İSTANBUL TECHNICAL UNIVERSITY ★ INSTITUTE OF SCIENCE AND TECHNOLOGY

EFFECTS OF EXHAUST GAS RECIRCULATION AND FUEL INJECTION STRATEGY ON ENGINE EMISSIONS

M.Sc. Thesis by
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Department : Mechanical Engineering

Programme : Automotive

JANUARY 2010

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Date of submission : **16 December 2009**
Date of defence examination: **28 January 2010**

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JANUARY 2010

İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

EGZOS GAZLARI RESİRKÜLASYONU VE YAKIT PÜSKÜRTME STRATEJİSİNİN MOTOR EMİSYONLARINA ETKİSİ

YÜKSEK LİSANS TEZİ

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Tezin Enstitüye Verildiği Tarih : 16 Aralık 2009
Tezin Savunulduğu Tarih : 28 Ocak 2010

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OCAK 2010

FOREWORD

I would like to express my deep appreciation and thanks to my advisor Prof. Dr. Metin ERGENEMAN who made this thesis possible. This work could not have been completed without his support and invaluable advice.

This work is supported by the test facility of Ford Otomotiv A.S. I would like to thank my supervisor Ahmet ERGAN for his great support in implementation of the tests and my colleague Kazi ADIL for his time and support during the measurements.

I would also like to thank 'The Scientific and Technological Research Council of Turkey (TUBITAK)' for the financial support they provided during my graduate life.

Last but not least, I would like to thank my family for their encouragement and assistance over my life.

December 2009

Aydın Bayraktar

Mechanical Engineer

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ABBREVIATIONS

ATDC : After Top Dead Center
BDC : Bottom Dead Center
BTDC : Before Top Dead Center
CLD : Chemiluminescence
CO : Carbon Monoxide
ECU : Electronic Control Unit
EGR : Exhaust Gas Recirculation
EUDC : Extra Urban Drive Cycle
FID : Flame Ionization Detector
FSN : Filter Smoke Number
FTP : Federal Test Procedure
HC : Hydrocarbon
IMEP : Indicated Mean Effective Pressure
NEDC : New European Drive Cycle
NO_x : Nitrogen Oxides
LFR : Lean Flame Region
LFOR : Lean Flame out Region
PM : Particulate Matter
PWM : Pulse Width Modulation
RoHR : Rate of Heat Release
SMD : Sauter Mean Diameter
SOF : Soluble Organic Fraction
TDC : Top Dead Center
VGT : Variable Geometry Turbocharger

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LIST OF SYMBOLS

d	: Droplet Diameter
d_n	: Nozzle Diameter
V	: Velocity
σ_d	: Surface Tension
ρ	: Density
W_e	: Weber Number
c_p	: Specific Heat Capacity at Constant Pressure
ϕ	: Equivalence Ratio
ϕ_L	: Equivalence Ratio at Lean Combustion Limit
ΔT	: Temperature Difference
ΔM	: Mass Difference

EFFECTS OF EXHAUST GAS RECIRCULATION AND FUEL INJECTION STRATEGY ON ENGINE EMISSIONS

SUMMARY

Emission regulations require substantial reduction in motor vehicle emissions. Additionally, maintaining the current performance and fuel economy is necessary to meet customer expectations. All of these requirements force diesel engine technology to keep improving. The improvement is possible both by in-cylinder emission reduction methods such as high pressure fuel injection, exhaust gas recirculation, intercooled turbocharging and by aftertreatment technologies such as Lean NO_x Trap (LNT), Selective Catalytic Reduction (SCR) and Diesel Particulate Filters (DPF). It is essential to reduce exhaust emissions from engine itself by combustion improvement rather than using aftertreatment devices.

In this study, two of the in-cylinder emission control systems; exhaust gas recirculation and common rail fuel injection systems are thoroughly investigated and their effects on engine emissions and performance are experimented. The tested variables are the exhaust gas recirculation rate, fuel injection pressure, main fuel injection timing and the pilot injection amount. The tests are conducted at two different test points selected from European emission cycle. These are the points which contribute most to the overall NO_x emissions. Test results showed that the optimized fuel injection strategy and EGR rate bring impressive advances in engine emissions profile. However, further improvements are required to achieve very low NO_x emissions together with low PM emissions. Combining various after-treatment techniques such as LNT, SCR and DPF can further enhance the chance for a future low emission diesel engine. In addition, different emission characteristics at different test points suggest that real world measurement is imperative for determination of the optimum engine control set points.

EGZOS GAZLARI RESİRKÜLASYONU VE YAKIT PÜSKÜRTME STRATEJİSİNİN MOTOR EMİSYONLARINA ETKİSİ

ÖZET

Emisyon regülasyonları motorlu taşıt emisyonlarında ciddi oranda iyileştirme gerektirmektedir. Buna ek olarak mevcut performans ve yakıt ekonomisi karakteristiğinin korunması müşteri beklentilerinin karşılanmasında bir zorunluluktur. Tüm bu gereksinimler dizel motor teknolojisinin gelişimini zorunlu kılmaktadır. Söz konusu iyileştirme, yüksek basınçlı yakıt püskürtme, egzos gazları resirkülasyonu, ön soğutmalı aşırı doldurma gibi silindir içi emisyon azaltma yöntemleri ile ve Azot Oksit Tutucu (LNT), Seçici Katalitik İndirgeme (SCR) ve Dizel Partikül Filtresi (DPF) gibi egzos sisteminde azaltma teknolojileri ile mümkündür. Egzos emisyonlarının egzos sistem teknolojileri kullanılarak azaltılmasından ziyade silindir içerisinde azaltılması daha önemlidir.

Bu çalışmada, silindir içi emisyon kontrol sistemlerinden egzos gazları resirkülasyonu ve yakıt püskürtme sistemleri detaylı olarak incelenmiş ve bu sistemlerin emisyon ve performans üzerindeki etkileri test edilmiştir. Testte kullanılan parametreler; egzos gazları resirkülasyonu oranı, yakıt püskürtme basıncı, ana püskürtme zamanlaması ve ön püskürtme miktarıdır. Testler Avrupa emisyon çevriminden seçilen iki noktada yapılmıştır. Bu iki nokta emisyon çevriminde toplam NO_x emisyonuna en çok katkıda bulunan noktalardır. Elde edilen sonuçlar, yakıt püskürtme stratejisi ve EGR oranının optimizasyonunun emisyon profilinde büyük oranda iyileştirmeler sağlayabileceğini göstermiştir. Ancak, aynı anda çok düşük NO_x ve PM emisyonunun sağlanabilmesi için EGR ve yakıt püskürme stratejisi optimizasyonu ile birlikte çeşitli egzos sisteminde emisyon azaltma tekniklerinin (LNT, SCR, DPF) de kullanılması gerekmektedir. Ayrıca, farklı test noktalarındaki farklı emisyon karakteristikleri, ölçümlerin en uygun motor kontrol parametrelerinin bulunması için bir zorunluluk olduğunu ortaya koymuştur.

1. INTRODUCTION

Combustion engines for road vehicles have to meet a number of requirements. These include emissions, fuel economy as well as drivability and comfort issues. The most significant one for the automotive industry is the emission legislations. Due to the harmful effects of the emissions, nitrogen oxides, NO_x, particulate matter, PM, carbon monoxide CO, and hydrocarbons, HC, government mandates have limited the output of these pollutants.

Diesel engines due to their low fuel consumption recently become very attractive for light duty vehicles and passenger cars. They were known to be harmful compared to spark ignition engines mostly due to their high particulate emissions. Recent engine designs with improved controls have resulted in significant reductions of emissions. However, NO_x emissions are still the key challenge for future.

It is essential to reduce exhaust emissions from the engine itself by combustion improvement rather than using after treatment devices. In-cylinder reduction of NO_x and PM in diesel engines has been achieved by use of high pressure fuel injection systems, exhaust gas recirculation, intercooled turbocharger and so on.

The fuel injection system has a great role in the combustion process and therefore offers a way for combustion improvement. Fuel injection systems are today capable in precise controlling of injection pressure, injection timing as well as are capable of doing multi injections contributing to the emission improvement. One well-known measure to reduce NO_x emission is late injection of fuel into the combustion chamber. However, this is not very effective due to the increase in fuel consumption.

A more effective measure for NO_x reduction is EGR. However, as a rule of thumb, the emissions of PM increase with the increasing rate of EGR. So exact metering of EGR as well as appropriate mixing with well-controlled charge air is required for emissions improvement.

In this study, two of the most important features of today's diesel engines, EGR system and fuel injection system are thoroughly investigated and the effects of exhaust gas recirculation and fuel injection strategy on emissions and engine performance have been reported. Tests are conducted on a 4.4L diesel engine in Ford Otosan Engine Test Facility in Golcuk, Kocaeli.

2. EGR SYSTEMS OVERVIEW

2.1 EGR System Definition

EGR is a commonly used technique for the reduction of nitrogen oxide (NO_x) emissions from internal combustion engines. EGR reduces NO_x emissions essentially by lowering the peak combustion temperature in cylinder. It basically involves the diversion of some of the exhaust gases back to the engine inlet system. During the operation of EGR, part of the exhaust gas is directed back to the intake manifold through an EGR valve as shown in Figure 2.1.

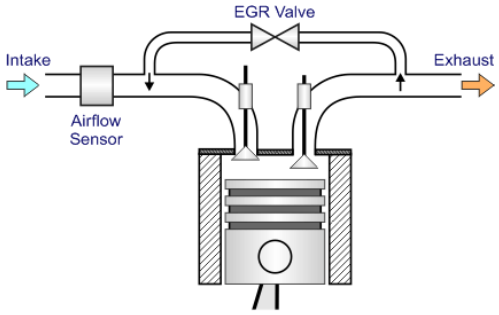

Figure 2.1 : EGR system schematic

2.2 EGR Effects in Diesel Engines

The effects of EGR leading to the lower peak combustion temperature can be broken down into;

- Dilution effect
- Thermal effect
- Chemical effect
- Added mass effect

2.2.1 Dilution effect

NO_x formation is a function of N_2 and O_2 concentrations, combustion temperature, residence time as well as the dissociation of NO and NO_2. This is expressed as:

$$\frac{d(NO)}{dt} = K_1(N_2, O_2) - K_2(NO, NO_2) \quad \textbf{(1.1)}$$

where K_1 and K_2 are reaction rate constants, which are mostly dependent on the combustion temperature [1].

Dilution can be described as the reduction in oxygen concentration in the inlet charge to the engine by means of the addition of inert gases. Therefore, reducing charge O_2 concentration by EGR reduces NO_x formation through reducing one of the four factors given above.

Ladommatos et al. (1996) reported that decreasing O_2 concentration in the inlet charge increases the particulate emissions hence fuel consumption. On the other hand, with reduced O_2 mole fraction, CO emissions increase due to the reduced rate of oxidation at lower flame temperatures associated with the reduction in O_2 mole fraction [2].

Figure 2.2 shows the effect of EGR on O_2 and CO_2 concentration. Introducing 50% EGR in mass reduces inlet O_2 from 21% to 14% whereas it introduces 5% CO_2 [1, 3].

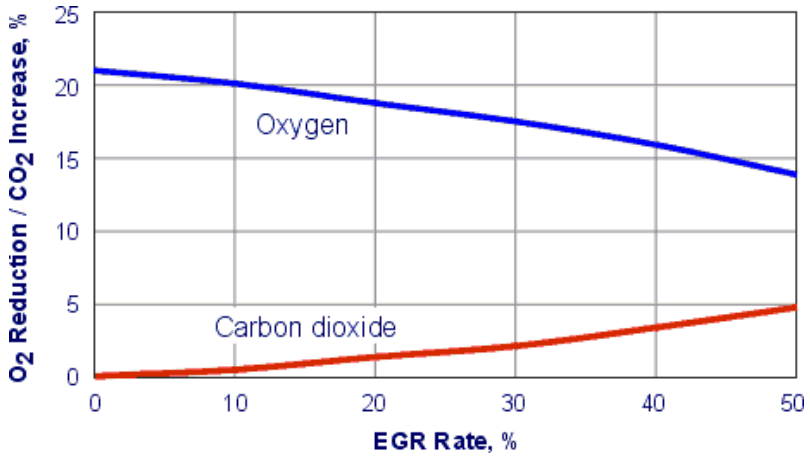

Figure 2.2 : Effect of EGR on inlet charge O_2 concentration

Adding EGR to the intake airflow will not only reduce the O_2 concentration and increase CO_2, it will also affect the properties of the intake charge such as the specific heat capacity and molecular mass, possibly introducing other effects.

Ladommatos et al. (1996) in the series of experiments isolated the dilution effect by an inert diluent having a specific heat capacity matched to that of air. They replaced part of the oxygen in fresh air by a mixture of nitrogen (having a higher specific heat capacity than oxygen) and argon (having a lower specific heat capacity than oxygen). As an example, in one of the tests, they used normal intake air including 23.3% O_2 and 76.7% N_2; in another test, they decreased O_2 concentration to 18.3% while increasing N_2 and Ar to 81.08% and 0.62% respectively. Thus, it was possible to synthesize mixtures of argon and nitrogen, which had the same specific heat capacity as that of air [2].

In this study, as shown in Figure 2.3 the dilution effect i.e. effect of O_2 concentration in the inlet charge on the indicated mean effective pressure (IMEP) and thermal efficiency found to be only minor.

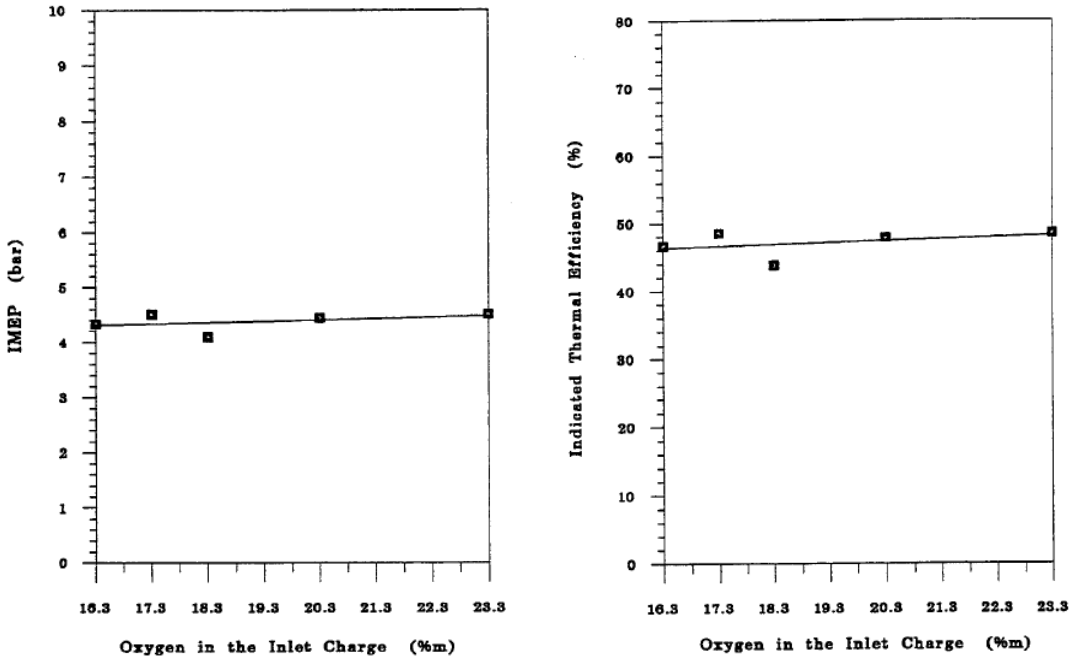

Figure 2.3 : Effect of O_2 concentration on IMEP and thermal efficiency

As shown in Figure 2.4 inlet charge dilution decreased peak cylinder gas temperature and pressures.

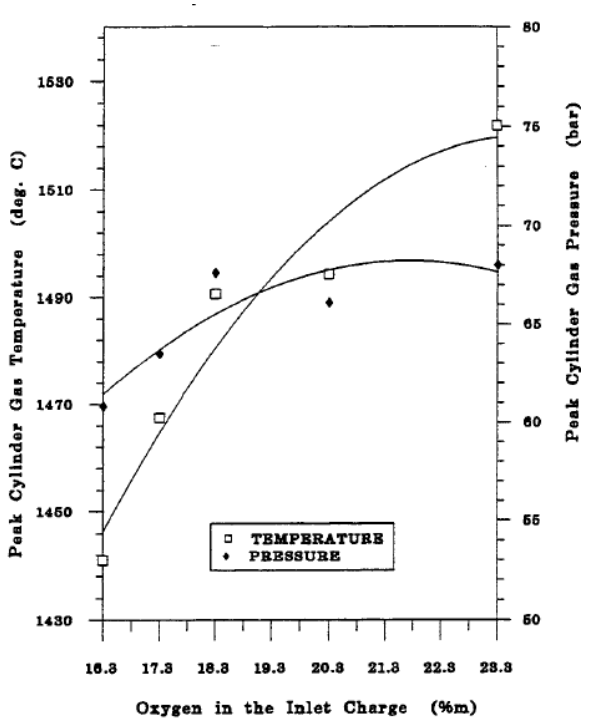

Figure 2.4 : Effect of O_2 concentration on peak cylinder pressure and temperatures

Figure 2.5 shows the effect of charge dilution on NO_x, HC and PM emissions. Charge dilution results in very large reductions in NO_x levels whereas PM and unburnt HC emissions are increased due to reduced oxidation [2].

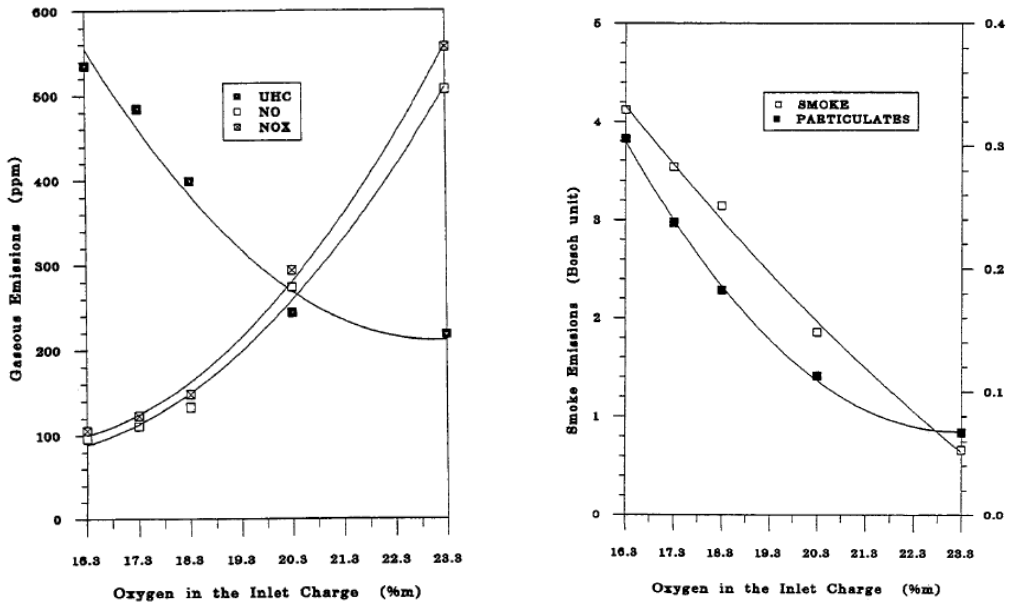

Figure 2.5 : Effect of O_2 concentration on HC, NO_x and PM emissions

It is concluded that the change in ignition delay associated with the inlet charge dilution had only minor effects on engine combustion and emissions while the oxygen reduction was the major factor that influenced both NO_x and PM emissions.

2.2.2 Thermal effect

The recirculated exhaust gas, which contains high concentrations of carbon dioxide and water vapor, increases the total heat capacity of intake charge to cause thermal effect. The increase in heat absorption of the non-reacting gases in EGR is proportional to the product of the increased mass in the cylinder (Δm), the average specific heat capacity at constant pressure (c_p), and the temperature differential (ΔT) between combustion temperature and that of the EGR. It can be expressed as follows:

$$\Delta Q = \Delta m \cdot c_p \cdot (\Delta T) \qquad \textbf{(2.2)}$$

Combustion products consist mostly of CO_2 and H_2O with specific heats higher than those of air. The change in the average specific heat capacity (c_p) in Equation 2.2 resulting from charge dilution is the thermal effect.

At standard temperature and pressure conditions, the specific heat capacity of CO_2, H_2O and N_2 are 36.0, 33.5, and 29.2 kJ/kmolK respectively. Gases of higher c_p can absorb more heat and can be more effective at achieving NO_x reductions. As an example, Figure 2.6 illustrates the effect of using the above-mentioned pure gases as intake air diluents. Cooling EGR would also increase the temperature differential term in Equation 2.2, increase the heat absorbing capacity and further reduce NO_x. [1, 3].

As already mentioned, intake air dilution with EGR can simultaneously introduce the dilution and thermal effects. In order to study the thermal effect of gases with higher specific heat capacities in isolation, the oxygen mass fraction in the intake air needs to be held constant to avoid interference from the dilution effect.

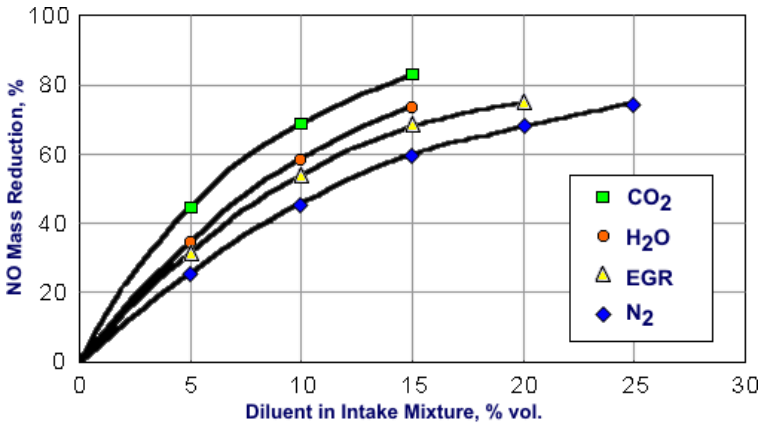

Figure 2.6 : Effect of various diluents on NO reduction

Ladommatos et al. (1996) simulated the thermal effect by replacing the nitrogen in the air with an inert gas, such as helium, that has a heat capacity higher than that of nitrogen. By this way, O_2 concentration kept constant and the total heat capacity of the intake charge is increased isolating thermal effect [4]. Isolated thermal effects of EGR constituents (CO_2 and H_2O) will be discussed in the following sections.

2.2.3 Added-mass effect

If adding a diluent to the intake charge results in an increased mass flow rate, an additional effect is introduced. This added flow has an additional heat capacity due to its mass. This is different from the thermal effect due to any specific heat capacity differences that may exist. The change in mass (Δm) given in Equation 2.2 is the added-mass effect [3].

2.2.4 Chemical effect

The chemical effect is a reduction in the combustion temperature due to chemical reactions with the participation of gases introduced through EGR. For instance, heat is consumed during endothermic reactions such as the dissociation of CO_2 and H_2O.

One way to isolate the chemical effect is to replace a part of nitrogen in the air with argon (having a lower specific heat ratio than nitrogen) and carbon dioxide (having a higher specific heat ratio than nitrogen) as experimented by Ladommatos et al. (1996). Thus, it was possible to maintain a constant average specific heat and oxygen concentration in the intake charge relative to the undiluted case while maintaining constant inlet charge and fuel mass flows. This avoids interference from the thermal and dilution effects [4].

The addition of CO_2 results in reduction in both NO_x and soot emissions. The reason for the reducing soot emissions was suggested to be due to the increase in premixed burning which is the result of the increase in ignition delay. It is also related to the chemical reactivity of CO_2, which accelerates the soot oxidation [4].

Isolation of thermal, chemical and dilution effects are illustrated below.

2.2.5 Isolation of EGR effects in diesel engines

Figure 2.7 illustrates the effect of charge dilution on the oxygen mass fraction and the product of intake charge mass and specific heat capacity (mc_p) as might occur in an engine with EGR. All effects may occur simultaneously making it difficult to ascertain which are most important. The dilution effect only accounts for the reduction in oxygen mass fraction; the thermal effect for differences in average specific heat capacity and the added mass effect for differences in intake charge mass. The chemical effect may also be present.

