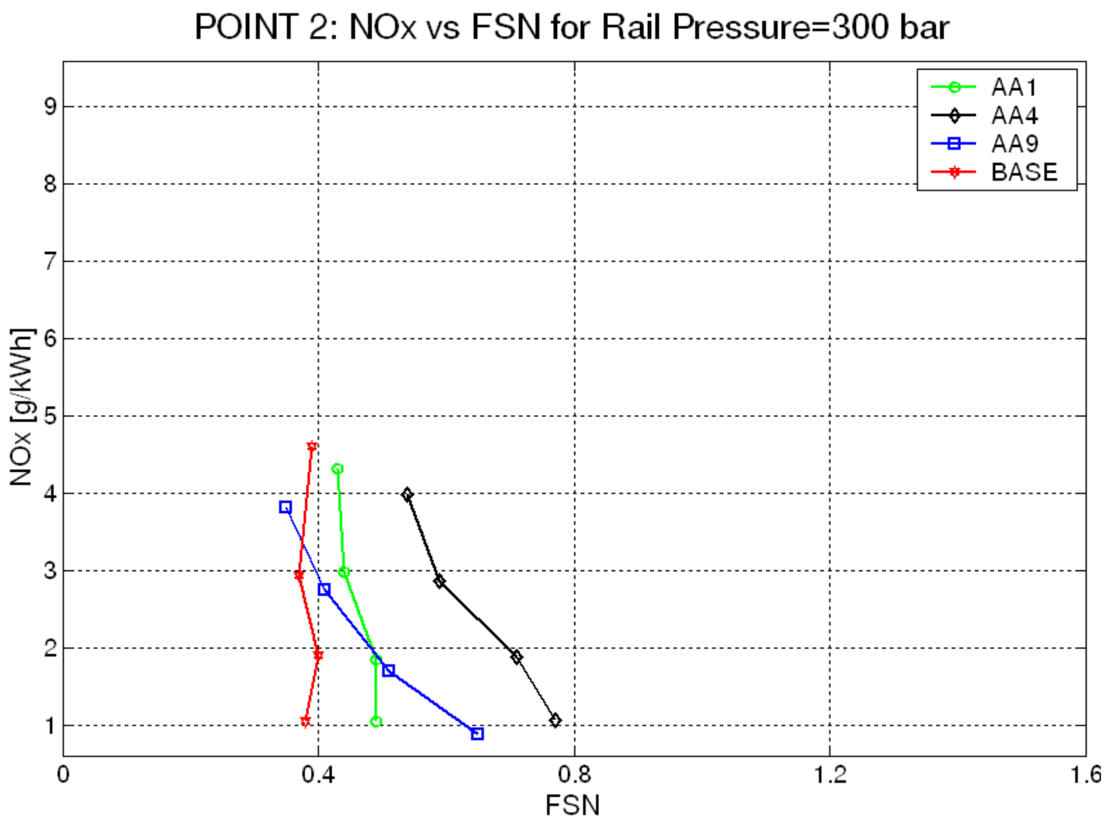

Figure A.21 Test data for point-2 rail pressure = 300 bar

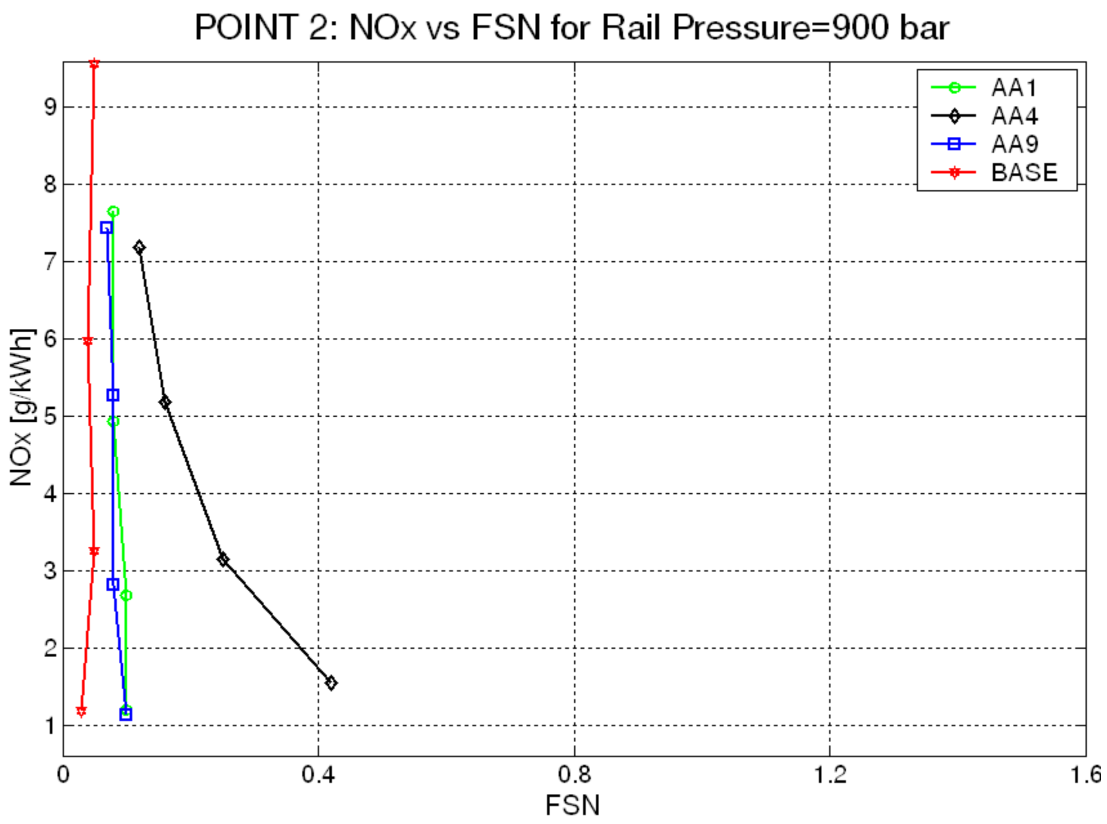

Figure A.22 Test data for point-2 rail pressure = 900 bar

BIOGRAPHY

Hasan Alper BOZKURT was born in Niğde in 1981. After graduation from Ankara Science High School, he entered in Mechanical Engineering Department of Middle East Technical University where he received his Bachelor in Science degree in 2004. He involved in the new 5-cylinder diesel engine development project at Ford Otomotiv Sanayii A.Ş. in Turkey where he has been working since 2004.