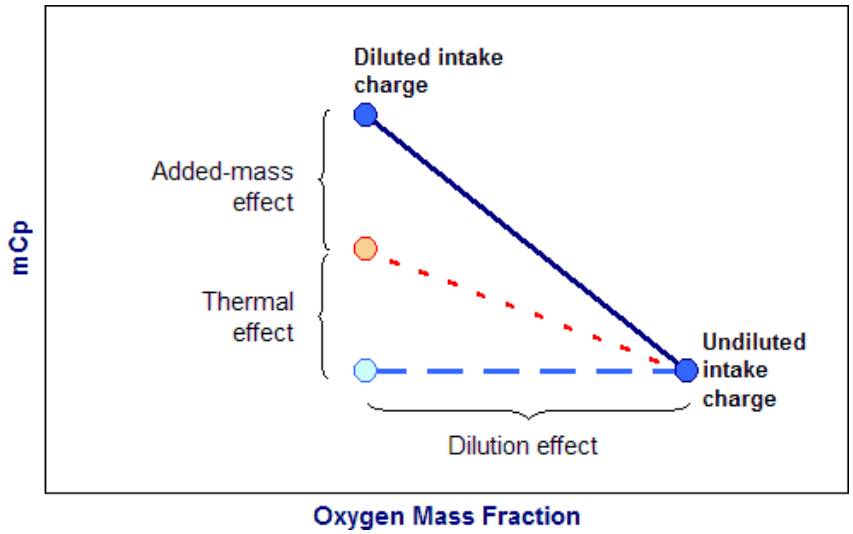

Figure 2.7 : Effect of charge dilution on mc_p

Ladommatos et al. (1996) investigated the effects of EGR constituents CO_2 and H_2O separately [4, 5]

2.2.5.1 Isolated effects of dilution by CO_2

Figure 2.8 shows the relative NO_x reductions with CO_2 dilution for the thermal, chemical and dilution effects in isolation and the total effect. In the tests, intake charge and fuel mass flows were constant so there is no added mass effect consideration. It is apparent that most of the reduction in NO_x is brought by the dilution effect with a small effect due to the chemical effect. Although CO_2 has a

higher specific heat capacity than air (1.24 kJ/kg and 1.16 kJ/kg at 1000K respectively) thermal effect was found to be minor at dilution levels up to 7%. Addition of 7% CO_2 into the air (the amount present with ~50% EGR) increases the specific heat capacity by less than 0.5% that is why the thermal effect is not significant.

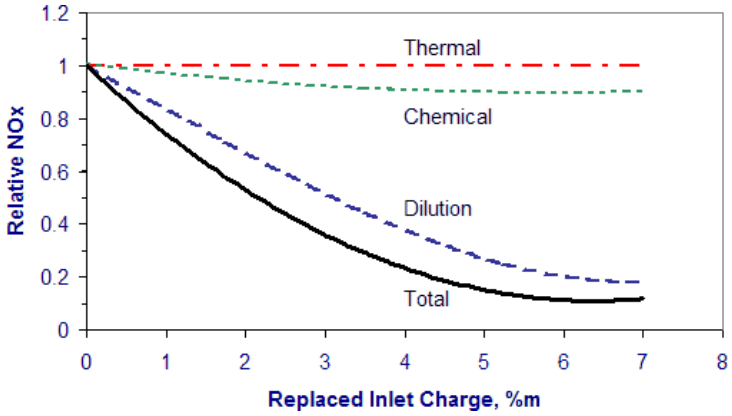

Figure 2.8 : Effect of CO_2 charge dilution on NO_x emissions [3,4]

Figure 2.9 shows the PM emissions with the CO_2 replacing O_2 in the inlet charge. It is clear that the O_2 availability plays a major role in formation and oxidation of PM; hence, the major effect is the dilution effect. Increase of CO_2 results in oxidation, decreasing the PM emission (chemical effect). The thermal effect is insignificant up to 7% replacement as explained above [4].

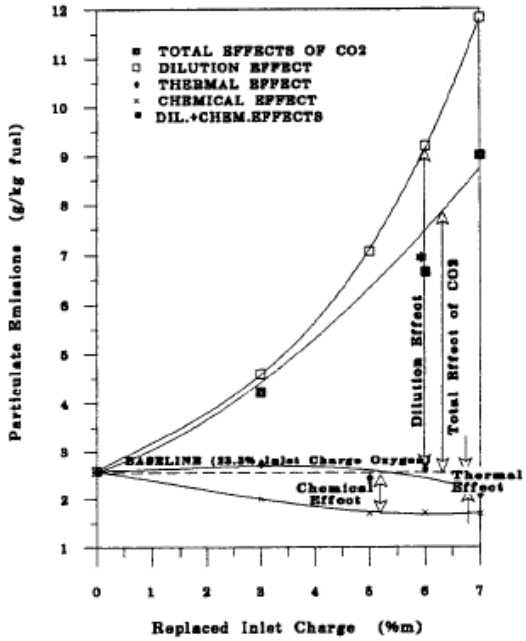

Figure 2.9 : Effect of CO_2 charge dilution on PM emissions

Unburnt HC emissions increase with the increase in CO_2 in the inlet charge. Again, the dilution is the most effective mechanism in increase of HC emissions. When all the effects are summed up, the total effect is not reached as shown in Figure 2.10. This suggests that there is an additional mechanism to the four effects when CO_2 replaced O_2 especially at high rates of replacement.

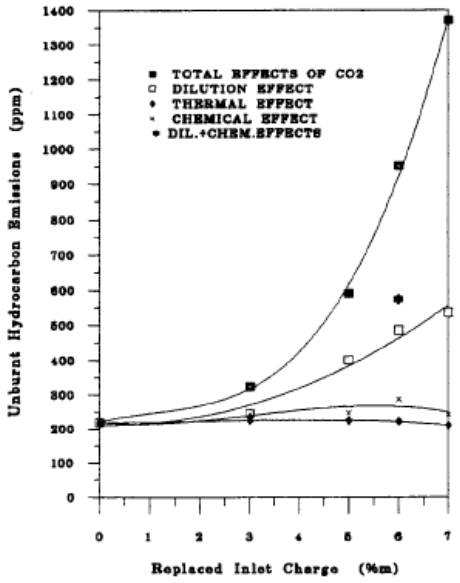

Figure 2.10 : Effect of CO_2 charge dilution on HC emissions

2.2.5.2 Isolated effects of dilution by H_2O

Figure 2.11 shows that CO_2 is less effective in reducing the NO_x emissions when similar amounts of replacements are considered. This is due to the lower specific heat capacity of CO_2 than H_2O.

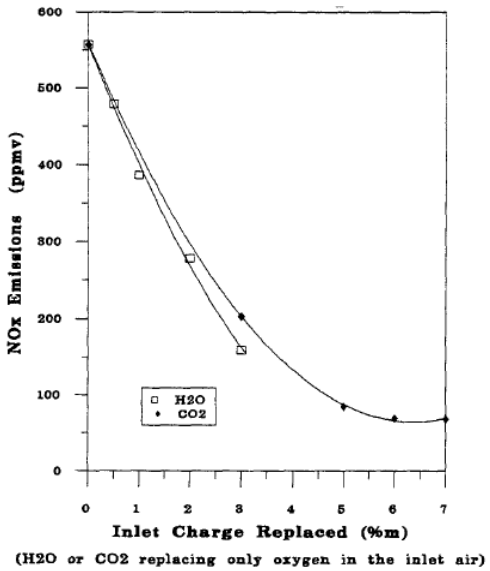

Figure 2.11 : Effect of H_2O and CO_2 charge dilution on NO_x emissions

Figure 2.12 illustrates the effect of dilution with H_2O. Most of the NO_x reduction is by the dilution effect that is similar to CO_2. The chemical effect is negligible to the level of 3% H_2O [5]. Thermal effect of H_2O dilution could not be verified since the experiment was limited due to the high specific heat capacity of water vapor, and the low density of helium, which is used as the replacement gas. A substantial volume of helium was required to replace the necessary mass of nitrogen in the inlet charge. The thermal effect was thought to effect the additional NO_x reduction over that achieved by the dilution effect. H_2O vapor has a considerably higher specific heat capacity than air (2.56 kJ/kg and 1.16 kJ/kg at 1000K respectively). Adding 3% H_2O (the amount present with ~50% EGR) to air increases the specific heat capacity of the mixture by about 3.6% [3, 5].

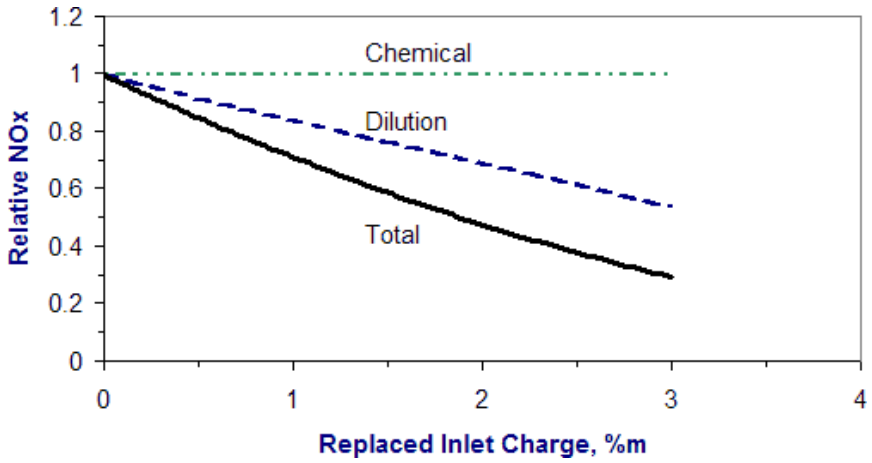

Figure 2.12 : Effect of H_2O charge dilution on NO_x emissions

In Figure 2.13, effect of H_2O in the inlet charge on the PM emissions is shown. The increase in these emissions is mainly due to the dilution and the chemical effects. The reduction in oxygen content of the inlet charge could have lowered the oxidation rate of the particulates.

Figure 2.14 show that 3% H_2O in the inlet charge caused a significant increase in unburnt HC emissions. The dilution effect in this case effects only a small part of the increase. The dissociation of H_2O had no effect on the HC emissions as shown in the figure and the thermal effect could be considered as being negligible [5].

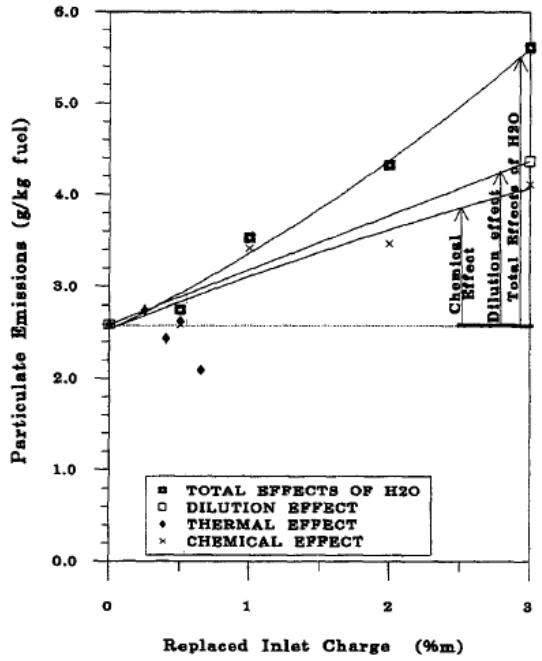

Figure 2.13 : Effect of H_2O charge dilution on PM emissions

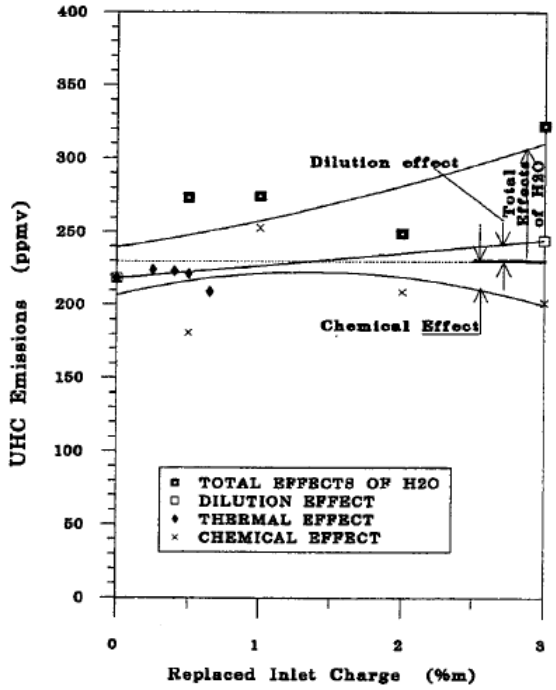

Figure 2.14 : Effect of H_2O charge dilution on HC emissions

Figure 2.15 illustrates the combined effects of CO_2 and H_2O dilution. It can be seen that the dilution effect contributes most to the NO_x reduction whereas thermal and chemical effects has small but measureable effects [3, 6].

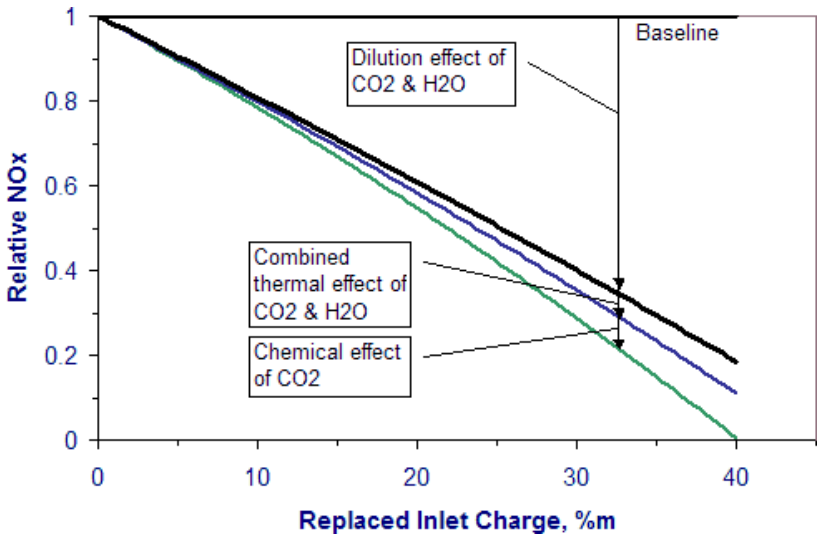

Figure 2.15 : Effect of charge dilution on NOx emissions with CO_2 and H_2O

In Figure 2.16, NO_x reduction due to the added mass effect is shown. In 'oxygen replacement' condition O_2 in the air is replaced by CO_2 so, the NO_x reduction is caused by the combined chemical, dilution and thermal effects. The intake charge and fuel mass flows kept same as the baseline so there is no added mass effect. In the 'added mass' condition, additional amount of CO_2 is introduced to the intake flow that increases the intake charge mass flow by 10%. O_2 concentration is kept same for maintaining the same dilution effect. In the 'added mass' condition, thermal effect (due to the additional c_p of extra mass) and added mass effect is combined. The chemical effect for the 'added mass' case is not known and may differ from the 'oxygen replacement' condition [3].

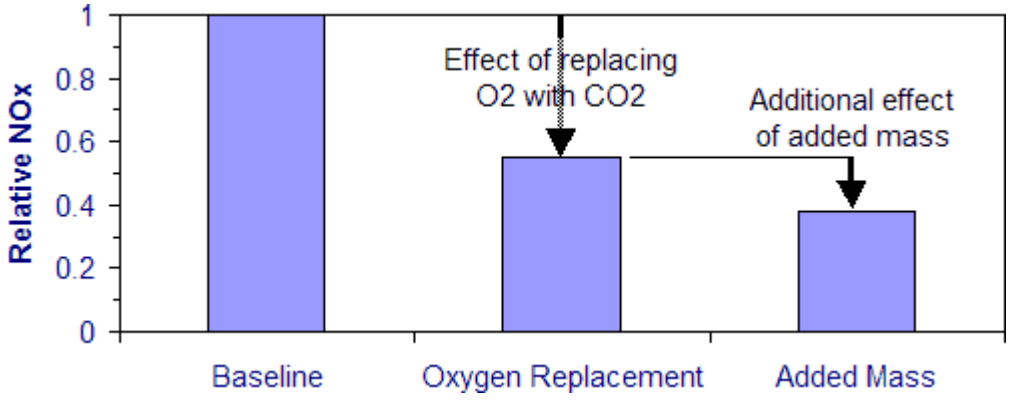

Figure 2.16 : Effect of additional mass on NO_x emissions

2.2.6 Remarks on EGR mechanisms

The reduction in combustion temperature is a result of the changes in the flame zone that result from differences in oxygen concentration relative to the concentration of

the ‘non-oxygen’ gases. Figure 2.17 shows a simplified representation of a diesel diffusion flame with and without EGR. The main reaction zone occurs in a region where the local oxygen and fuel ratio is essentially stoichiometric (φ ~ 1). As some of the oxygen in the inlet charge is displaced with other gases, more inert gas will be present in the combustion zone relative to oxygen and because the amount of fuel added will remain constant, the fuel will have to diffuse and the shape and size of the flame will adjust to maintain the stoichiometric conditions in the flame zone. The added mass of non-reacting gas in the combustion zone absorbs heat and lowers the temperature [3, 6]

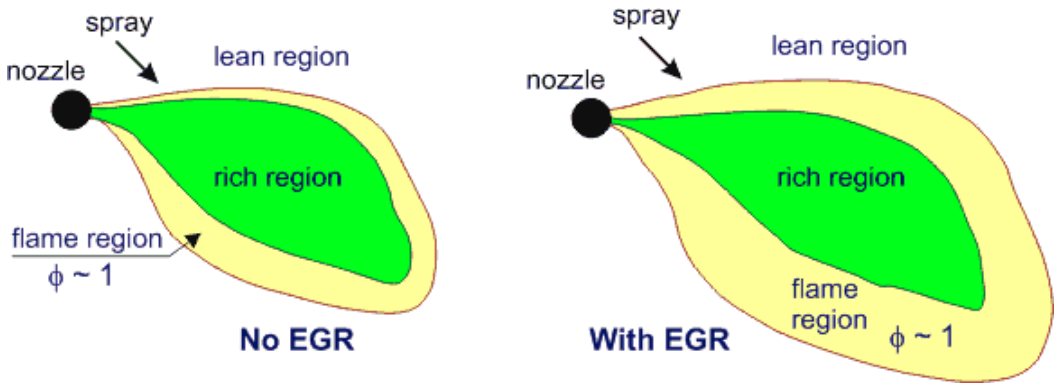

Figure 2.17 : Diesel diffusion flame with and without EGR

2.3 EGR System Configurations

2.3.1 High pressure EGR systems

In high pressure EGR systems, which are mostly used in current production diesel engines, exhaust gas is taken from the exhaust manifold from a separate port prior to the turbocharger turbine. The pressure difference between the exhaust and intake manifolds force the exhaust gas flow to the intake manifold. At low load points where this pressure is not sufficient to flow the gas, the flow may be increased by including a venture in the EGR system as shown in Figure 2.18, an intake throttle or a variable nozzle turbine turbocharger [1, 3].

2.3.2 Low pressure EGR systems

In low pressure EGR systems, the exhaust gas is taken from the exhaust after the after-treatment. EGR flow may be increased by including an intake throttle or an exhaust throttle as illustrated in Figure 2.19.

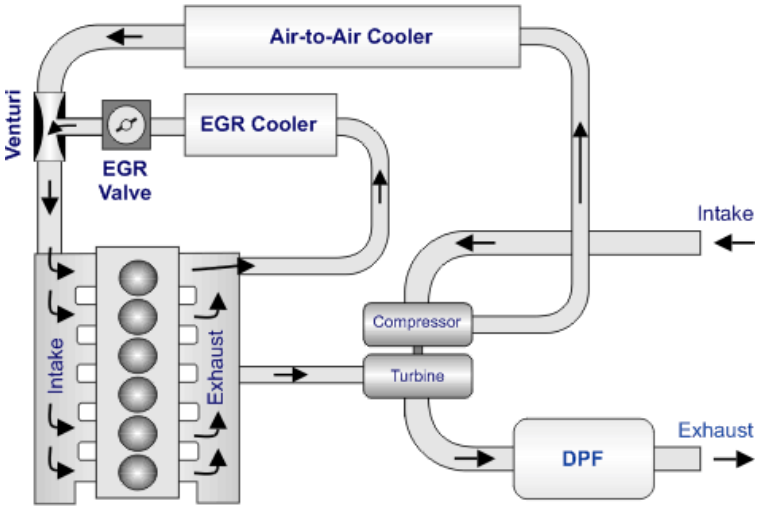

Figure 2.18 : High pressure EGR system schematic

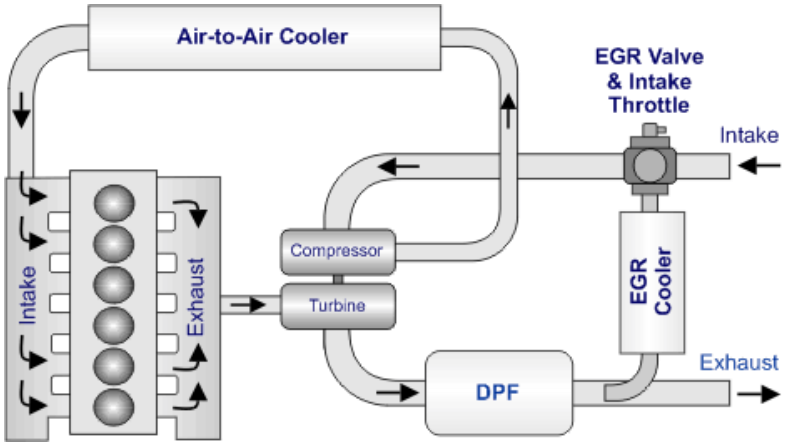

Figure 2.19 : Low pressure EGR system schematic

Low pressure EGR systems are not common today due to the disadvantages listed below [1, 3]

- In these systems, the exhaust gas is flown through the turbocharger compressor and intercooler. This would cause condensation and contamination of corrosive elements in intercooler and turbocharger. In addition, carbonaceous material in the exhaust may impact on the compressor wheel at high speed and it can potentially erode the compressor wheel.
- The operating temperatures of turbocharger and intercooler will increase due to the hot exhaust gas.
- Since the exhaust gas is treated before it is sent to the intake system, it may include debris from the catalyst, which may cause engine damage.

- The transient response (from low load to full load) of the EGR system will be poor due to the large volume effect.
- Diesel particulate filter (DPF) will require more frequent regenerations as the exhaust gas flow though the DPF will increase.
- The exhaust gas includes unburned hydrocarbons, which are re-used when sent back to the cylinder. Burning of HC in catalyst will reduce the benefit that can be get from the HC in the next combustion so the fuel economy benefit will reduce.
- During the regeneration of the diesel particulate filter, the EGR system will need to be disabled to prevent soot particles going into the engine.
- The pumping losses through turbocharger will increase.

Although low pressure EGR system has above disadvantages, it has following advantages:

- EGR distribution between cylinders will be even as the exhaust gas is mixed with air prior to the compressor.
- The feed gas is clean since the gas is taken after after-treatment. Therefore, engine durability can be preserved.
- EGR cooler fouling due to the contaminants will be reduced as the cooler will get the clean feed gas i.e. after the after-treatment. In addition, the system will require smaller EGR cooler due to the already cooled exhaust gas.
- Higher EGR rates are possible, as the flow is not limited by turbo performance.

2.3.3 Hybrid EGR systems

Hybrid EGR systems are the combination of the low pressure and high pressure systems as its name suggests. EGR is tapped from a point prior to the turbine as in a high pressure EGR configuration and injected to pre-compressor as in low pressure EGR configuration. A schematic of a hybrid EGR system is shown in Figure 2.20.

In engines that are equipped with a waste-gated turbocharger exhaust gas can bypass the turbine for the regulation of the boost pressure. Bypassing the exhaust gas

through the turbine reduces the boost pressure as well as the pressure difference between exhaust and intake manifolds above peak torque speed.

Although this configuration includes some of the disadvantages of low pressure EGR configuration, it provides a sufficient pressure difference between the exhaust and intake manifolds to improve flow. Needing a pump or an application of excessive backpressure to drive EGR into the engine is avoided in this configuration.

To improve the EGR flow more, a venture at the point of EGR entry into the inlet system can be implemented as previously shown in high pressure EGR system configuration. This will convert exhaust potential energy to kinetic energy. However, at maximum power and rated speed conditions, the venture may restrict the flow into the intake system. So, the systems including ventures should be accompanied by a bypass and a bypass valve to adjust the airflow at full load conditions [1, 3].

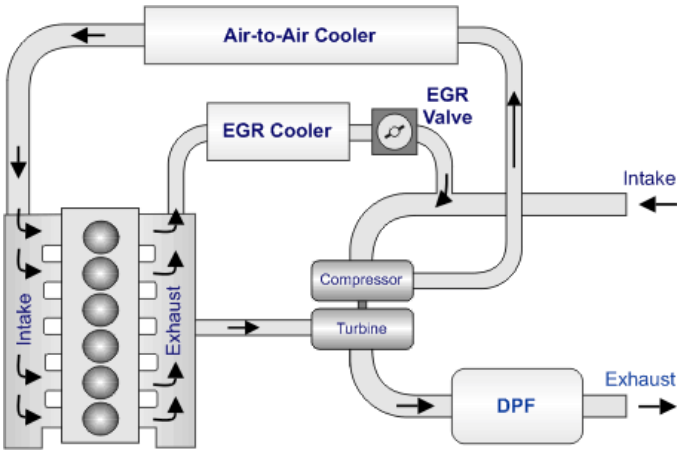

Figure 2.20 : Hybrid EGR system schematic

2.3.4 Fast acting EGR systems

The presence of the large residual volume between the EGR valve and combustion chamber is one of the major drawback common to the EGR configurations discussed so far. When rapid accelerations are required, the residual volume has a negative effect on smoke control. In rapid acceleration conditions, the residual exhaust gas in the pipes between the EGR valve and the combustion chamber will be introduced into the cylinders with the increased fuel amount and this will lead to sluggish acceleration combined with a puff of smoke.

When this problem combines with the turbo lag phenomenon which is defined as the time between injecting the fuel for acceleration and delivering the air into the intake

system by turbocharger, addition to getting the residual exhaust gas in the piping between EGR valve and combustion chamber, the combustion may not occur due to the insufficient fresh air due to the turbo lag. Due to this reason, researchers have designed new configurations to minimize the EGR residual volume effect. This new configuration is named as fast acting EGR systems shown in Figure 2.21 [1, 3].

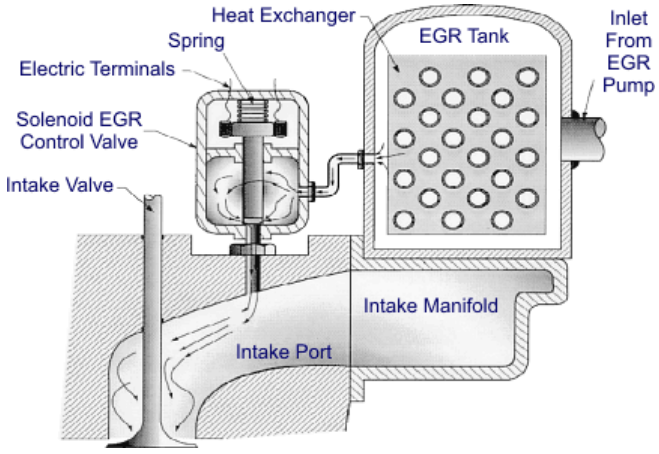

Figure 2.21 : Fast acting EGR system

In this system, the EGR is sourced from a point close to the intake port. The EGR valve is also very close to the intake port thus reducing the volume of the residual gases that cause the sluggish acceleration. Fast acting EGR systems fits the description of high pressure EGR system when the exhaust gas is picked up from a point prior to the turbine. If the gas were sourced from a point after the after-treatment, it would be a low pressure EGR system, which may require a pump to increase its pressure above the intake manifold pressure. This is shown in Figure 2.22 [1].

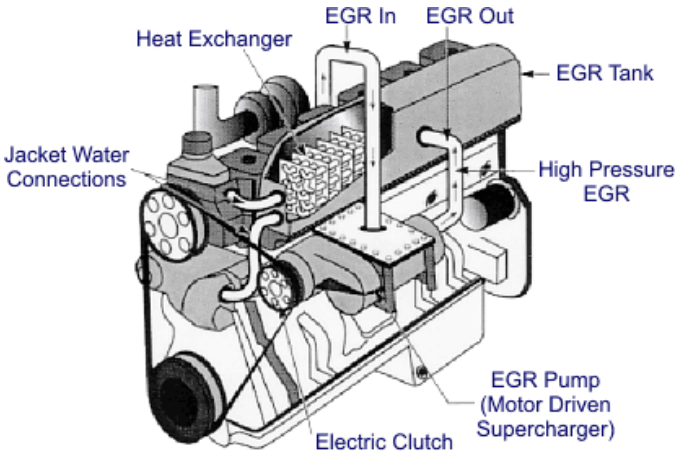

Figure 2.22 : EGR pump to overcome intake manifold pressure

2.3.5 Internal EGR

The use of residual gas for NO_x reduction is referred as internal EGR. Internal EGR is a technology, which uncooled EGR is provided by engine valve actuation. In this system, the exhaust gas is not recirculated with piping but retained in cylinder as residual gas. In the internal EGR system, the exhaust valve reopens during the intake stroke with a modified exhaust valve cam lobe design. This is a second lobe on the valve cam refereed as sub lift lobe. During the intake stroke, sub lift lobe lifts the exhaust valve to allow high pressured exhaust gas to return into the cylinder. Optimization of the system in terms of opening timing and valve lift is very important for controlling the rate of returning EGR gas [3].

Additionally, the pulse of the exhaust caused by the discharging of exhaust gas by other cylinders (in a multi cylinder engine) needs to be considered in the optimization of the system since pulses generated from the blow down process are very important to the creation of the proper pressure differential across the exhaust valve at the time of sub-lifting for EGR.

Internal EGR has benefit in terms of cost and complexity compared to external EGRs. It does not require a variable geometry turbocharger (VGT), venture or EGR pump to make the EGR flow from the exhaust manifold to the intake manifold, as well as EGR cooler and EGR valves. Moreover, the precise control of the EGR is difficult due to contamination of the system and during transients due to the lag associated with long EGR pipes and large volumes, EGR control is difficult. All of those complexities are avoided by an internal EGR. An example of internal EGR is Hino's Pulse EGR system shown in Figure 2.23 comparatively with high pressure EGR system [3].

The disadvantages are that internal EGR provides less reduction in NO_x emissions than cooled EGR; average fuel economy is typically 5% lower than cooled EGR at the same NO_x emission level; and the reduced intake charge density associated with uncooled EGR may cause excessive PM emissions or loss of power at full load. The latter may be avoided by turning internal EGR off at high load, provided this strategy can meet emissions regulations.

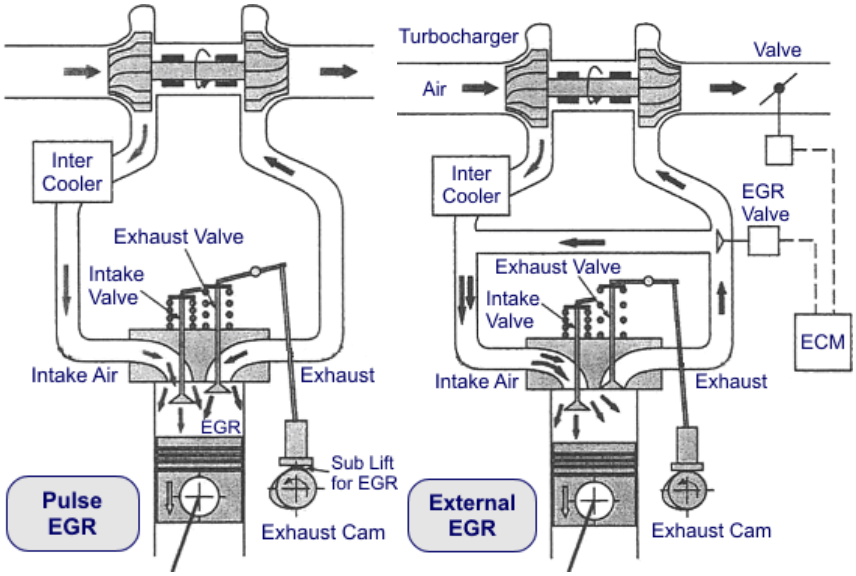

Figure 2.23 : Hino's pulse EGR vs. conventional external EGR System

2.4 EGR System Components

The most common EGR systems in mass production are the high pressure EGR systems. Those systems include an EGR valve and an EGR cooler and in most cases an EGR bypass valve for reduction of HC and CO emissions at cold start conditions.

An overview of the EGR systems with I flow EGR cooler and U flow EGR cooler bypass are shown in Figures 2.24 and 2.25 respectively.

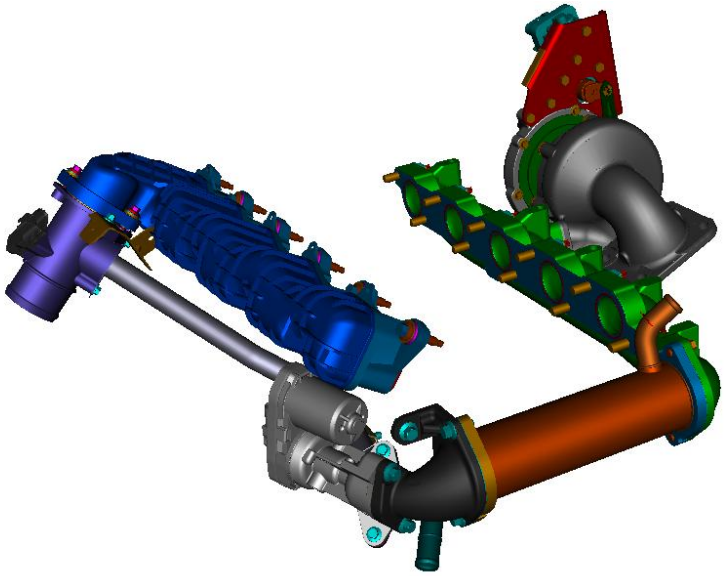

Figure 2.24 : EGR system with I flow EGR cooler and DC motor EGR valve

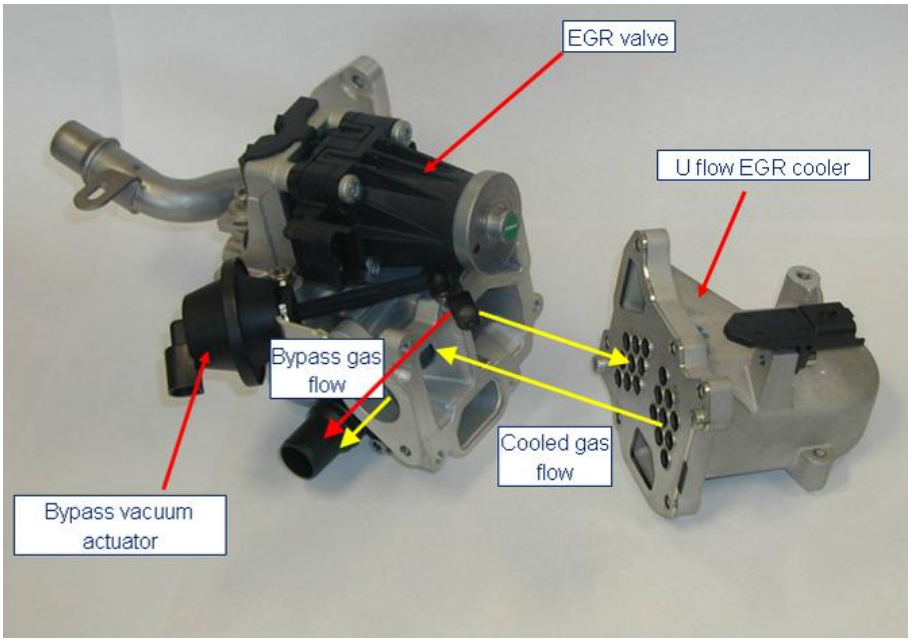

Figure 2.25 : EGR system with U flow cooler and vacuum actuated bypass valve

2.4.1 EGR valves

There are several types of EGR valves, most commonly used are the vacuum actuated and DC motor EGR valves.

2.4.1.1 Vacuum actuated EGR valves

The vacuum actuated EGR valve is used to regulate exhaust gas flow to the intake system by means of a pintle valve attached to the valve diaphragm. A ported vacuum signal and calibrated spring on one side of the diaphragm are balanced against atmospheric pressure acting on the other side of the diaphragm. As the vacuum signal applied to the valve increases, the valve is pulled further from its seat. A schematic of a vacuum actuated system is shown in Figure 2.26.

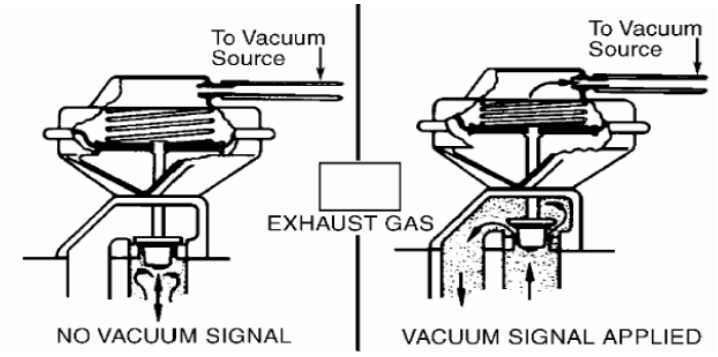

Figure 2.26 : Schematic of a vacuum actuated EGR valve

The key accurate EGR metering is the EGR vacuum modulator assembly, which precisely controls the strength of the applied vacuum signal. Position feed back to the engine control unit is achieved by a lift sensor, which is generally a potentiometer, or a contactless hall sensor. An example of a vacuum actuated EGR valve with potentiometer position feed back is shown in Figure 2.27.

Figure 2.27 : Vacuum actuated EGR valve with potentiometer [7]

2.4.1.2 DC motor EGR valves

The DC motor EGR valves are made up of an electric motor, gearing, a cam or gear system and a position sensor. With the DC motor, the valve can be fully opened or closed in a very short period such as 100 milliseconds. The rotary motion of the DC motor is converted to a linear movement via a cam or a gear mechanism. This movement makes the valve poppet open or close.

The DC motor is activated by a Pulse Width Modulation (PWM) signal, which is a square wave or an on-off step when viewed on a lab scope. The high portion of the waveform is usually the battery voltage or electronic control unit voltage of approximately 5 volts. Thus, the opening of EGR valve is modulated by means of PWM signal generation.

The position feed back to ECU is done by means of a potentiometer or a contactless hall sensor. If it is a potentiometer, the lift of the EGR valve is converted to a resistance value and fed back to the ECU. If the position sensor is a hall sensor, then the magnet at the tip of the valve stem generates a magnetic field depending on the

position of the valve, which is then converted to a voltage value and fed back to the ECU. A section view of a DC motor EGR valve is shown in Figure 2.28.

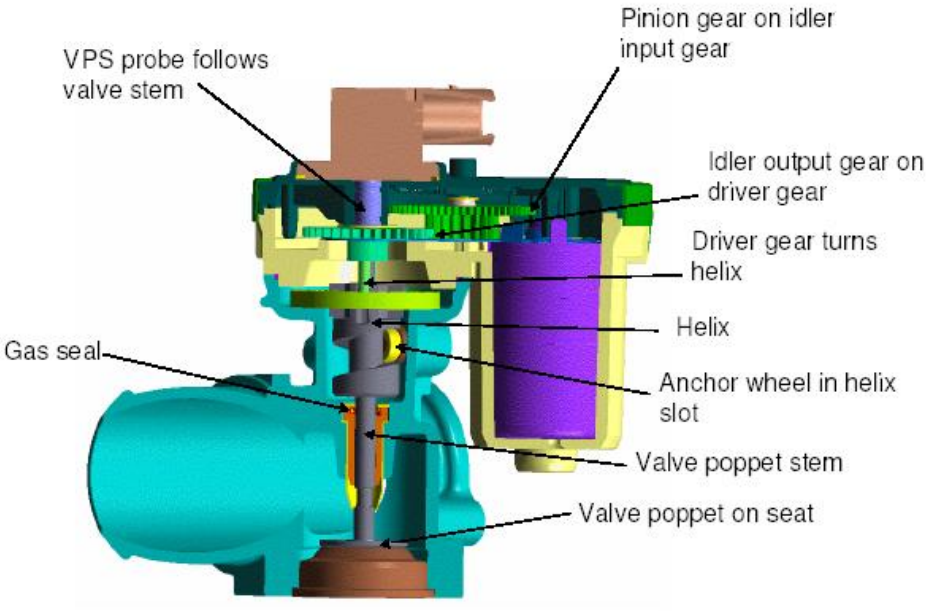

Figure 2.28 : DC motor EGR valve [3]

2.4.2 EGR bypass valve and EGR cooler

EGR bypass valve function is to reduce level of EGR cooling for cold start and low speed-low load conditions to reduce CO and HC emissions. It can also be used as a protective measure for EGR cooler fouling. It is usually driven by a vacuum actuator.

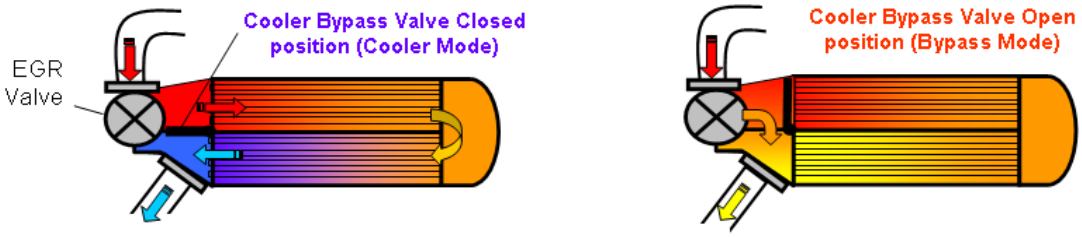

Figure 2.29 : EGR cooler and bypass assembly in cooling mode and bypass mode[3]

EGR cooling reduces engine out NO_x and increases intake charge density, allowing for increased EGR ratio without compromising desired air to fuel ratio.

The EGR cooler includes two circuits: one for the exhaust gas and one for the engine coolant. The exhaust gas is water cooled by either tube in shell type or fin type construction. The heat exchange surfaces are designed to optimize heat exchange, pressure loss and robustness to soot and HC build up. When the soot and HC are

contaminated in the heat exchange surfaces, the heat transfer degrades and pressure loss increases.

EGR coolers are generally stainless steel for corrosion resistance. Aluminum coolers have also been developed to reduce cost and improve the power/volume ratio [3].

2.5 Control of EGR System

To achieve a good NO_x and PM control, EGR flow rate should be precisely controlled. Those control elements include shutting of EGR during acceleration, reducing EGR at high load conditions and shutting EGR off when the A/F ratio is lower than 25:1. Additionally EGR rate should be limited to keep exhaust gas temperatures in the limits for protection of turbocharger components.

By varying the EGR valve position the flow of EGR can be controlled. The EGR mass flow also depends on the pressure difference between exhaust and intake manifolds. There are two control concepts for EGR system: open loop and closed loop control systems.

In open loop control systems, the ECU monitors the engine speed and load and selects the required EGR rate from a look up table and operates the EGR valve accordingly. A scheme of an open loop EGR control is illustrated in Figure 2.30 [1].

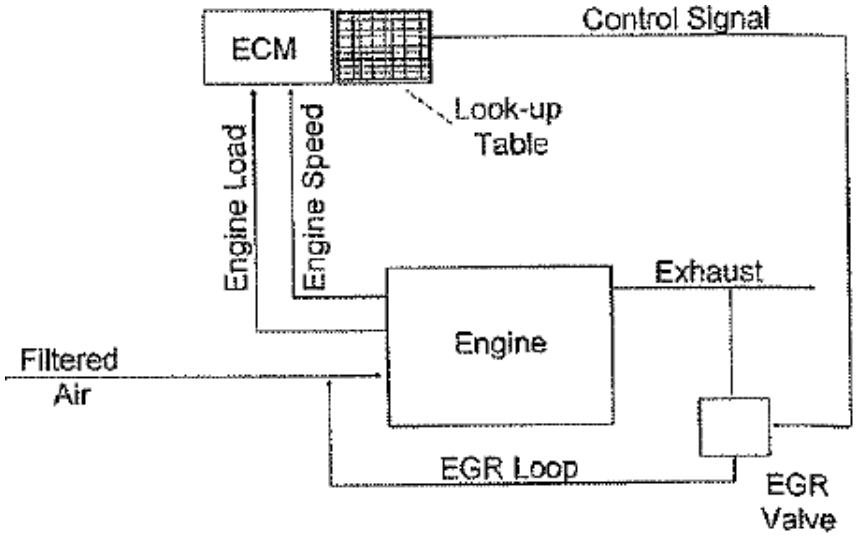

Figure 2.30 : Open loop EGR control

In closed loop control systems, the control layout is similar to that of open loop control system. Additionally, a value, which is usually EGR valve lift or mass airflow, is controlled. This value is compared to the desired value and an adjustment

is made in response to the difference between desired and actual values of the controlled parameter. A schematic of a closed loop EGR control is illustrated in Figure 2.31 [1]. The closed loop control of valve position is appropriate for the transient control whereas mass airflow control is more appropriate for steady state control [3].

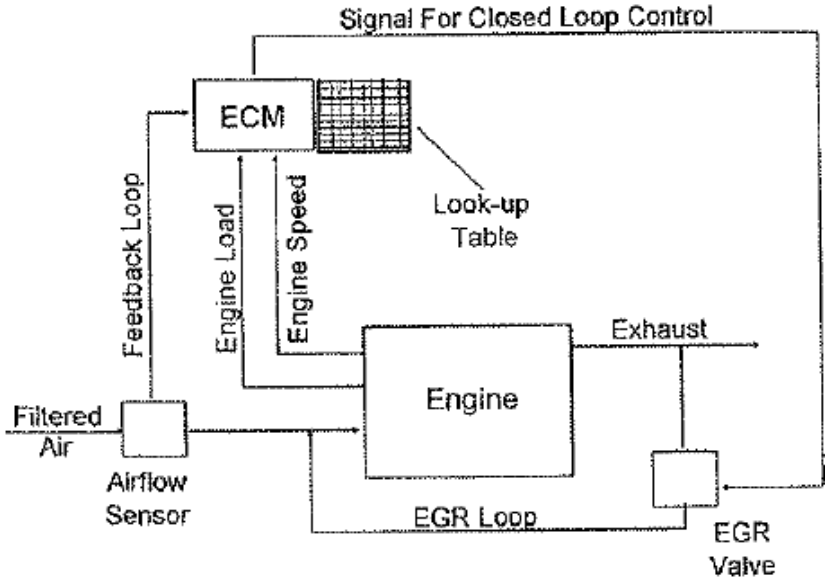

Figure 2.31 : Closed loop EGR control

2.6 Effects of EGR on Combustion and Emissions

Acroumanis et al. (1995) had studied the effects of EGR on combustion and emissions. The research engine was a high speed, four cylinders 1.9L direct injection diesel engine with an EGR system. Their study showed that replacing 50% of intake air with exhaust gas decreased O_2 from 21% to 14% while introducing 5% CO_2 into the cylinders. The emission tests confirmed that for different intake temperatures, increasing the EGR rate leads to reduced NO_x and O_2 levels but increased soot, CO, CO_2 and HC concentrations. With O_2 being replaced, the soot oxidation rate drops and leads to higher concentrations of carbonaceous particulate matter as shown in Figure 2.32 [8].

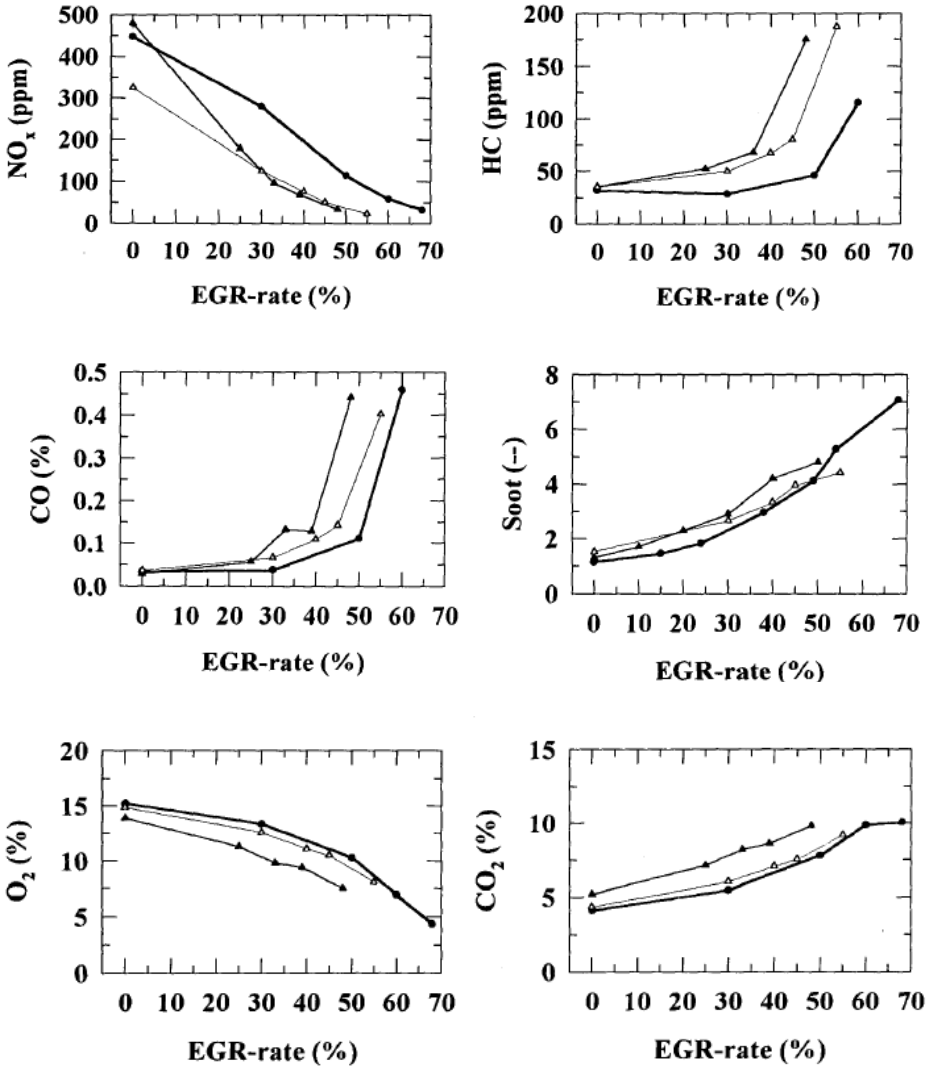

Figure 2.32 : Effects of EGR on engine emissions

As expected, intake manifold temperature increases with increasing ratio of recirculated exhaust and charge density increases. Figure 2.33 shows the intake manifold temperature varying with EGR ratio [8].

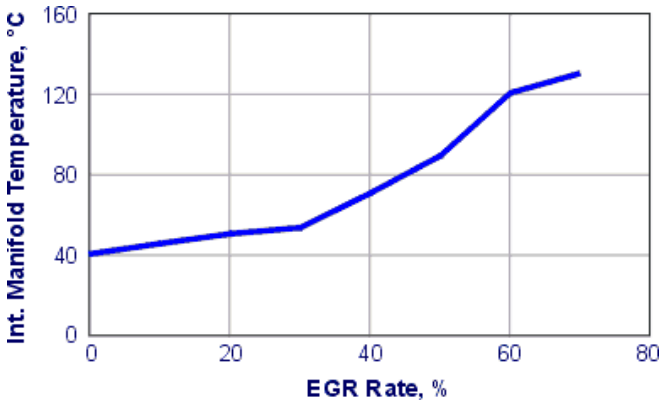

Figure 2.33 : Effect of EGR rate on intake manifold temperature

Conversely, flame luminosity and its temperature were reduced with increasing EGR. At 50% EGR, the flame temperature was reduced by about 100 K. Since NO_x formation is strongly flame temperature dependent, it was suggested that reduced combustion flame temperature is the major reason for NO_x reduction [8].

Through its effect on both intake air and combustion temperatures, EGR also affects the temperature of exhaust gases. This concept is being used by several emission aftertreatment manufacturers that the regeneration of the DPFs may be facilitated through an increase in the exhaust gas temperature caused by EGR. At high engine loads, uncooled EGR increases the exhaust gas temperature however in case of cooled EGR; there is no general rule as to its impact on exhaust temperature, which depends on the effectiveness of EGR cooling and other variables [3].

2.6.1 Effect of EGR on NO_x emissions

It has been previously referenced from Ladommatos et. al (1996) that the most significant effect of EGR was found to be the reduction of O_2 flow rate to the engine i.e. the dilution effect. This resulted in a reduced flame temperature during combustion thus a reduced rate of NO_x formation. In contrast, the reduced O_2 flow rate and local flame temperature caused an increase in particulate emissions due to reduced oxidation rate [2]. Moreover, EGR increases the inlet charge temperature resulting in the increased temperature throughout the entire cycle, which causes increase in NO_x emissions [9].

Figure 2.34 shows the Φ-T map for soot and NO based on experimental results by Kamimoto and Bae. NO_x forms at high temperatures (T) and low equivalence ratios (Φ). This suggests that the local combustion temperatures should be kept below approximately 2200K to avoid high NO concentrations at the same time with low equivalence ratios. When the equivalence ratio is increased, it becomes necessary to further decrease the maximum allowable temperature to avoid soot formation. If the temperature is kept below approximately 1650K, both concentration areas are avoided completely, regardless of Φ [9].

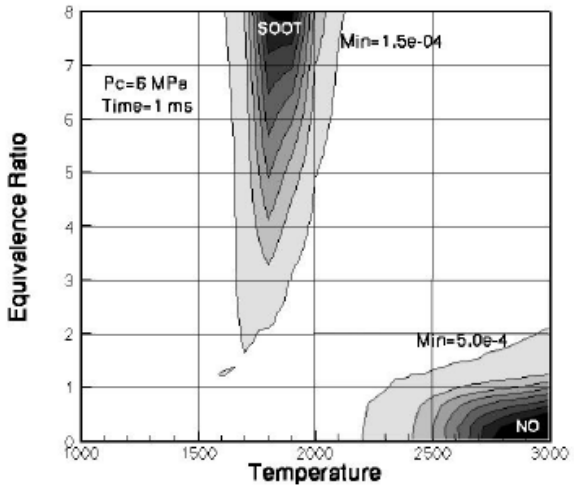

Figure 2.34 : Soot and NO concentration as a function of equivalence ratio and temperature. Soot in g/m^3,NO in mole fraction and temperature in K

NO_x reduction by means of EGR has been worked by number of researchers and had been concluded that increasing EGR rate decreases NO_x formation rate due to the above-mentioned effects.

At higher loads, NO_x is higher due to the higher in cylinder temperatures. EGR needs to be cooled for high load usage, as cooled EGR will displace less of the fresh air volume thus maintaining a sufficient overall A/F ratio for good combustion efficiency. Dependence of NO_x emissions on engine load is illustrated in Figure 2.35 [1].

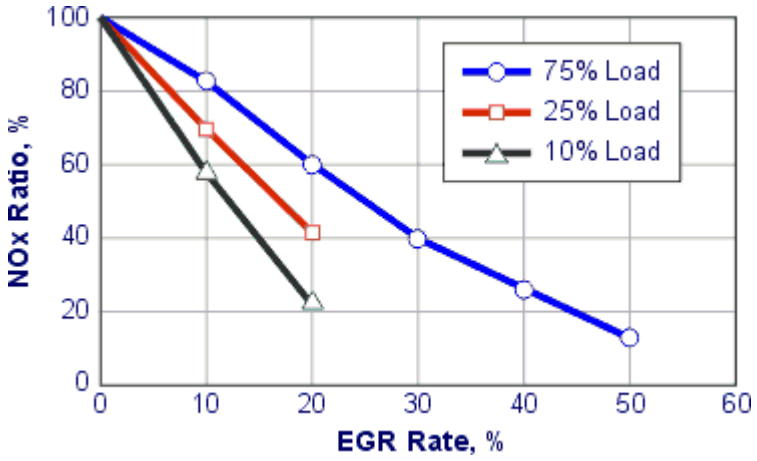

Figure 2.35 : Effect of EGR on NO_x emissions at variable loads

Effects of cooled and uncooled EGR on NO_x, PM and the intake temperature are investigated by Herzog et al. (1992). It is important to note that at low loads, NO_x reduction capability of cooled EGR is less than that of uncooled. This is the result of

the increased premixed burning due to the increased ignition delay period. This is illustrated in Figure 2.36 where NO_x emissions with cooled EGR are found to be higher than with uncooled EGR. However, in any case, cooled EGR reduces the increase of particulates [10].

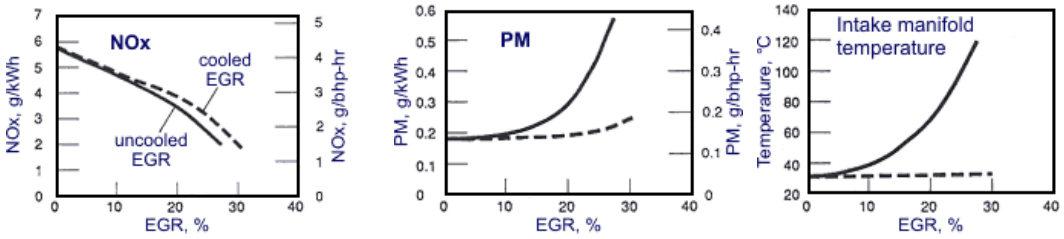

Figure 2.36 : Effect of cooled and uncooled EGR on NO_x and PM at low load

2.6.2 Effect of EGR on PM emissions

The reduction in oxygen availability in the burning regions of the combustion chamber by means of EGR impairs the soot oxidation process. Additionally, the reduction in oxygen concentration in the burning regions reduces the local flame temperature, which then reduces the soot oxidation rate. Therefore, with more EGR, more soot formed during combustion remains un-oxidized. The change of soot with A/F ratio is shown in Figure 2.37 [11].

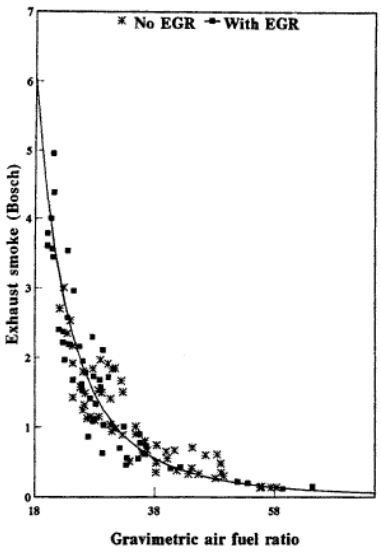

Figure 2.37 : Change of soot with A/F ratio

Ladommatos et al. (1996) also investigated the effect of cooled EGR. It is given that cooling EGR improves the NO_x-PM trade off as shown in Figure 2.38. At a given level of EGR, soot emission is reduced with cooled EGR as NO_x is also reduced [11].

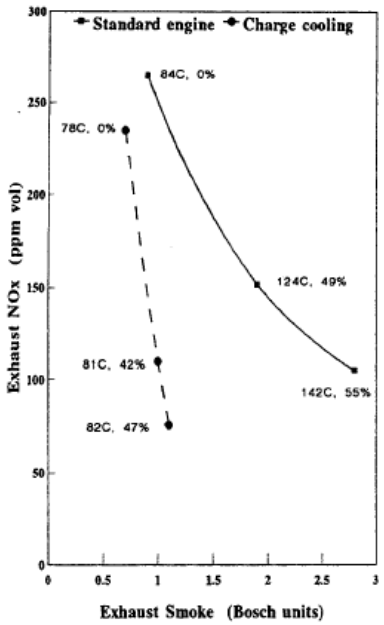

Figure 2.38 : Trade-off between exhaust NO_x and smoke (cooled and uncooled)

Particulate sample analyses showed that the soluble organic fraction (SOF) remained constant while insoluble fraction (mostly carbonaceous) of the particulates increase with the increasing rate of EGR (Figure 2.39). Soluble organic fraction of particulates can be reduced by oxidation type converters but insoluble fraction cannot be. DPFs are designed to control the insoluble PM fraction of EGR where very low PM emission levels are required [12].

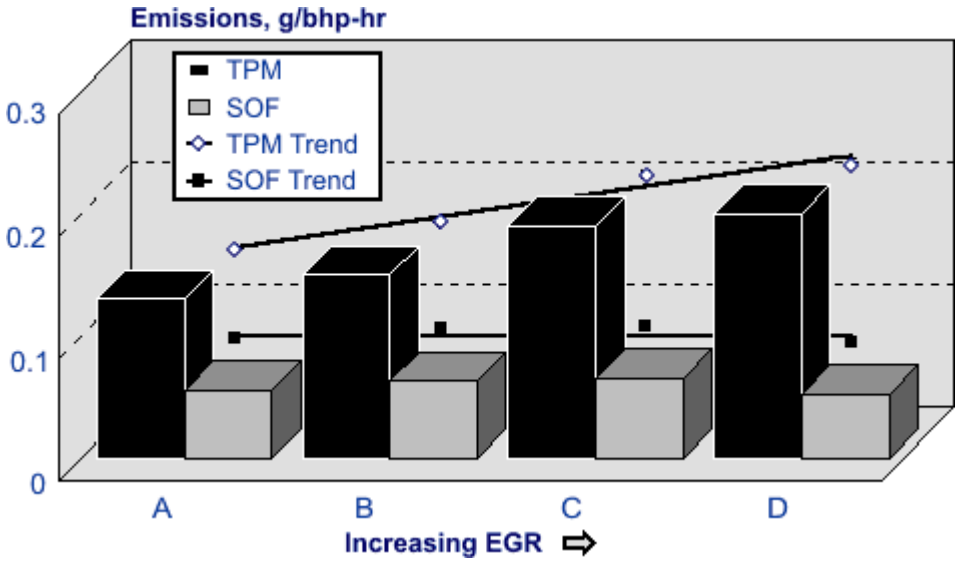

Figure 2.39 : Effect of EGR on PM composition

Effects of excessive EGR on PM has been investigated by Alriksson et al. (2005) and it has been reported that increasing EGR levels up to 55% increases the soot

emissions due to the reduction in soot oxidation rates with lower in cylinder temperatures rather than to increased soot formation. However, when the EGR is increased more, soot reduces significantly as shown in Figure 2.40. The reason for the steep change is that the soot emission is steeply changing with changes in temperature than with changes in equivalence ratio. It can be concluded that the reduction in soot emissions is the result of low temperature combustion due to low O_2 availability [9].

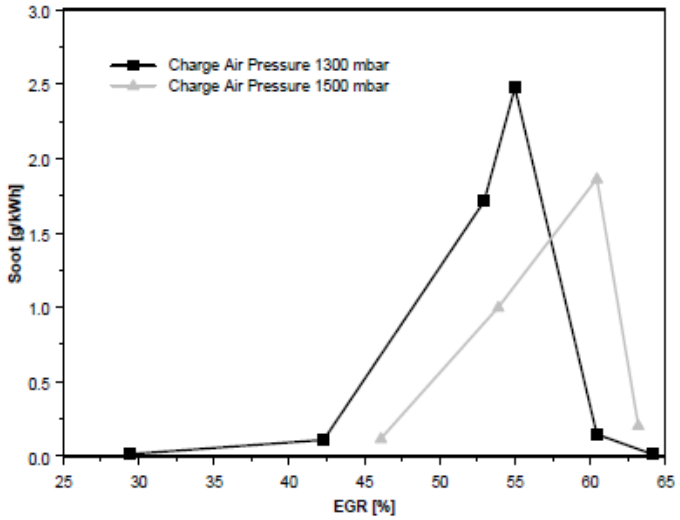

Figure 2.40 : Change in soot with excessive EGR

2.6.3 Effect of EGR on CO and HC emissions

In Figure 2.41, the Φ-T map for soot, NO_x and CO is shown over the same temperature and equivalence ratio range. It is clear that CO increases with increased equivalence ratio i.e. decreased O_2 concentration. Lack of O_2 at high equivalence ratios obstructs the oxidation of CO to CO_2. In addition, fuel droplets fail to vaporize forming fuel rich mixtures that do not combust properly. At temperatures below $1400\,^{o}K$ and low equivalence ratios, CO concentration increases as there are few OH radicals at low temperatures thus CO is partially oxidized to CO_2 [9].

It can be concluded that decreased O_2 concentration with EGR obstructs the oxidation of CO thus increasing its levels in exhaust.

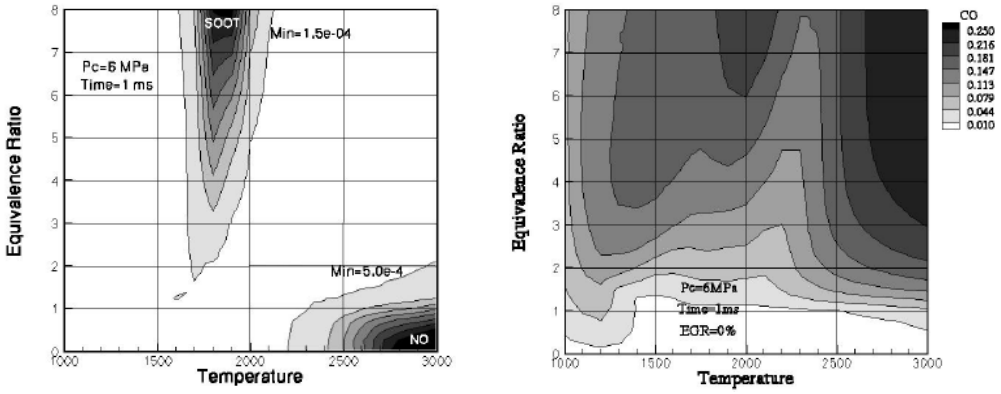

Figure 2.41 : Φ-T maps for soot, NO_x and CO

HC increase dynamics are generally similar to the CO formation. Effect of EGR on HC and CO emissions is shown in Figure 2.42 [9].

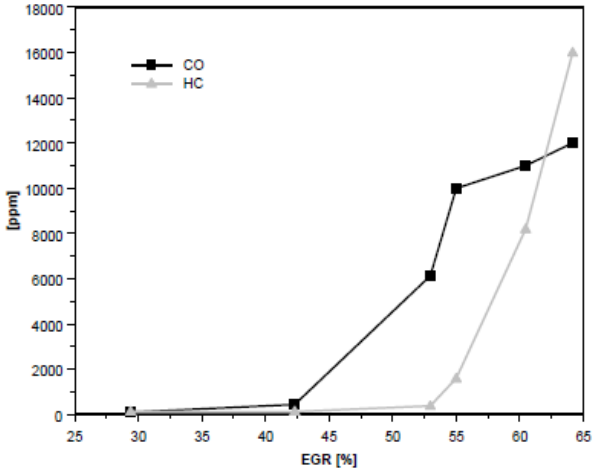

Figure 2.42 : Effect of EGR on HC and CO emissions

2.6.4 Effect of EGR on fuel consumption

EGR tends to reduce the amount of fuel burned in the power stroke, which is evident by the increase in particulate emissions that corresponds to an increase in EGR. Particulate matter (mainly carbon) that is not burned in the power stroke is wasted energy.

Considering NO_x reduction via injection timing retard, EGR increases fuel consumption less than timing retard. In a study by Majewski and Khair, it is reported that using EGR in reducing NO_x from 4.0 g/bhp-hr to 2.8 g/bhp-hr is more efficient than achieving the same reduction through injection timing retard. This is shown in Figure 2.43. PM emissions on the other hand are increased more with EGR than timing retard [1].

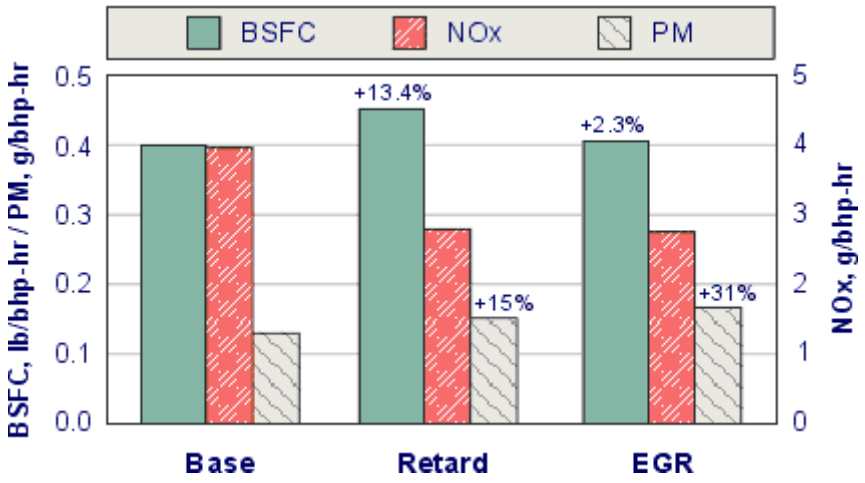

Figure 2.43 : Effect of EGR vs. injection timing retard on BSFC

2.6.5 Effect of EGR on heat release

Use of EGR modifies the inlet charge properties, which then affects the combustion process and consequently the exhaust emissions.

Introduction of exhaust gas into the inlet charge increase the ignition delay period due to the lack of O_2 and giving more time for the spray to penetrate and the auto ignition locations to be shifted towards the wall of the combustion chamber. With EGR, combustion shifts further towards the expansion stroke. This results in the products of combustion spending shorter periods at high temperatures, which lowered the NO_x formation rate as well as lower soot oxidation with increased incomplete combustion products. Although, the longer ignition delay periods, associated with EGR, could have resulted in higher rates of pre-mixed burning, the lower oxygen availability reduced the peak rate of pre-mixed burning. This leads to reduction in combustion temperatures and pressures and, consequently, to lower NO_x and higher particulates. In addition, the reduction in pre-mixed burning resulted in increased diffusion burning, which in turn, increased the particulate emissions [13].

Effect of increasing EGR on the rate of heat release (RoHR) with a constant air pressure, constant amount of injected fuel and fixed SOI is given by Alriksson et al. (2005). The ignition dwell (which is defined as the time from the end of injection to the start of the high temperature reactions) correlates positively with EGR (Figure 2.44) [9].

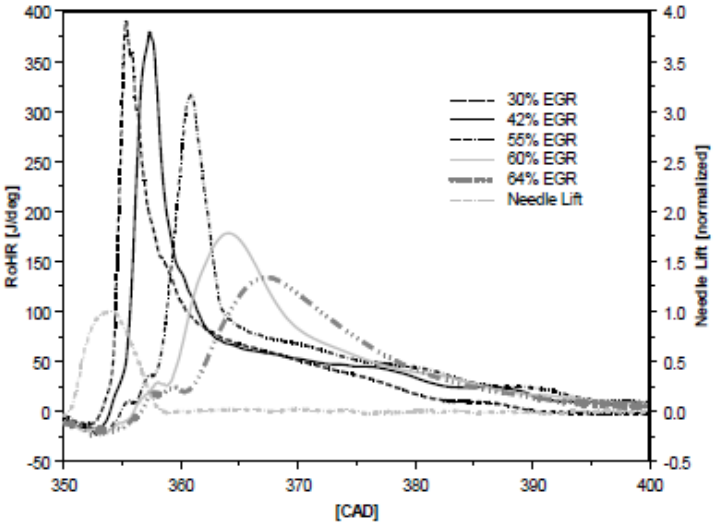

Figure 2.44 : Effect of increasing EGR rate on rate of heat release

2.6.6 Additional benefits of EGR

Reduced heat rejection: Lowered peak combustion temperatures not only reduce NO_x formation, it also reduces the loss of thermal energy to combustion chamber surfaces, leaving more available for conversion to mechanical work during the expansion stroke.

Reduced pumping losses: By recycling the exhaust gas prior to the turbocharger as in high pressure EGR configuration, pumping losses are reduced.

Reduced chemical dissociation: The lower peak temperatures result in more of the released energy remaining as sensible energy near TDC, rather than being bound up (early in the expansion stroke) in the dissociation of combustion products. This effect is relatively minor compared to the first two.

Reduced noise: By reducing peak firing pressures, EGR reduces noise and can eliminate diesel knock. This effect is particularly noticeable at idle [3].

2.6.7 Additional disadvantages of EGR

Increased wear and oil degradation: The level of carbon present in the cylinder has a fundamental influence on piston ring and liner wear and on oil quality. When EGR was applied a significant increase in carbon content was observed. It is suggested that soot can absorb the antiwear additives of the oil, inhibiting their ability to form a protective layer on the component surface. Additionally, soot has an abrasive quality, which is able to remove the antiwear layer, which does form [14].

3. FUEL SYSTEMS OVERVIEW

As more limits were imposed on emissions and pressure from customers on better fuel economy and performance, the demand on the fuel injection systems have increased.

The fuel injection system is important due to its significant effect on engine performance, emissions and noise. The main purpose of a fuel injection system is to deliver the fuel into the cylinders. The fuel is injected through a nozzle with a pressure difference across the nozzle orifices. Unlike gasoline engines, in diesel engines, large injection pressures are required so that the injected fuel enters the cylinder with a sufficient velocity to achieve good atomization for enabling rapid evaporation and improved mixing with air.

A modern fuel injection system should accurately meter the fuel, divide the fuel equally among the cylinders, well time the injection and properly atomize the fuel. Even those are all achieved; the system may not provide the desired combustion efficiency. Addition to these capabilities, a fuel injection system should also be capable of doing multiple injections (pilot injection and post injection) and scheduling the injected fuel, which is known as rate shaping. For instance, for best noise control and avoid white smoke during a cold start pilot injection is preferred. Under the low load and part load conditions, split injections may be preferred whereas at high speed and high load conditions, a ramped injection might be preferred. Therefore, a fuel injection system should be flexible to achieve desired fuel injection strategies. With the introduction of electronic control systems into engine engineering, injection strategy can now be adjusted more precisely depending on the engine running conditions.

In fuel injection systems, the fuel is drawn from the fuel tank by a supply pump and forced though the fuel filter to the injection pump. The injection pump generates the required fuel injection pressure and sends the fuel under pressure to the injectors located at the top of each cylinder. In general, the fuel travels in high pressure pipes when going to the injectors or as in common rail systems, fuel is first send from fuel

pump to fuel rail then to the injectors. The fuel injection pressure varies from system to system but it is generally between 200 to 2000 bars.

3.1 Diesel Fuel Injection Systems

Fuel injection systems can be classified among the basis of whether they are mechanically or electronically controlled or among the basis, how the high pressure for injection is generated. A summary of most commonly used fuel injection systems is summarized below.

3.1.1 In-line fuel injection pumps

In this system, fuel is delivered by the supply pump to the high pressure pump and introduced into the plunger and barrel assembly via one or two fill ports. Each cylinder has its own plunger and barrel assembly. The clearance between the barrel and plunger is on order of a few μm in order to seal even under very high pressures and low pump speeds. Each plunger is raised by the cam on the pump camshaft and is forced back by the plunger return spring [1, 15].

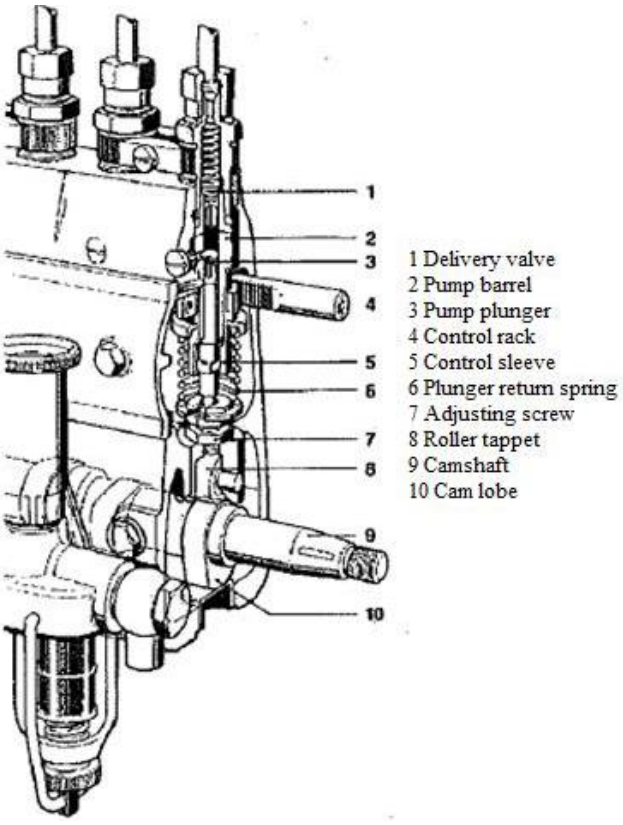

Figure 3.1 : In-line fuel injection pump

Fuel from the fuel supply pump enters into the top of the plunger through the fill ports. The vertical movement of the plunger blocks the fill ports and stops the fuel supply into the fuel injection pump. As the vertical movement continues, fuel pressure increases and nozzle opens when necessary pressure is obtained leading to the injection into the cylinders. Each plunger has a machined slot and a helix near its top. Further movement of plunger causes the direct fuel communication between the high pressure fuel at the injection side and low pressure at the supply side. The fill ports now act as the spill port and injection ends. Plunger stroke phases in a complete injection cycle are shown in Figure 3.2.

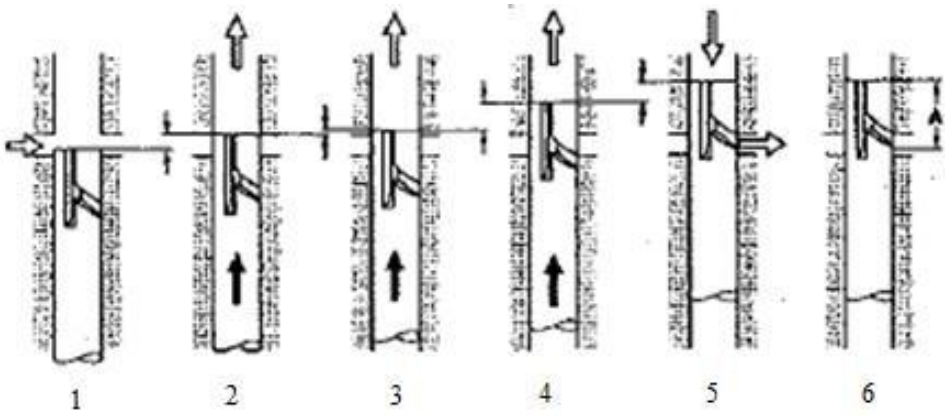

Figure 3.2 : Plunger stroke phases in a complete injection cycle

When the plunger is at the bottom dead center (1), fuel flows from the injection pump's fuel gallery into the high pressure chamber of the plunger and barrel assembly. Prestroke phase (2) is the plunger stroke from BDC to closure of the inlet port by the top edge of the plunger. In retraction stroke (3) plunger moves from the end of prestroke until the delivery valve opens. Further movement of the plunger increases the fuel pressure, which is referred as the effective stroke (4). This continues until a direct fuel communication between the high pressure and low pressure sides (5). After the completion of injection stroke, plunger moves back to the BDC (6).

The stroke of the plunger is fixed thus it depends on the cam lobe profile. However, the effective stroke can be changed by means of rotating the plunger in the barrel. This is done with a control rack, which is shown in Figure 3.3. When the load is increased by the driver using the gas pedal, linkages from the pedal to pump system rotates the plunger to the desired position thus adjusts the fuel quantity. When the

relative position of the plunger helix to the fill port is changed, the quantity of fuel introduced into the assembly changes [1].

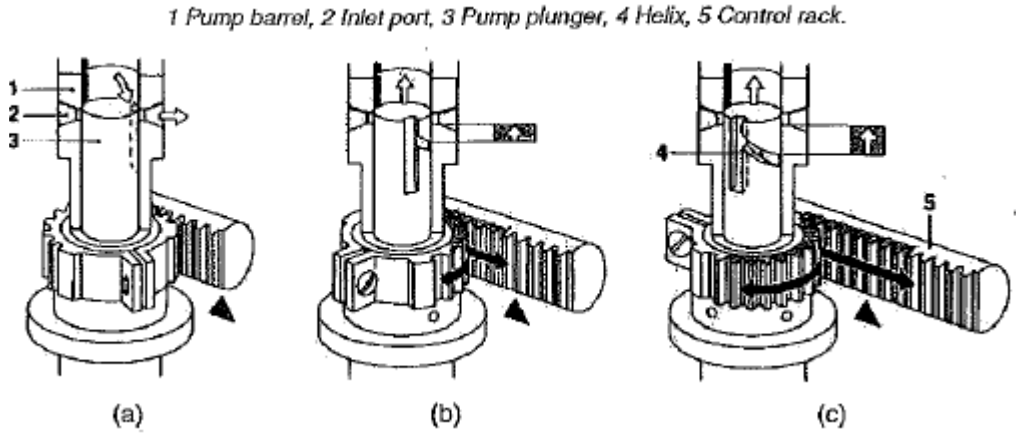

Figure 3.3 : Fuel quantity control: (a) zero (b) partial (c) maximum delivery

3.1.2 Distributor type fuel injection pumps

These pumps are used in multicylinder engines with less than 30 kW power per cylinder with a max injection pressure of 750 bars. These pumps have only one barrel and plunger thus cheaper than the inline fuel injection systems. The pump is driven by either belt or a gear [15].

In these pumps, as illustrated in Figure 3.4, a surface multilobe cam plate (1) is rotated through the driveshaft dogs. It is connected to the plunger on which slides a precision fit collar (2). The plunger (3) has machined slots near its end (4). By these slots, the fuel is supplied from the passage (5) to the high pressure chamber (6). The pressurized fuel goes to the delivery valve (8) through the outlet passages (7).

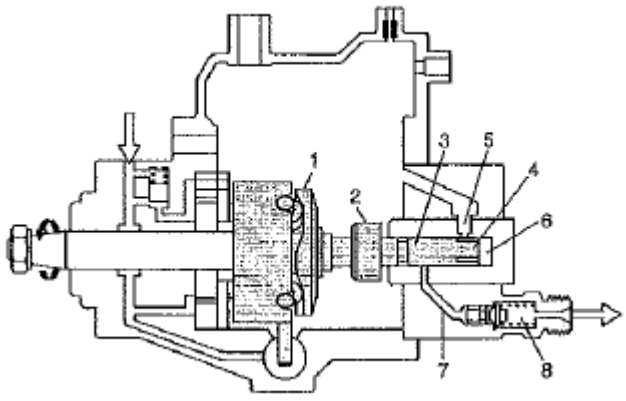

Figure 3.4 : Distributor type fuel injection pump

More details of the distributor type pump are given in Figure 3.5. In step d, fuel enters into the high pressure chamber. At this point outlet and transverse cutoff bore

are closed. With the rotation of the plunger, the inlet is blocked and outlet opened (a). With the travel of plunger to the right fuel is pressurized and fuel goes through the internal channel in plunger to the injectors (b). The fuel delivery ends with more rotation of plunger causing the cutoff bores open (c).

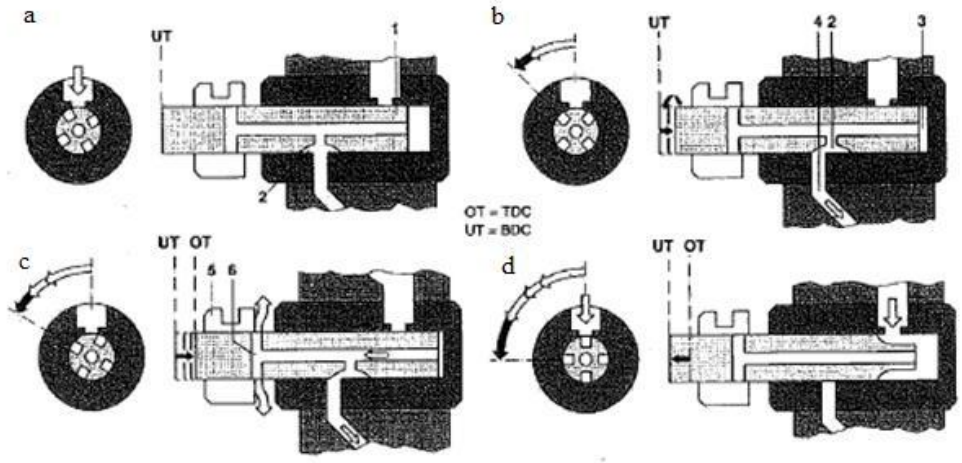

Figure 3.5 : Distributor pump stroke and delivery phases [1]

3.1.3 Unit pump systems

These pumps are used on large engines with outputs of more than 100 kW per cylinder.

Unit pumps systems can be thought as unconnected pumping elements from an inline pump system but separated from the multicylinder injection pump. In unit pump systems, each pumping element is placed in proximity to its power cylinder. The short distance between the pump and the injector nozzle allows shortening of the length of the high injection pressure lines and helps alleviate the problem of after injection, secondary injection or any other injection abnormalities in the injection system. Unit pumps operate on the same principle as the inline pump.

As shown in Figure 3.6, there is a rocker mechanism (2, 3) between the injection cam (10) and tappet roller (7). By adjusting the eccentricity, the relative position of the tappet roller to the cam can be varied thus giving ability to modify injection timing according to fuel quantity [1, 15]. Unit pump systems are controlled electronically by a control unit, which actuates the solenoid valves to adjust start and duration of injection.

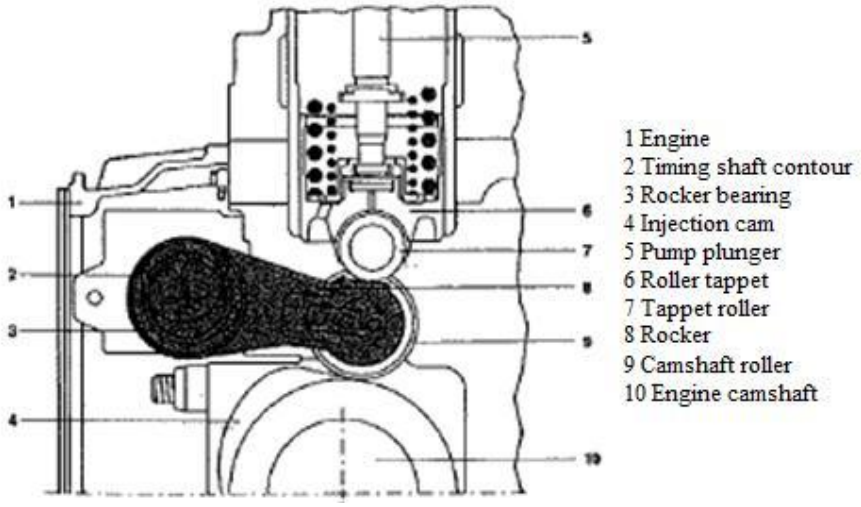

Figure 3.6 : Unit pump system

3.1.4 Common rail systems

In fuel injection systems summarized above, the fuel injection pressure depends on the engine speed. When the engine is running at low speeds, fuel pressure is low and at higher engine speeds, higher injection pressure is obtained. This dependence on engine speed restricts the designers in reduction of emissions and improving fuel economy.

In common rail systems;

- Functions of generating high pressure and fuel injection are separated. Even at low engine speeds, high injection pressures can be obtained
- Fuel quantity can be adjusted accurately
- Injection timing is more flexible
- Up to six injections (pilot injection, post injection, main injections are possible in one combustion stroke)

This gives flexibility to the engineers for obtaining the desired injection strategy at each engine running condition.

The common rail consists of a supply pump, high pressure pump, fuel rail, high pressure pipes, injectors, electronic control unit and various sensors and actuators. The supply pump draws the fuel from the fuel tank and delivers to the high pressure fuel pump. The high pressure pump is actuated by the engine continuously ensuring that the pressure inside the fuel rail is kept at the desired value. The high pressure

pump is generally a radial type with three pistons. It is driven by the engine via a gearwheel, chain or toothed belt depending on the application. The power consumed by the pump increases when the injection quantity increases at a given injection pressure. The fuel is supplied to fuel rail via a high pressure feed pipe. A common rail system layout is shown in Figure 3.7.

The pressure in the rail is measured via a sensor and this information is provided to ECU. ECU compares the actual pressure with the set value, which depends on the engine calibration map, and adjusts the delivery of the fuel pump to achieve a minimal deviation from the set point.

To keep the fuel pressure at the desired level, a pressure control valve at the high pressure side is used. This valve is generally mounted on the rail. The fuel that is not necessary in the injection is sent back to the fuel pump to keep the rail pressure at the desired level. This is also used when there is a sudden change on fuel pressure demand.

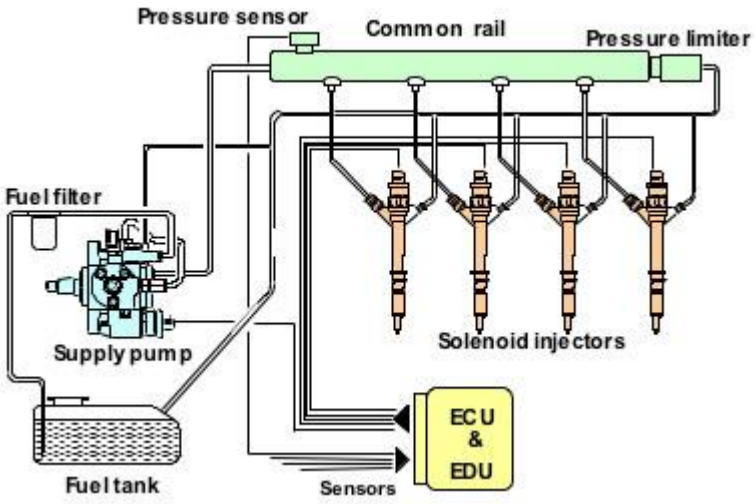

Figure 3.7 : Common rail fuel injection system

In recent common rail systems, fuel is metered at the low pressure side by a solenoid valve. This reduces the power demand of the fuel pump as well as reduces the fuel temperature in the system and fuel tank.

The injectors are the key components of the common rail fuel injection system. Injectors are fitted on the cylinder head via injector clamps that are fitted on cylinder head. They are supplied with fuel from the rail by short pipes. The fuel injectors are typically ECU controlled. When the fuel injectors are electrically activated, they are

mechanically or hydraulically opened and fuel is sprayed into the cylinders at the desired pressure. The most commonly used types are solenoid valve injectors and piezo actuated injectors.

The solenoid operated injectors are controlled by a special solenoid operated valve. It consists of an injector body similar to a conventional nozzle holder, a multihole nozzle, a valve body with two orifices controlling the pressure in the control chamber, a control piston, an armature device with a small ball to open and close the throttle, a solenoid coil assembly and the electrical connection for the ECU.

With the solenoid valve closed, the complete chamber volume and the rail are at the same pressure. The nozzle needle is forced against its seat by a spring. When the solenoid valve opens, fuel flows from the valve control cavity and into the fuel return. The feed throttle prevents complete pressure equalization, and the pressure in the cavity drops. The excess pressure in the chamber volume overcomes the spring force and lifts the needle so that injection can start. The injection continues until the pulse applied to solenoid valve is cut. The ECU calculates the start of injection and injection duration and then actuates the injector. Injection quantity is determined by the rail pressure, hydraulic flow of the nozzle and pulse length from the ECU. A common rail injector is shown in Figure 3.8 [1].

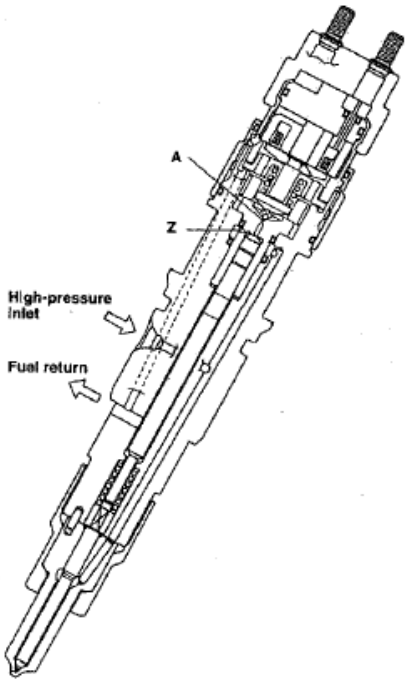

Figure 3.8 : A common rail injector

The operating principle is almost common between a solenoid and a piezo injector. The piezo injectors use an actuator made of piezo crystals to control the injection process. The piezo crystals expand when an electrical field is applied. The electronically controlled piezo actuator is capable of switching five times as fast as a solenoid and the movement of the piezo package is transmitted non mechanically (therefore entirely without friction) to the nozzle needle. This doubles the injector's switching speed, allowing a more precise measurement of the amount of fuel injected [16].

3.2 Combustion in Diesel Engines

The basic operation principle of diesel combustion of its unique way of releasing the chemical energy stored in the fuel. To facilitate combustion, oxygen must be made available to the fuel. The most important aspect of this process is mixing of fuel and air, which is referred as mixture preparation.

In diesel engines, fuel is injected into the cylinder just a few degrees before the top dead center during compression stroke. The fuel is injected at high velocity through the small nozzles in the injector tip. The fuel then atomizes into small droplets and penetrates the combustion chamber. Due to the high temperature around the injected fuel in the cylinder, fuel vaporizes and mixes with air. As the piston continues to move closer to TDC, temperature increases more and reaches the fuel's ignition point. Instantaneous ignition of some premixed fuel and air occurs after the period that is referred as ignition delay. The end of ignition delay is the start of combustion, and this point shows a sharp increase in cylinder pressure due to the combustion of premixed fuel. This increased pressure shortens the delay in ignition of the unburned portion of the fuel as well as increases the evaporation rate. Atomization, vaporization, mixing and combustion continues until all the injected fuel is consumed.

The overall A/F ratio in diesel engines is characteristically very lean. At peak torque, it is maintained above 25:1 to avoid excessive smoke formation. In turbocharged engines, at idle, the A/F ratio may exceed 160:1. The majority of the air induced into the cylinder is compressed, heated but never used in combustion. However, this excess air helps oxidation of gaseous hydrocarbons and carbon monoxide reducing them to extremely small concentrations in the exhaust gas [1, 15].

Therefore, in summary, the induced charge air, its temperature, its kinetic energy, the injected fuel, its atomization, spray penetration, temperature and its chemical characteristics play a major role in the combustion process.

Addition to the major items, there are more items affecting the combustion significantly. Those are intake port design which effects the mixing, intake valve size which controls the total mass of air induced into the cylinder, compression ratio which effects fuel vaporization thus the mixing rate and combustion quality, the injection pressure which controls the fuel atomization, nozzle geometry effecting spray penetration and the spray geometry effecting the emissions.

3.3 Phases of Diesel Combustion

The combustion process is explained in four phases as follows.

3.3.1 Ignition delay

In diesel engines, combustion starts by auto ignition of fuel at several locations in the combustion chamber. The time required for the fuel to produce the ignition nuclei and detectable combustion is referred as 'ignition delay'. This is the period between a and b as shown in Figure 3.9. This period ends where the heat release appears.

Ignition delay period is a very important parameter due to its effect on combustion. The duration of ignition delay affects the mechanical stresses, engine noise as well as the emissions [15].

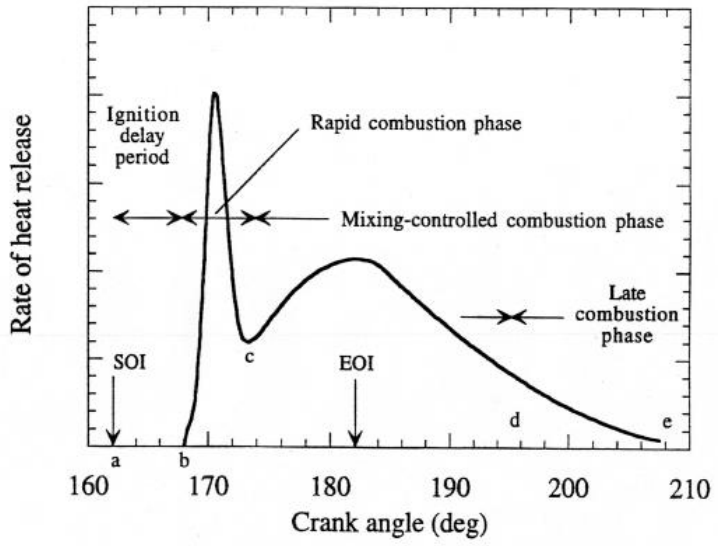

Figure 3.9 : Diesel heat release diagram

During the ignition delay period, physical and chemical processes take place. All those periods are generally completed in an extremely short time. The physical processes are;

- Spray break up and droplet formation
- Fuel and air mixing
- Fuel evaporation
- Diffusion of vapor fuel into the air to form combustible mixture

Chemical processes on the other hand are;

- Preflame oxidation of hydrocarbon fuel that has been premixed
- Localized ignition at several points in the combustion chamber

The chemical processes start as soon as fuel vapor gets in contact with the air. Since in early stages of combustion, the fuel is not yet vaporized; it can be considered that in the early stages of ignition delay period, physical processes are dominant.

The main parameter affecting the ignition delay is fuel cetane number. A fuels ability to ignite is defined by the cetane number, which is a very important parameter due to its effects on fuel conversion efficiency, smoothness of operation, misfire, smoke emissions, noise and ease of starting. If the fuel cetane number is low, the ignition delay is longer resulting in most of the fuel being injected until the combustion starts. This results in rapid burning process thus produces high rates of pressure increase and higher peak pressures. Under extreme conditions, when auto ignition of the most of the injected fuel occurs, this produces an audible knocking sound, which is referred as ‘diesel knock’. If the cetane number is high, then the ignition delay period is shorter and ignition occurs before most of the fuel is injected. The heat release rate and pressure rise is controlled primarily by the rate of injection and air/fuel mixing and this results in smoother engine operation.

Addition to the cetane number, there are several parameters effecting ignition delay.

Engine load: Figure 3.10 shows the effect of engine load on ignition delay. With the increasing engine load, ignition delay decreases. As load increases, cylinder wall temperatures increase leading to shorter ignition delay period. It is also apparent that with higher fuel cetane number, the ignition delay is shorter [15].

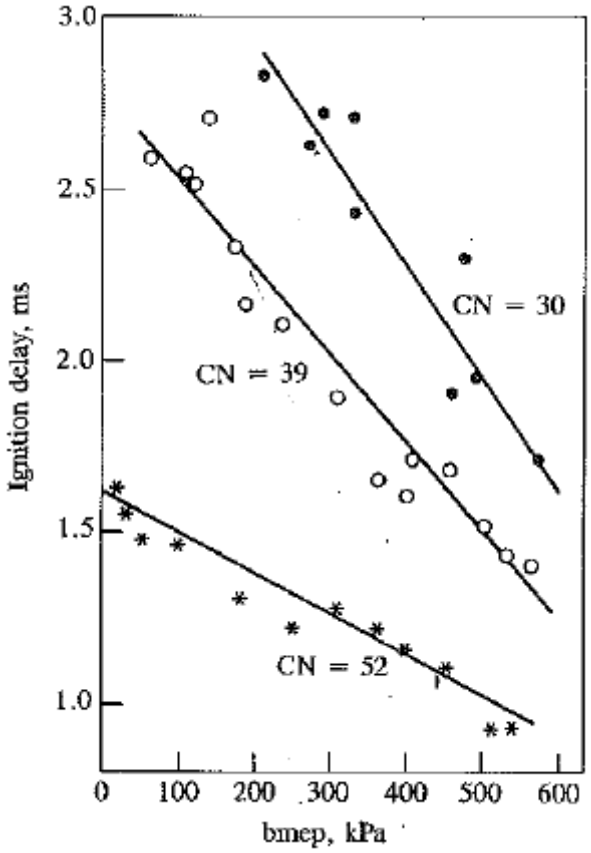

Figure 3.10 : Ignition delay as a function of engine load and fuel cetane number

Injection timing: There is an optimum injection timing for a minimum ignition delay. If the injection is retarded, the in cylinder temperature and pressures are initially higher but decreases as the delay proceeds. This leads to a longer ignition delay. If the injection starts earlier than the optimum crank angle, the pressure and temperatures are not sufficient to start auto ignition, which leads to longer ignition delay periods.

Injection pressure: Increasing injection pressure increases fuel atomization thus shortens the period of physical processes during ignition delay. Therefore, increasing injection pressure decreases ignition delay.

Intake air temperature and pressure: Figure 3.11 shows the effects of intake air pressure and temperature on ignition delay. The studies showed that until 1000K charge temperature, increasing charge air temperature reduces ignition delay. Also increased charge pressure shortens ignition delay. Therefore, a parameter that is affecting the charge air temperature and pressure (such as turbocharging or increasing the compression ratio) affects the ignition delay [15].

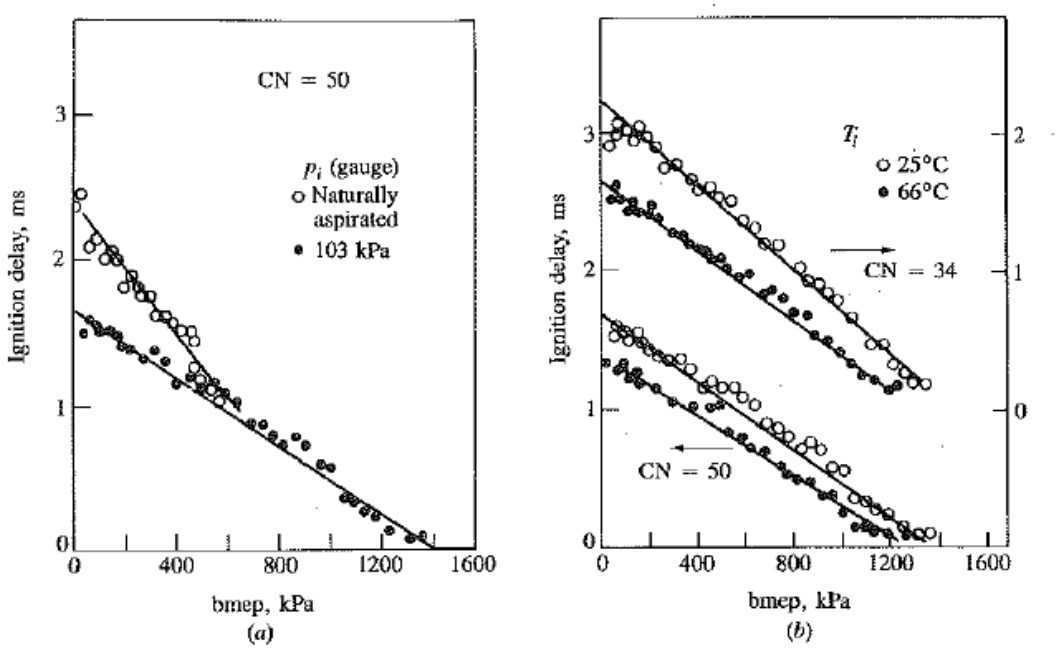

Figure 3.11 : Effect of charge pressure and temperature on ignition delay

Engine speed: Increase in engine speed results in a slight reduction in ignition delay. The change in engine speed affects the temperature/time and pressure/time relationships. Additionally, as the engine speed increases, injection pressure increases. The peak combustion temperature increases with increasing speed due to less heat loss during the compression stroke.

Oxygen concentration: As the oxygen concentration decreases 'such as with use of EGR for NO_x reduction', ignition delay increases.

3.3.2 Premixed combustion

Premixed combustion phase is the period between b and c in Figure 3.9. In this period, combustion of a portion of the fuel injected during ignition delay period occurs. Here the premixed portion of the fuel burns in a very high rate producing high cylinder pressure and temperatures. It is accepted that the rate of pressure rise resulting from the premixed combustion is proportional to the noise intensity in diesel engines.

3.3.3 Rate controlled combustion

In this phase, which is the period between c to d in Figure 3.9, the bulk of the fuel that did not participate to the premixed burning is consumed. This fuel controls the burning rate, which depends on the rate it mixes with air and gets ready to burn.

3.3.4 Late combustion

The cylinder charge is non-uniform and mixing during the late combustion phase promotes more combustion. The source for the combustion may be the soot and fuel rich combustion products. Heat release in this phase continues at a lower rate well into the expansion stroke. It is the period between d and e in Figure 3.9.

Figure 3.12 shows the heat release rate, cylinder pressure and rate of fuel injection extracted from a single injection. The initial sharp rise in the heat release results from the burning of the premixed portion of the fuel. The negative heat release just after the start of injection is due to the heat transfer from the air to fuel. The magnitude of the initial peak of the diagram depends on the ignition delay and is higher for longer ignition delays.

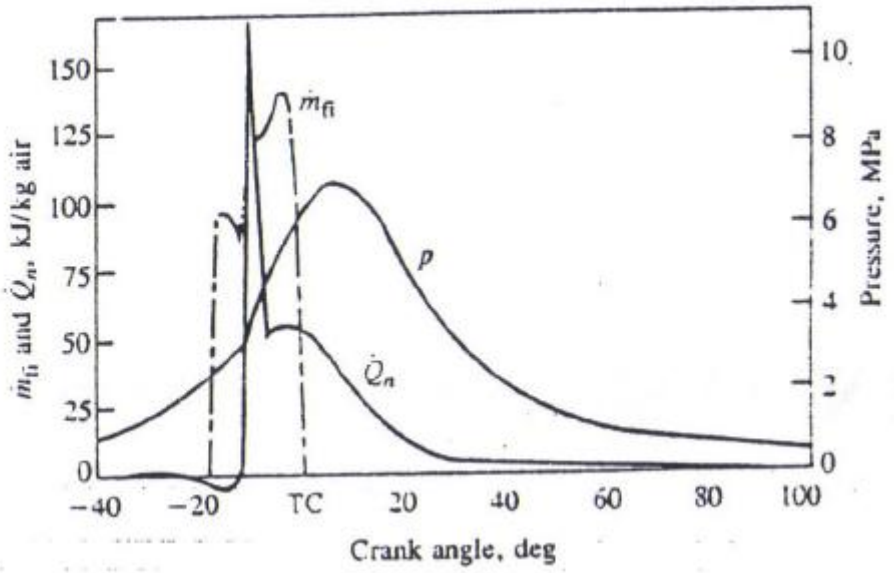

Figure 3.12 : Cylinder pressure, rate of fuel injection and net heat release rate [15]

3.4 Fuel Spray Structure

The combustion in direct injection diesel engines is heterogeneous and three-dimensional process. Majewski and Khair (2006) had shown a schematic of a fuel structure that has been derived from the photographic film studies of combustion in direct injection diesel engines. This illustrates the case when a fuel jet is injected radially into a swirling flow. As per this study, the spray at the beginning of combustion is similar to the one shown in Figure 3.13. The average distance is greatest near the edge downstream from the centerline of the spray where smaller

droplets are concentrated. The local A/F ratio is highest along the centerline of the spray and decreases as moved to the outside from the centerline. It is assumed that the most of the droplets away from the spray core are completely or mostly evaporated before the start of ignition [1].

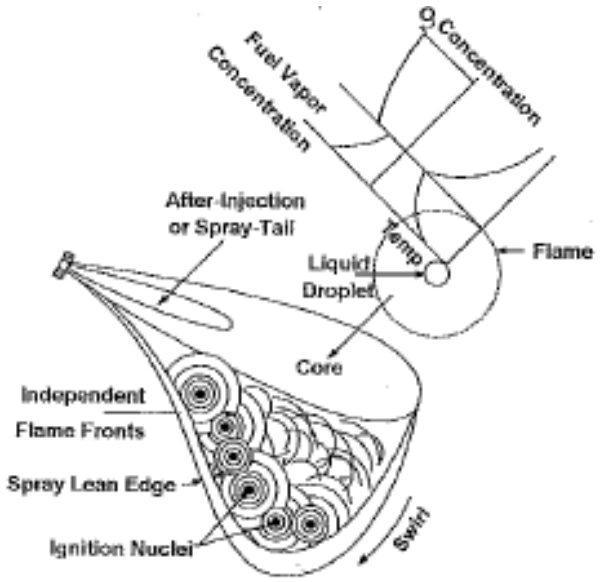

Figure 3.13 : Mixing of fuel spray with air

The spray is divided into 4 regions as follows:

3.4.1 Lean flame region

Ignition nuclei forms at several locations where mixture will most likely to auto ignite. Once auto ignition starts, small independent flame fronts propagate from the nuclei and ignite the combustible mixture around. The region where these independent ignition nuclei propagate is described as lean flame region (LFR). Generally, this is the region where NO_x is formed at high loads, when the load is lower, the temperature may not be sufficient for NO_x formation.

3.4.2 Lean flame out region

Outside the lean flame region is referred as lean flame out region (LFOR). In this region, the mixture is too lean to support combustion. Majeski and Khair (2006) suggested that the LFOR is the one of the major sources of partial oxidation e.g. unburned hydrocarbons [1].

3.4.3 Spray core

Spray core is the region between the LFR and the spray tail. In this region, the fuel droplets are larger and the evaporation rate is higher than in the LFR due to the heat gained from the already established flames. The droplets in this region may be completely or partly evaporated. If they are completely evaporated, the flame will burn all the mixture within the rich ignition limit. If they are not completely evaporated, the remaining droplets may be surrounded by a diffusion type flame as in the LFR and burn as individual droplets.

Spray core region is the source for NO_x at part loads due to the complete combustion in the sufficient O_2 environment. NO_x concentration in this region is mainly effected by the flame temperature. Near the full load conditions, incomplete combustion occurs in many locations in the fuel rich zone and incomplete combustion products such as unburned hydrocarbons increase.

3.4.4 Spray tail

This region includes the largest fuel droplets through the spray. This is caused by the decreased fuel injection pressure and increased cylinder gas pressure near the end of injection process. The spray tail penetration is very poor and it has little chance of finding regions with sufficient O_2 thus in this region incomplete combustion occurs even the evaporation is so quick [1].

The spray in the combustion chamber is divided into six parts as shown in Figure 3.14. It shows the progress of combustion in the spray and the cylinder pressure and temperature at the start of combustion in each region [1].

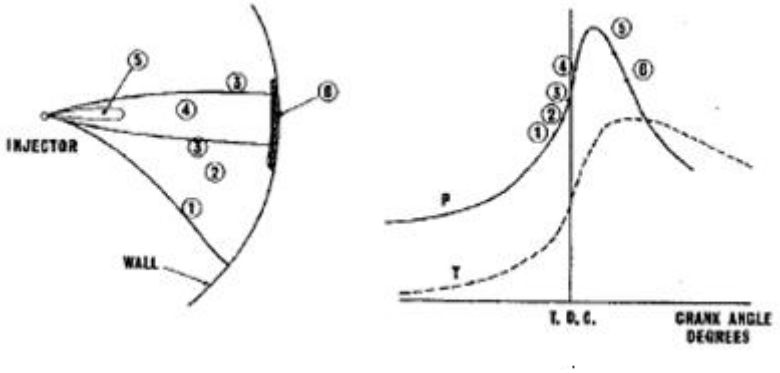

Figure 3.14 : Progression of combustion in the spray

3.5 Atomization

The breakup of the fuel jet as it exits the nozzle hole is described as the spray atomization. The size of the fuel droplets caused by this break up is smaller than the nozzle hole diameter. The atomization continues until the Weber number exceeds a threshold value, which is suggested to be 10. Weber number is defined as the ratio of the inertia forces to the surface tension forces and given by the following formula:

$$W_e = \rho d^2 V^2 / \sigma_d \qquad \textbf{(3.1)}$$

where ρ is mass density, d is droplet diameter, V is velocity and σ_d is surface tension. Schematic of a spray from a single hole nozzle is given in Figure 3.15.

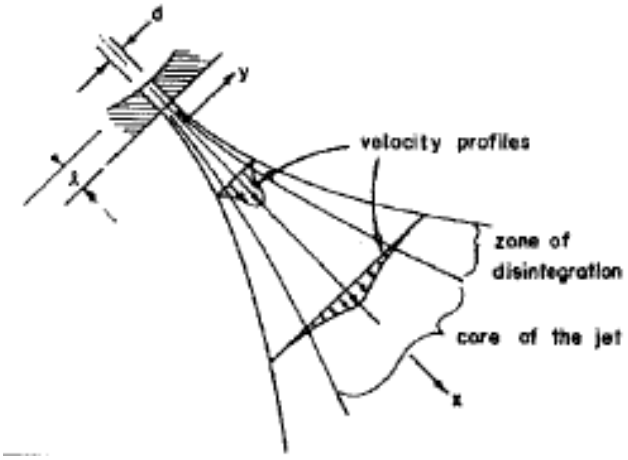

Figure 3.15 : Schematic of a spray from a single hole nozzle

Atomization is divided into 4 regimes [15].

At low velocities, the breakup is due to the unstable growth of the surface waves caused by the surface tension and results in droplets larger than the jet's diameter.

At medium jet velocities, the relative motion of the jet and the air increase the surface tension force. In this case, droplets sizes are in the order of the jet's diameter. This is referred as 'wind-induced breakup regime'.

At high jet velocities, breakup is characterized by the divergence of the jet spray after an intact or undisturbed length downstream of the nozzle. This is referred as 'second wind induced breakup regime'. In this regime, the unstable growth of short-wavelength waves induced by the relative motion between the liquid and surrounding air produced droplets whose average size is much less than the jet's diameter.

At very high jet velocities, the breakup of the outer surface of the jet occurs at or before 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 major atomization components in this regime.

3.6 Spray Penetration

The penetration of spray in the combustion chamber is one of the most important factors for the air utilization. When the droplets travel farther into the combustion chamber, more air is utilized. Additionally, how fast the spray penetrates into the chamber is important as the faster the spray, the greater the mixing rates as well as the air utilization. On the other hand, over penetration is not desired as it causes fuel spray to hit the cold walls increasing unburned hydrocarbon emissions.

The penetration of spray as a function of time and at different injection pressures and ambient temperatures is shown in Figure 3.16. It can be concluded that the spray tip penetration changes linearly with time and is higher for high injection pressures and low ambient temperatures. For the initial motion, injection pressure plays the primary role while the ambient gas density becomes a factor on the penetration of droplets after breakup [1, 15].

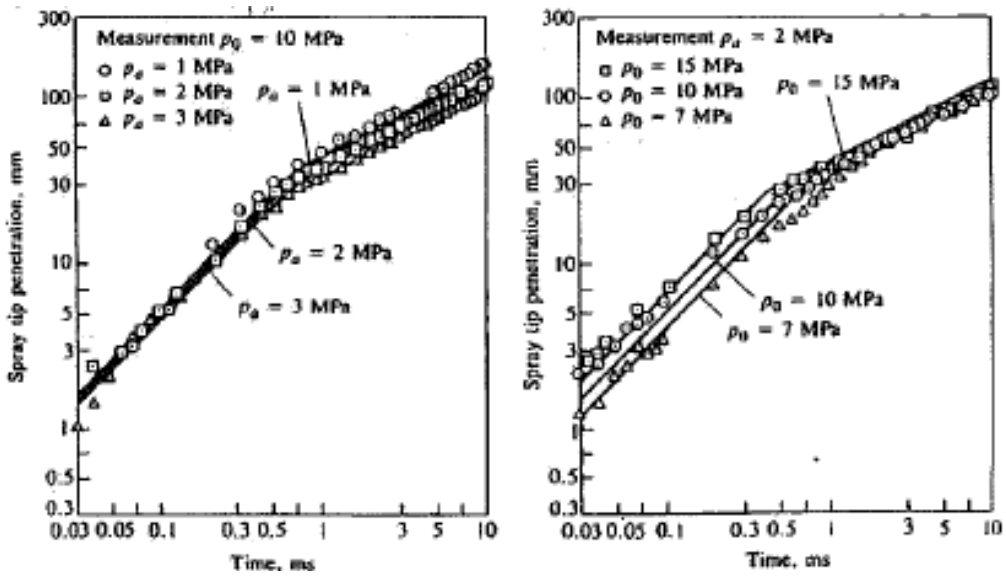

Figure 3.16 : Spray penetration as a function of time

3.7 Droplet Size Distribution

It is expected that when the number of small size droplets in the spray increase, the overall surface area that facilitates heat transfer increases leading to improved vaporization and mixing.

A commonly used description for characterization of the mean diameter of spray is Sauter Mean Diameter (SMD), which is defined as the diameter of the droplet that has the same surface/volume ratio as that of the total spray.

The studies on SMD show that the nozzle geometry has a significant effect on it. It is given that nozzle length/diameter ratio of 4 gives the minimum mean droplet size at low and intermediate injection pressures. As shown in Figure 3.17, for d_n=0.3mm increasing nozzle length increases SMD while decreasing nozzle diameter decreases the mean diameter as expected [15].

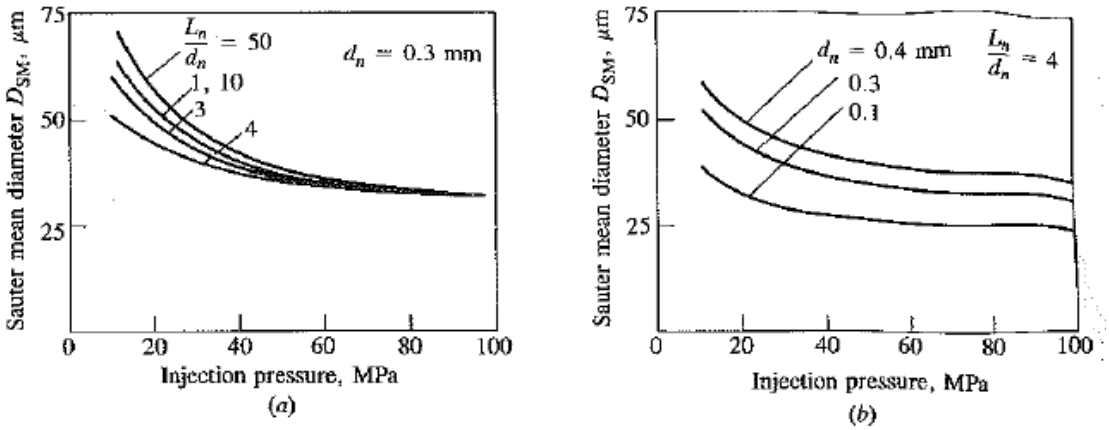

Figure 3.17 : Effect of injection pressure on Sauter Mean Diameter

3.8 Spray Vaporization

Fuel has to vaporize and mix with air to form a combustible mixture. There are three steps that a droplet passes through.

1. Slow down of the droplet due to the aerodynamic drag
2. Heat transfer from the air to the droplet
3. Mass transfer of vaporized fuel away from the droplet

As the droplet temperature due to the heat transfer increases, fuel vapor pressure increases increasing the evaporation rate. As the mass transfer from the droplet

increases i.e. droplet mass decreases, the fraction of available heat required to increase the droplet temperature further decreases. Moreover, as the droplet velocity decreases, the convective heat transfer coefficient between air and fuel decreases. The behavior resulting from the combination of all these factors is shown in Figure 3.18 [15].

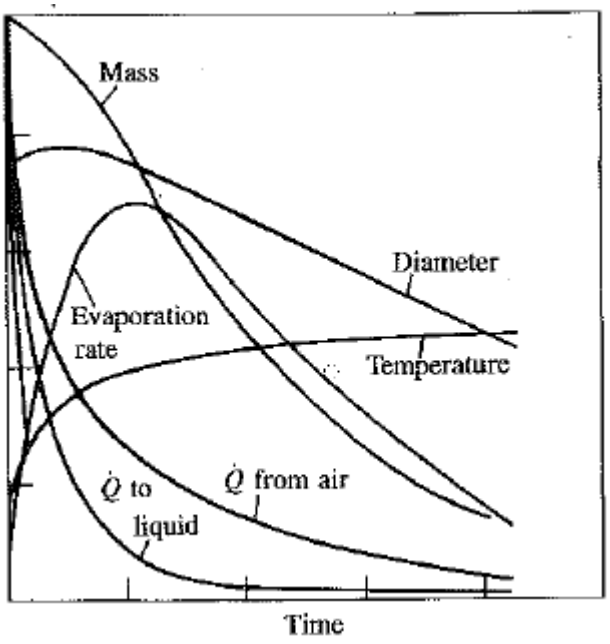

Figure 3.18 : Fuel droplet evaporation process

4. EMISSIONS FORMATION

The combustible mixture in diesel engines is mostly heterogeneous. Fuel is injected into the cylinder where the pressure and temperatures are high. Emissions are formed as a result of burning this heterogeneous air/fuel mixture depending on the prevailing conditions not only combustion but also during the expansion and prior to the exhaust valve opening. Main combustion byproducts consist of NO_x, CO, HC and PM.

There are several parameters effecting engine emissions such as mixture preparation during ignition delay, fuel ignition quality, residence time at different combustion temperatures, expansion duration and general engine features. Incomplete combustion products formed in the early stage of combustion may be oxidized later during the expansion stroke. More combustion is permitted by the mixing of unburned hydrocarbons with oxidizing gases, a high combustion chamber temperature and adequate residence time for oxidation process.

The combustion is a complex series of chemical reactions, reaction rates and heat transfer processes, the resultant emissions formation requires further analysis and understanding. In Figure 4.1, a model for heterogeneous combustion is shown with the sources of emissions and the phases they are formed in [15].

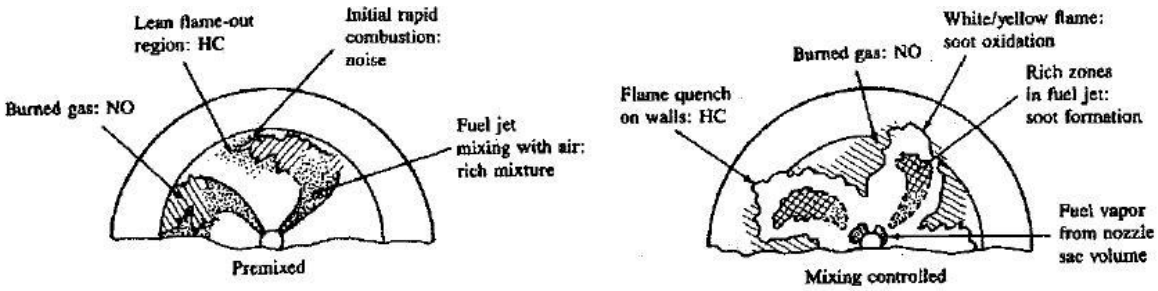

Figure 4.1 : Emission formation mechanism in direct injection combustion system

NO is formed where the temperature is high and the A/F distribution within the burned gases are non-uniform and formation rates are highest in the close-to-stoichiometric regions. Soot on the other side, forms in the rich unburned fuel-containing core of the fuel sprays, within the flame region, where the fuel vapor is

heated by mixing with hot burned gases. Soot then oxidizes in the flame zone when it comes to contact oxygen giving rise to the yellow luminous character of flame. HC forms in regions where the flame quenches both on the walls and where excessive dilution with air prevents combustion process from either starting or going to completion. Fuel that vaporizes from the nozzle sac volume during the later stages of combustion is also a source of HC [15].

4.1 NO_x Formation

NO_x (nitrogen oxides) is the common name for nitric oxide (NO) and nitrogen dioxide (NO_2) in the exhaust of an internal combustion engine.

NO is a colorless, odorless, tasteless and relatively non-toxic gas that rapidly oxidizes to NO_2. NO_2 is a reddish-brown gas with a penetrating odor and is highly toxic even at very low concentrations. It has the capability to destroy lung tissue and cause increased resistance to breathing. It is particularly dangerous to smokers and asthmatics causing coughing, sore throat, bronchitis, pulmonary oedema and running noses. It also damages plants [3].

NO constitutes about 85% of the total NO_x produced in the cylinder. Nitrogen available in the atmosphere is the main source of the NO_x. It is produced at high temperatures (2200K) as a result of chemical dissociation where N_2 and O_2 can react. Well-known mechanism for NO_x formation is Zeldovic mechanism as given below.

$$O + N_2 = NO + N \quad \textbf{(4.1)}$$

$$N + O_2 = NO + O \quad \textbf{(4.2)}$$

$$N + OH = NO + H \quad \textbf{(4.3)}$$

When oxygen and nitrogen are exposed to high temperatures NO forms and an unstable nitrogen (N) atom is left (reaction 4.1). This nitrogen atom, to reach stable form, combines with the available O_2 in the cylinder. This reaction also produces NO leaving an O atom in an unstable condition (reaction 4.2). In reaction 4.3, an unstable N atom combines with OH radicals and leaves an atom of hydrogen [1].

The conversion of NO to NO_2 occurs via reactions such as:

$$NO + H_2O = NO_2 + OH \quad \textbf{(4.4)}$$

Each of these reactions has forward and reverse temperature dependent rate constants, which govern the formation and destruction of the NO respectively.

NO formation rate is given in literature as:

$$\frac{d[NO]}{dt} = \frac{6\times10^{16}}{T^{1/2}} \exp\left[\frac{-69090}{T}\right][O_2]^{1/2}[N_2] \qquad \textbf{(4.5)}$$

where T is temperature and $[O_2]$ and $[N_2]$ are the concentrations of O_2 and N_2 respectively. Equation suggests that the conditions for peak NO_x production are when the highest temperatures are experienced and when there is a reasonable concentration of oxygen, i.e. just after TDC for maximum cylinder temperature and where the mixture is close to stoichiometric. The most of the NO forms within the 20° crank angle after the start of combustion.

As the main factors that affect NO_x production are the peak temperature and the oxygen concentration, NO_x can be reduced by changing:

- **Engine load:** NO_x increases with load due to the increased cycle temperatures. Therefore, at lower engine loads, NO_x emissions are lower.
- **Fuel injection timing:** Advancing fuel injection extends the ignition delay. The reason for the increased ignition delay duration is that the fuel is injected into a lower pressure and temperature ambient. As the ignition delay increases, the fuel burning in the premixed combustion phase increases thus increasing the levels of NO_x. So retarding the injection timing is one of the NO_x reduction methods, which on the other hand increases PM emissions and fuel consumption. This is shown in Figure 4.2.
- **Oxygen concentration:** As discussed in chapter 2, dilution of the charge such as via EGR reduces NO_x emissions.
- **Fuel injection pressure:** Increasing fuel injection pressure improves atomization. With smaller fuel particles, combustion efficiency improves releasing higher heat thus raising the combustion temperature and NO_x emissions.

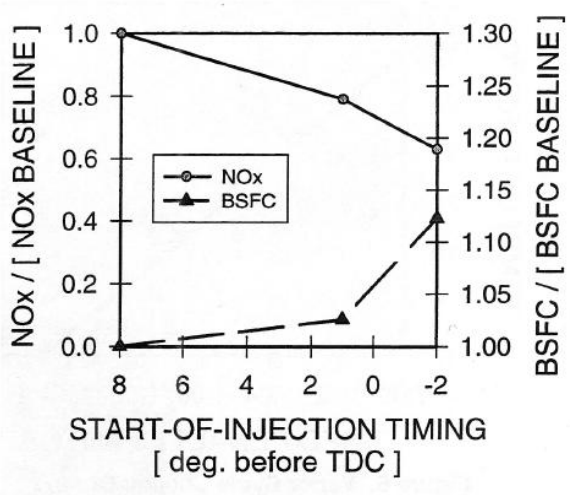

Figure 4.2 : Effect of injection timing on NO_x and fuel consumption [17]

- **Compression ratio:** The higher the compression ratio, the higher the combustion temperatures. Therefore, NO_x emission increases with increasing compression ratio.
- **Charge temperature:** Increasing charge temperature causes the temperature to increase during all the cycle thus increasing the NO_x emissions. Lower boost or increased charge cooling reduces NO_x emissions. Effect of intake manifold temperature on NO_x emissions is shown in Figure 4.3.
- **Cetane number:** Cetane number is a measure for the fuel about its ability to evaporate. The higher the cetane number, the easier it evaporates. Therefore, fuels with high cetane number have shorter ignition delays thus lower NO emissions.
- **Charge c_p:** Increasing charge c_p (e.g. from EGR) reduces NO_x emissions.
- **Swirl rate:** Air motion in the cylinder effects the mixture formation so the combustion. An increase in swirl motion improves mixing and combustion so NO emissions increase. The mechanism responsible for the NO formation is higher heat release. If the swirl is excessive (overswirling), increase in HC and CO, PM and fuel consumption may be experienced.

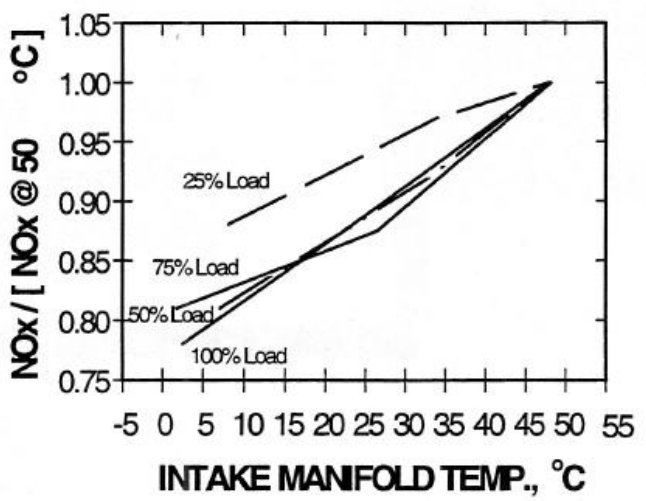

Figure 4.3 : Effect of intake manifold temperature on NO_x emissions [17]

- **Pilot injection:** Introduction of a portion of a fuel before the main injection shortens the ignition delay. Therefore, the fuel burning in the premixed combustion decreases and the peak of heat release drops. This reduces the maximum cylinder temperatures thus NO_x emissions.

Reducing NO_x in this fashion will however cause a detrimental change in other performance parameters and emissions.

4.2 PM Formation

Particulates are typically carbonaceous matter on which some organic compounds are condensed or absorbed. Legally, particulate is anything in the exhaust stream that can be captured on a filter paper at 52 ^{o}C. Addition to the main PM constituent carbon, dust in air or inorganic material in the fuel may appear as particulates in exhaust (ash, oxides etc.). Trace metals from the engine wear may be carried by the lube oil and eventually migrate into exhaust. The sulphur in the fuel contributes the formation of sulphate particulates. In addition, soluble organic fraction, which is composed of lube oil derived hydrocarbons, contributes to PM. Loss of oil through the piston rings is the major source of oil consumption thus the SOF [1].

Soot production is inherent in the diesel combustion process. The highest particulate concentrations are present in the core region of fuel spray where local average equivalence ratios are high. Soot concentrations rise rapidly soon 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 [15]. Soot formation process starts with a molecule containing 12 to 22 carbon atoms and an H/C ratio of about 2 and ends up with particles typically a few hundred nanometers in diameter. At temperatures above 500 ^{o}C the individual particles are principally clusters of many small spheres of carbon with diameters of about 15 to 30 nm. As temperature decreases below 500 ^{o}C, the particles become coated with absorbed and condensed high molecular weight organic compounds including unburned hydrocarbons, oxygenated hydrocarbons and aromatic hydrocarbons.

Firstly, the condensed phase material arises from the fuel molecules by oxidation. These products typically include various unsaturated hydrocarbons. The condensation reactions of gas phase species leads to the appearance of first recognizable soot particles often called nuclei which have very small diameter (less than 2nm). After the formation of nuclei, the surface growth occurs. The bulk of solid phase material is generated and the gas phase species attaches to the surface of particles. The amount of soot is increased by the surface growth reactions but the number of particles remains same. On the other hand, in surface growth by coagulation, where the particles collide and coalse, number of particles decrease but the soot amount remains constant. After surface growth stops, aggregation of particles into chains and clusters continues.

At each stage in the soot formation process, oxidation can occur where soot is burned in the presence of oxidizing species to form gaseous products such as CO and CO_2. The emission of soot from the engine will depend on the balance between oxidation and formation processes. 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 mixes to the atmosphere. Figure 4.4 shows the particulate formation and oxidation process [15].

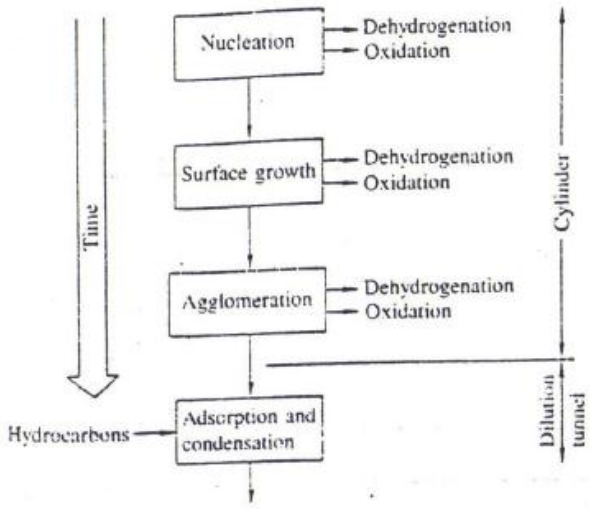

Figure 4.4 : Particulate formation and oxidation process

Generally, factors that reduce NO_x tend to increase PM emissions. The factors effecting PM emissions are listed below [1].

- **Mixing:** Since PM results from the heterogeneous combustion, increasing mixing increases the homogeneity of the mixture and reduces PM. This can be achieved through intake manifold, intake port, combustion bowl shape design.
- **O_2 concentration:** Increasing O_2 availability increases the soot oxidation thus reduces PM. Since EGR decreases the available O_2 in the cylinder, PM emissions are increased.
- **Cetane number:** Increasing the cetane number improves the evaporation thus the mixing process and reduces diffusion burning and PM formation.
- **Compression ratio:** In general, increasing compression ratio reduces PM emissions and improves fuel economy.
- **Residence time:** This gives additional opportunity for the incomplete combustion products to find lean pockets with which to mix. Providing more time can be achieved by reducing the engine speed.
- **Oil consumption control:** As previously mentioned, one of the constituents of PM is the lube oil that is loss through the piston rings. This can be reduced by a good cylinder honing process.

- **Fuel injection timing:** Retarded injection which is a common method for NO_x reduction increases PM missions.
- **Fuel injection pressure:** Fuel injection pressure effects atomization, penetration and mixing. So, increased fuel injection pressure decreases PM emissions due to better air utilization.
- **Charge air cooling:** Cooling the charge air decreases the charge density and allows more O_2 into the cylinder which increases the oxidation of soot. As the PM oxidation increases, more fuel is converted into power which means that the fuel is consumed more effectively.

4.3 CO Formation

Carbon monoxide is a toxic gas that is capable of killing humans by displacing oxygen in their bloodstream.

CO is an intermediate product in the combustion of hydrocarbons. There are two principle sources of CO in a diesel engine:

- Oxygen deficient combustion
- Dissociation of CO_2

Although overall the combustion in diesel engines runs in oxygen excess the combustion is heterogeneous therefore various areas are oxygen deficient. As combustion proceeds to completion, oxidation of CO to CO_2 occurs through the reactions between CO and oxidants. If the oxidants are not sufficient or the temperatures are low or there is less time left for CO to form CO_2, CO will be left without oxidation [1].

The methods for CO control are primarily improving mixture uniformity and leaning out the intake mixture. CO emissions during warm up are higher than the emissions in fully warmed up state due to low temperatures. Further, in transient engine operations during acceleration CO emissions become important.

4.4 HC formation

Hydrocarbon emissions are essentially the emissions of unburned fuel. If the air-fuel mixture is too lean to autoignite or to propagate flame inside the combustion

chamber, fuel leaves the combustion chamber unburnt contributing the HC emissions. Or another state that the fuel escapes the cylinder without burning is during the premixed combustion process when the air-fuel mixture is too rich to ignite to support a flame. This fuel can then be consumed only by slower thermal oxidation reactions later in the expansion process after mixing with additional air [15].

Figure 4.5 shows the HC formation process during the ignition delay and while the combustion is occurring respectively. Fuel injected during ignition delay will mix with air to produce wide range of equivalence ratios. Some will be leaner than the lean limit of combustion, some will be in combustible range and some will be too rich to burn. Locally overlean mixture will not autoignite resulting in slow reaction and products of incomplete combustion. Locally overrich mixture also can not ignite to support a flame propagation. Moreover quenching of the combustible mixture by thermal boundary layers reduces the combustible mixture leading to the products of incomplete combustion. During the combustion process, rapid oxidation of fuel or the products of fuel pyrolysis results in complete combustion. Slow mixing or lack of oxygen on the other hand causes locally overrich mixture thus incomplete combustion products.

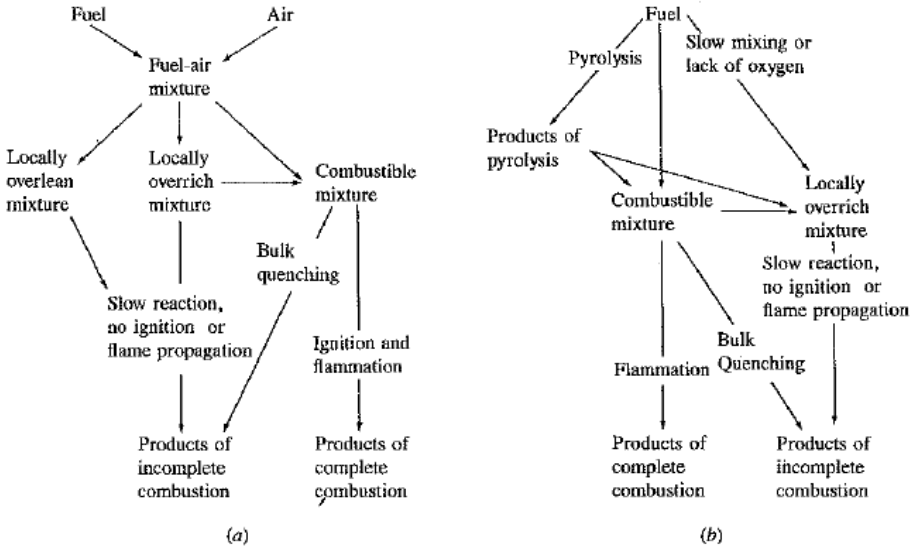

Figure 4.5 : HC formation process during the ignition delay and combustion

HC emissions depend on the operating conditions. At idle and low loads, HC emissions are much more than the full load operation. When the engine is overfuelled, HC emissions increase significantly.

Mechanisms of HC formation are summarized below:

4.4.1 Overleaning

As previously mentioned, fuel injected during ignition delay will mix with air to produce wide range of equivalence ratios. Figure 4.6 shows the equivalence ratio distribution at the time of ignition [15].

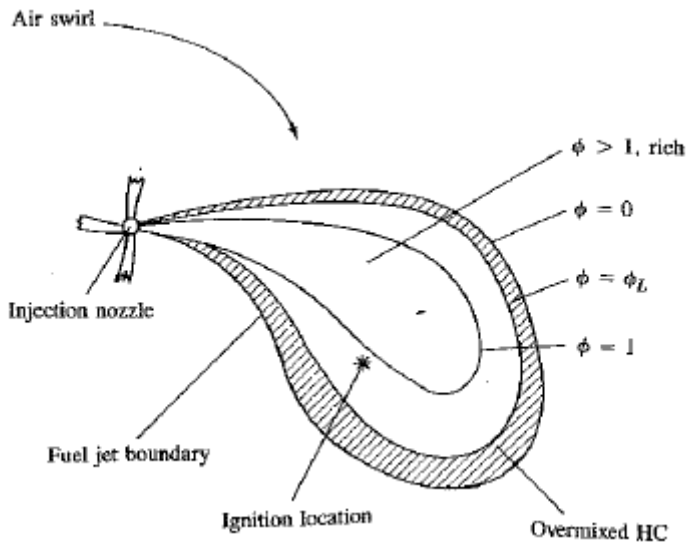

Figure 4.6 : Equivalence ratio distribution (ϕ_L~0.3 lean combustion limit)

In a swirling flow, ignition occurs in the slightly lean of stoichiometric region downstream of the spray core where the fuel which has spent most time within the combustible limits is located.

The fuel outside the lean combustion limit ($\phi=\phi_L$) will not autoignite. This mixture can only oxidize by relatively slow thermal oxidation reactions, which will be incomplete. The magnitude of the unburned HC depends of the amount of fuel injected during ignition delay and mixing rate with air during this period.

Overleaning of fuel injected during ignition delay is significant source of HC emissions especially where the ignition delay is long. Effect of ignition delay on HC emissions is given in Figure 4.7 [15].

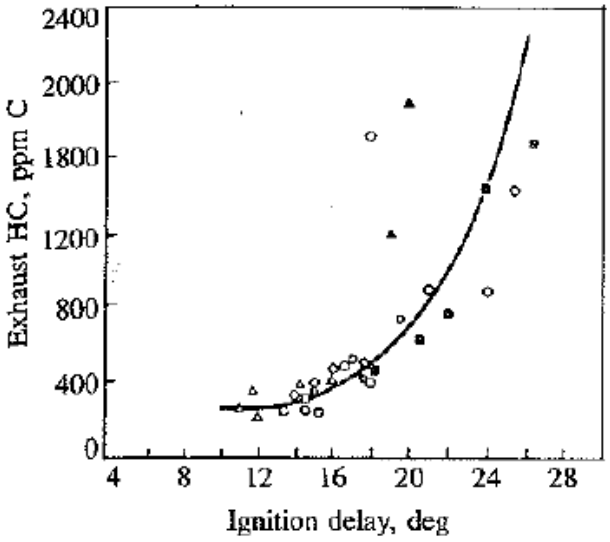

Figure 4.7 : Exhaust HC concentration with duration of ignition delay

4.4.2 Undermixing

There are two sources of fuel which enter the cylinder during combustion and which result in HC emission due to slow mixing with air. First one is the portion of fuel left in the injector nozzle sac volume. Second is the excess fuel that enters the cylinder under overfuelling conditions.

At the end of fuel injection, a small volume of fuel is left in the tip of injector. As the expansion process proceeds this fuel in the tip of the injector vaporizes and mixes with air. This mixing occurs relatively slow so fuel can escape the cylinder without burning. Figure 4.8 shows the effect of nozzle sac volume on HC emissions. The larger the nozzle sac volume, the higher the HC emissions.

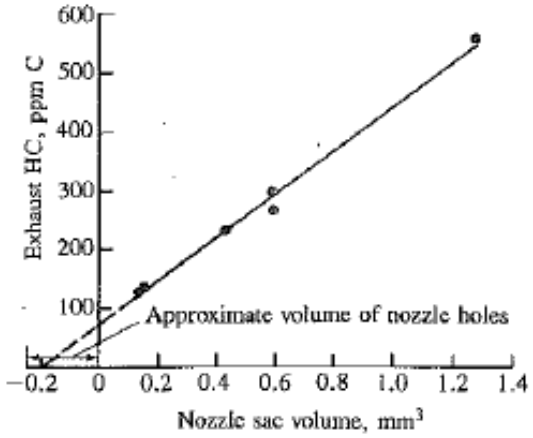

Figure 4.8 : Effect of nozzle sac volume on exhaust HC

Under transient conditions like accelerating, overfuelling can occur. The overall mixture will still remain lean but locally overrich conditions may exist through the expansion process which leads to incomplete combustion.

4.4.3 Quenching

Hitting of the fuel spray to the combustion chamber walls can also be a source for HC emission. This is important when the temperatures and pressures are low and injection timing is retarded.

The factors effecting HC emissions are listed below.

- **Engine load:** At low loads, less fuel will be injected into the cylinder and the possibility for the fuel spray to impinge the cylinder walls will be lower than the high load state. The HC emissions will be mostly originated by undermixing or lean flame out region. As the engine load increases, decreasing A/F ratio causes more fuel to deposited on the walls. The HC emissions will increase due to quenching. However, as the temperatures increases with increasing engine load, oxidation rates will be enhanced and HC emissions will be reduced.
- **Post injection:** During the expansion, gases are cooling down. When some amount of fuel is injected into cylinder during that period, excessive HC emissions may occur.
- **Cold start:** Lack of heat from the cylinder walls makes it difficult for fuel to evaporate. If the compression ratio is not sufficient, misfires can be experienced.
- **Charge pressure:** Increasing charge pressure increases swirl and mixing thus higher reaction and oxidation rates occur leading to lower HC emissions.
- **Fuel injection timing:** When the injection is advanced, more fuel will be introduced into the cylinder before ignition occurs. This will lead to a wider lean flame out region thus more HC emissions. This is shown in Figure 4.9. [1].

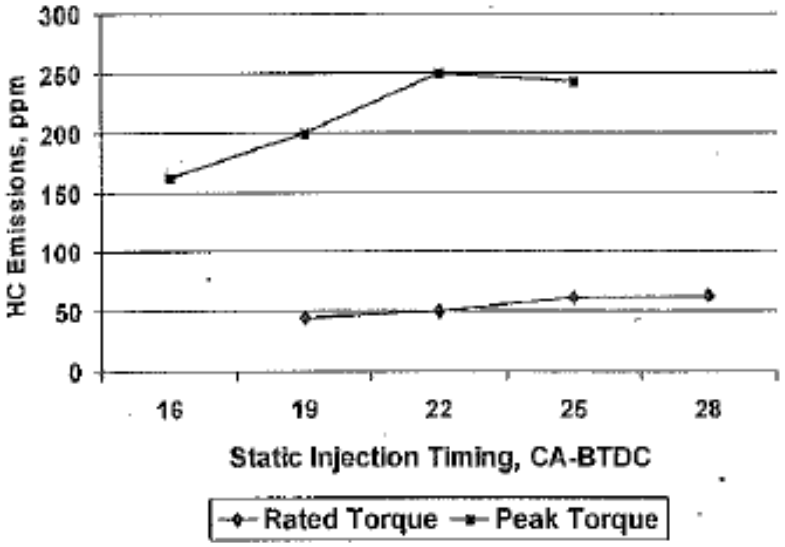

Figure 4.9 : Effect on injection timing on HC emissions

- **Fuel injection pressure:** Increased fuel injection pressure improves atomization. This widens the lean flame out region leading to higher HC emissions. Effect of injection pressure on HC emissions is shown in Figure 4.10 [1].

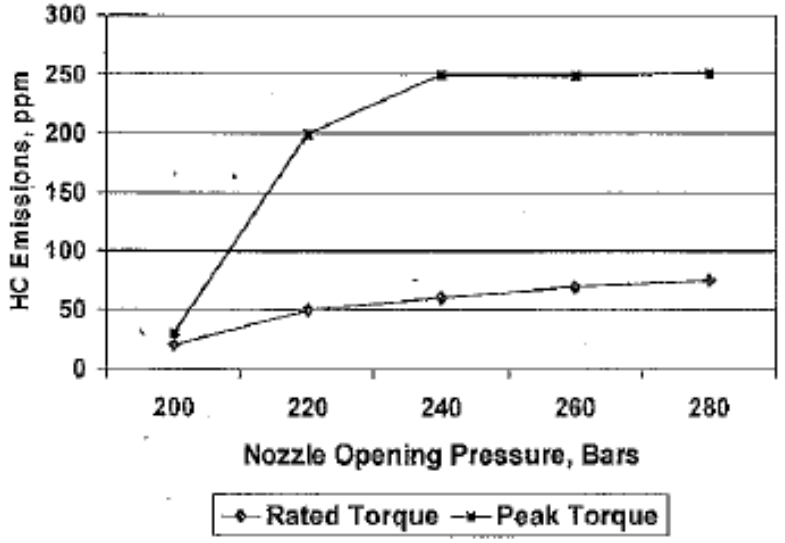

Figure 4.10 : Effect of nozzle opening pressure on HC emissions

5. EXPERIMENTAL WORK

5.1 Introduction

An experimental study has been conducted on a 4.4L diesel engine under two test points from the NEDC emission test cycle in order to quantify the effects of exhaust gas recirculation, fuel injection pressure, fuel injection timing and pilot injection quantity on NO_x, PM, HC and CO emissions.

5.2 Effects of Injection Strategy on Combustion and Emissions

5.2.1 Effect of fuel injection timing

To decrease NO_x emissions and meet increasingly stringent NO_x emission regulations, fuel injection timings of diesel engines have been retarded. However, this approach cannot be the ultimate solution due to the deterioration fuel economy and the exhaust gas emissions.

Injection retard (and hence combustion retard) reduces NO formation rate because it reduces peak cylinder pressures and burned gas temperatures. Advancing fuel injection extends the ignition delay. The reason for the increased ignition delay duration is that the fuel is injected into a lower pressure and temperature ambient. As the ignition delay increases, the greater portion of the fuel is injected during ignition delay so the fuel burning in the premixed combustion phase increases thus increasing the levels of NO_x. Additionally, with the earlier injection timing, the HC emissions are higher. The reason for this is the longer ignition delay period when the injection is advanced. This will allow more fuel vapor and smaller droplets o be carried away with the swirling air, producing wider lean flame out region, another factor increasing the HC emissions suggested to be the increase in fuel impingement on combustion walls [1].

Park et al. (2004) investigated the effects on combustion characteristics. It is reported that as the injection timing is retarded, due to the increased portion of the premixed burning, the most of the energy is released by premixed combustion and only small

portion of the energy is released by diffusion-controlled combustion. In addition, it is given that the in-cylinder temperature is increased at retarded injection timing, but the increase in NO_x is not observed due to the decrease of combustion period during expansion process. As shown in Figure 5.1, the combustion period decreased with retarded injection timing. The decreasing temperature due to the decrease of combustion period freezes the NO chemistry. The change of emissions with injection timing is shown in Figure 5.2 [20].

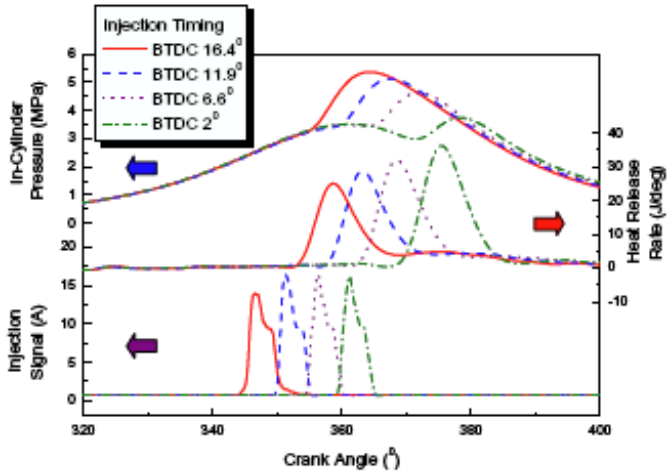

Figure 5.1 : Effect of main injection timing on combustion characteristics

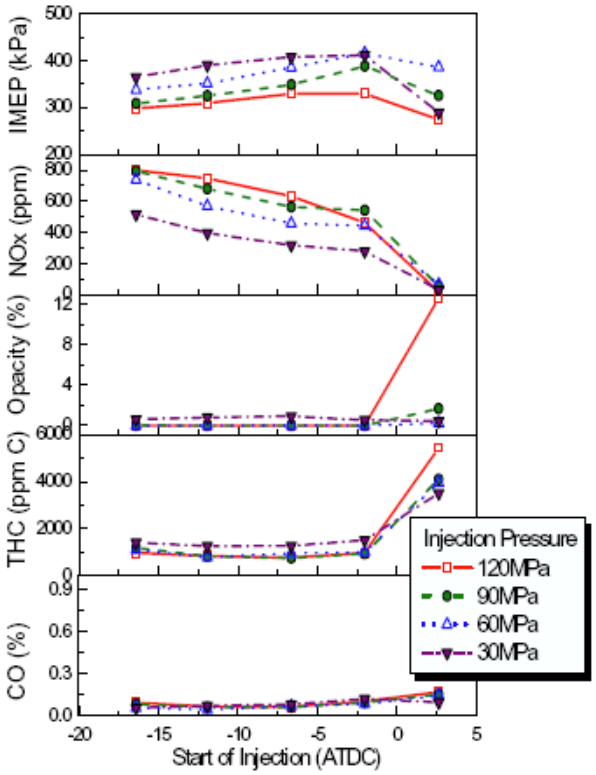

Figure 5.2 : Effect of main injection timing on emissions

5.2.2 Effect of fuel injection pressure

The effects of injection pressure on combustion and emissions have been investigated by Park et al. (2004) and it is given that the high injection pressure brings smaller fuel droplets and a shorter ignition delay. The reason for the shorter ignition delay period is the well-atomized spray and the promotion of fuel and air mixing. This activates combustion and reduces fuel consumption, PM and HC. On the other hand, activated combustion leads to noticeable increases in NO_x emissions and combustion noise. The effects of injection pressure on emissions and engine performance obtained in this study is shown in Figure 5.4. When the injection pressure is increased, indicated mean effective pressure decreases due to the improved combustion with smaller fuel droplets and shorter combustion period. With lower injection pressure, the injection time for the same amount of fuel is longer than that of high pressure injection, so the combustion period is increased. This produces more work during the expansion stroke thus IMEP increases. At higher injection pressure, the peak cylinder pressure and heat release is higher than low injection pressure state (Figure 5.3). These results showed the reason for IMEP increase owing to increased portion of work during expansion when the lower injection pressures are applied. It is also reported that the HC emissions increase with decreasing injection pressure. At the end of injection process, the fuel remaining in the injector sac volume vaporizes and enters the cylinder at low velocity. This mixes relatively slow with air and causes unburned fuel, which results in HC emissions. In case of a lower injection pressure, this effect is higher. Moreover, as discussed before, overleaning is also a source of hydrocarbon which Park et al. (2004) observed in their study [20].

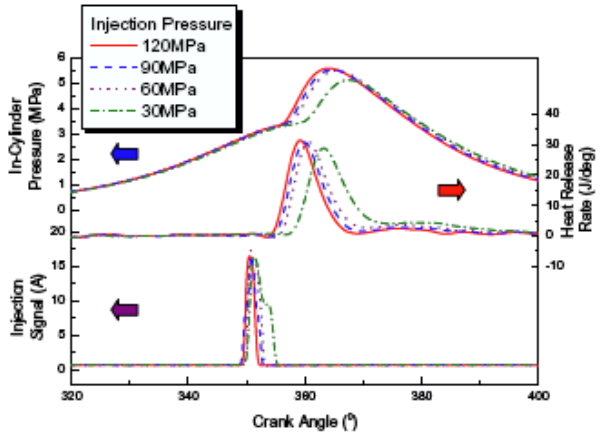

Figure 5.3 : Effect of fuel injection pressure on combustion characteristics

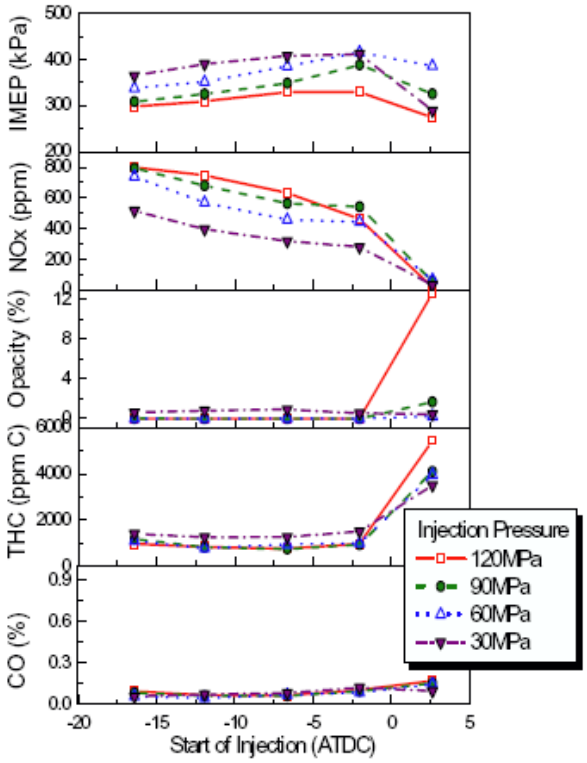

Figure 5.4 : Effect of fuel injection pressure on emissions

5.2.3 Effect of pilot injection

It is known that introducing a little amount of fuel into the cylinder before the main injection, decreases the ignition delay of the main injection. This allows to retard the main injection without effecting the ignition delay so that the NO_x emissions and combustion noise is suppressed. Effect of pilot injection on emissions and combustion characteristics are given in Figure 5.5 and Figure 5.6 respectively. Pilot injection provides a significant reduction in the peak of heat release rate. This is suggested to be due to the effect of pilot injection on shortening the ignition delay. This reduces the portion of premixed combustion and slows down the combustion rate. IMEP and smoke emissions are increased with pilot injection due to the extension of the diffusion controlled combustion duration. This is explained as that the smoke generation is increased by high temperatures in the fuel-rich zone during diffusion combustion. One way to reduce smoke emissions is shortening the diffusion combustion phase allowing less time for soot formation and more time for soot oxidation. It is clear from the Figure 5.6 that the maximum heat release rate is reduced with pilot injection and the diffusion combustion period is longer [20].

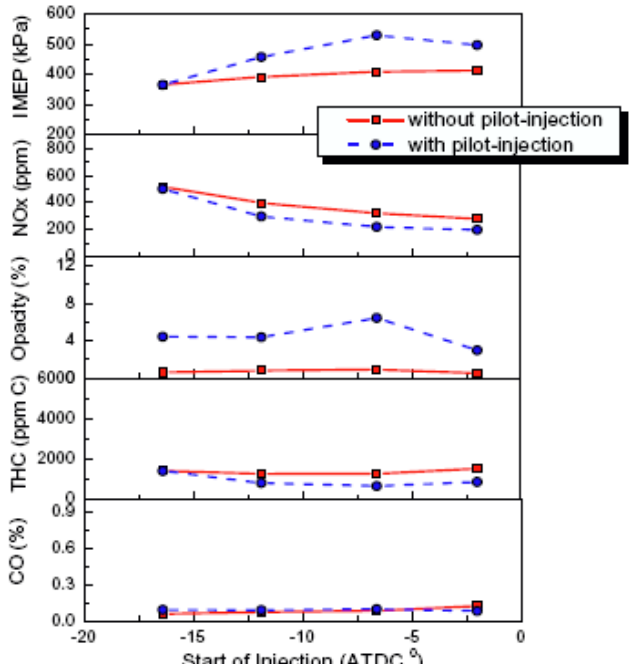

Figure 5.5 : Effect of pilot injection on emissions

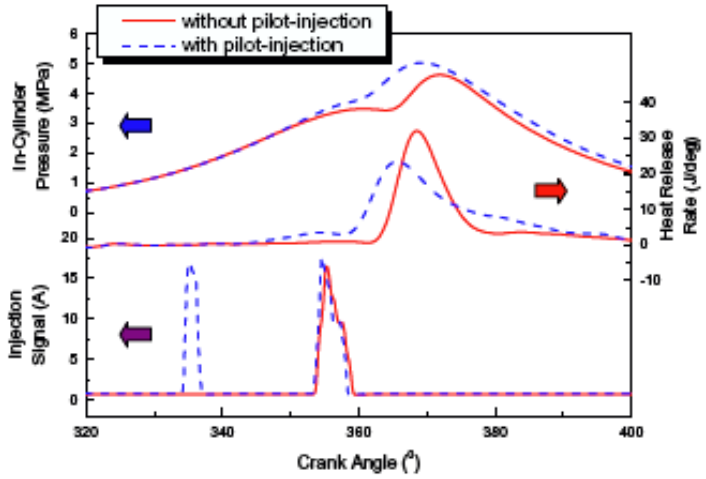

Figure 5.6 : Effect of pilot injection on combustion characteristics

In another study by Dronniou et al. (2005) the effects of pilot injection quantity and its timing is investigated. Three quantities of pilot injection expressed as a percentage of main injection is tested. As per this investigation, HC and CO emissions both increase with advancing the pilot injection timing. This is suggested to be due to the uncompleted oxidation of fuel during combustion.

It is also suggested that the increase of HC emissions with advanced pilot injection is due to the increased wall impingement and incomplete combustion of gases near cold walls. Advanced pilot injection timing promotes the location of HCs in top dead volumes and higher spray penetration promotes wall impingement and quenching.

All these mechanisms are consistent with results shown in Figure 5.7 since HC increases with increasing pilot quantity and advancing its timing.

On the other hand, increase of CO emissions with advanced pilot injection is suggested to be due to the influence of premixed combustion on final level of CO emissions. CO increase with advancing pilot injection timing is not a result of insufficient mixing since advancing pilot injection timing is likely to increase mixture homogeneity The simultaneous increase of HC and CO suggest the existence of regions when local temperature is too low to burn out intermediate species issued from premixed combustion.

CO reduces with increasing pilot injection quantity. This is suggested to be due longer time available for oxidation. It is clear that increasing pilot injection quantity improves oxidation of carbonated species.

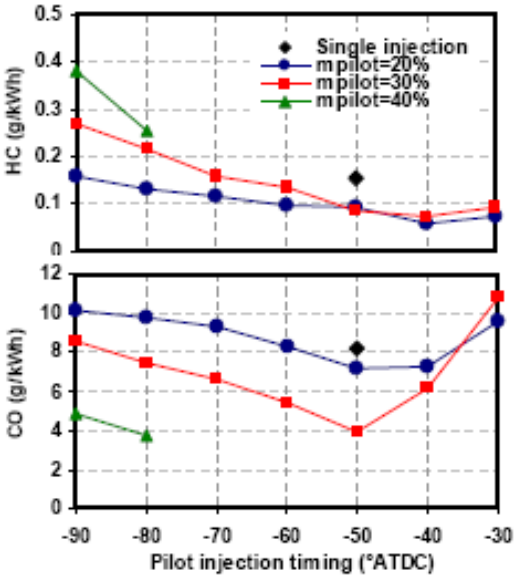

Figure 5.7 : Effect of pilot injection quantity and timing on HC and CO emissions

5.3 European Emission Regulations and Test Cycles

Due to the harmful effects of pollutants from motor vehicles, government mandates have made the output of those pollutants the most important factor in engine development. Each governing regulatory body has its own emission testing method such as the U.S. Federal Test Procedure (FTP) and New European Drive Cycle (NEDC).

NEDC is conducted on a chassis dynamometer. The cycle consists of four ECE segments followed by one EUDC segment. Before the test, the vehicle is held at a temperature between 20-30 ^{o}C for 6 hours. The cycle starts without an idling period that is engine starts at 0 s and the emission sampling begin at the same time. Before the year 2000, the engine was allowed to idle for 40 s. Before the collection of samples in the sampling bags, exhaust gas is diluted with ambient air. After the analyze of the samples, the emissions are expressed in grams per kilometer (g/km).

The ECE cycle (also known as Urban Driving Cycle UDC) represents the city driving conditions. The cycle points are low vehicle speed, low engine load and low exhaust temperature points. The speed profile is given in Figure 5.8. The Extra Urban Driving Cycle (EUDC) on the other hand is more aggressive and consists of high speed driving modes. The maximum speed of EUDC cycle is 120 km/h. For low power vehicles, the maximum speed in EUDC is 90 km/h. EUDC cycle speed profile for high power vehicles is shown in Figure 5.9. Summary of ECE and EUDC cycle parameters are given in Table 5.1 [1].

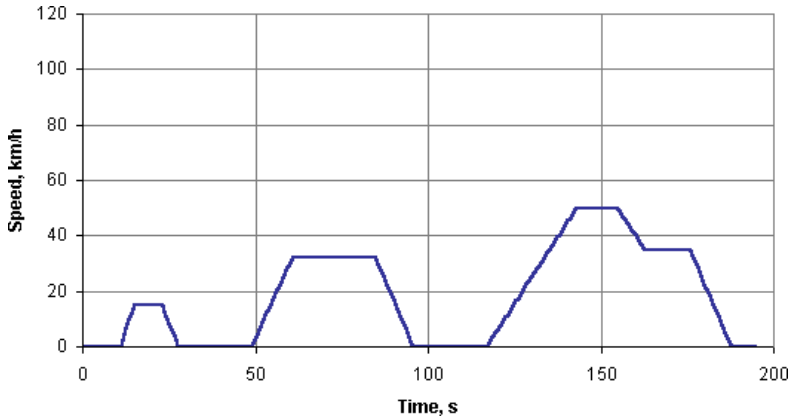

Figure 5.8 : ECE cycle [18]

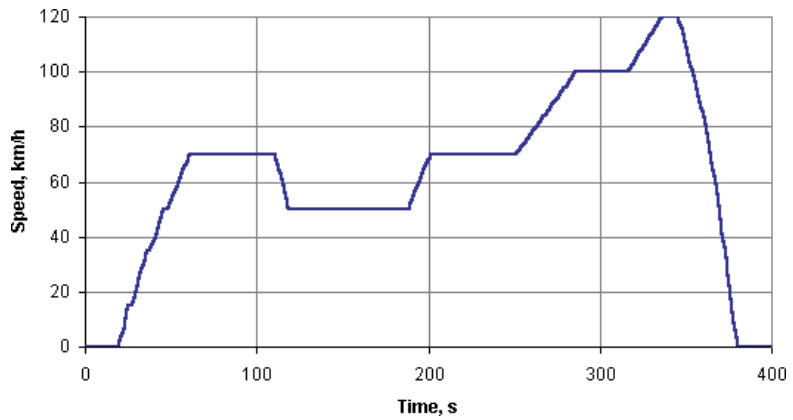

Figure 5.9 : EUDC cycle [18]

Table 5.1 : Summary of ECE & EUCD cycle parameters [1]

Characteristics	Unit	ECE	EUDC
Distance	km	4 x 1.013=4.052	6.955
Duration	s	4 x 195=780	400
Average speed	km/h	18.7 (with idling)	62.6
Maximum speed	km/h	50	120

The emissions limits are continuously being revised and new stringent emission standards are introduced. The European emission regulations are referred to as EuroI through EuroVI emission classes. EuroIV emission standard was in place for passenger cars (category M1) and light commercial vehicles (category N1 class I) since 2005. And since September 1st, 2009 EuroV applies. The EuroV limits reduce the emissions of particulates by 80% compared to EuroIV and this forces manufacturers to use diesel particulate systems as reduction through engine management is nearly impossible. EuroV is however more lenient as regards NO_x emissions, which is only has to decrease from 250 mg/km to 180 mg/km.

Beyond EuroV, EuroVI, which will set significantly lower emission limits for NO_x emissions will be introduced in September 2014. Table 5.2 shows the EU emission standards for passenger cars and light commercial vehicles [18].

Table 5.2 : EU emission standards for category M1 and N1 class I vehicles (g/km)

Tier	Date	CO	HC	HC+NOx	NOx	PM
Euro 1	1992.07 (M1) 1994.10(N1)	2.72	-	0.97	-	0.14
Euro 2	1996.01 (M1) 1998.01 (N1)	1.0	-	0.7	-	0.08
Euro 3	2000.01	0.64	-	0.56	0.50	0.05
Euro 4	2005.01	0.50	-	0.30	0.25	0.025
Euro 5	2009.09*	0.50	-	0.23	0.18	0.005
Euro 6	2014.09	0.50	-	0.17	0.08	0.005
*2009.09 is for new types, 2011.01 for all production						

5.4 Test Engine Specification

A 4.4L Euro5 direct injection diesel engine is used in dynamometer tests. It has a common rail fuel injection system, which has a capability of 2000 bars injection pressure. It has parallel sequential turbocharger system with hot side water-cooled

EGR valve. Engine specification is given on Table 5.3 and engine power torque curve is shown in Figure 5.10.

Table 5.3 : Test engine specification

4.4L Common Rail Diesel Engine	
Displacement	4367 cc
Number of cylinders	8
Valves per cylinder	4
Bore x stroke	84 mm x 98.5 mm
Compression ratio	16:1
Maximum power	225 kW @ 4000 rpm
Maximum torque	700 Nm @ 1500-3000 rpm
Maximum engine speed	4500 rpm

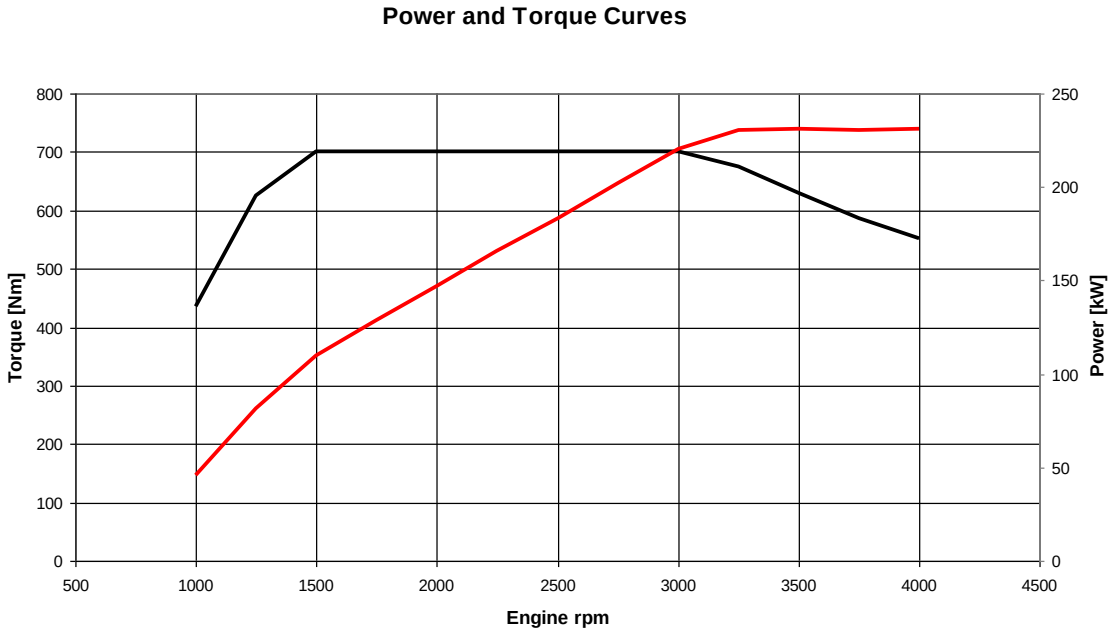

Figure 5.10 : Test engine power and torque curves

5.5 Test Equipment

Tests are conducted in Ford Otosan's Product Development Test Facility in Gölcük vehicle plant. Test equipment details are summarized below.

5.5.1 Engine dynamometer

APA engine dynamometer manufactured by AVL is used for the tests. APA is cradled AC machine with hydrostatic cradled bearings with squirrel cage rotor, equipped with load cell for torque reading. Dynamometer specification is given on Table 5.4.

Table 5.4 : Dynamometer specification

AVL APA ELIN Dynamometer	
Torque capacity	800 Nm constant up to 4000 rpm
Power capacity	300 kW between 4000-9000 rpm
Type of operation	Active and passive mode (for driving & braking)
Torque response	~ 10-20 ms for zero to maximum torque
Speed response	5000 rpm/sec for nominal torque
Inertia	0.2 kgm^2

5.5.2 Smoke meter

AVL 415S Variable Sampling Smoke Meter is used for soot measurement during the tests. The soot concentration in the exhaust can be displayed in mg/m^3 soot content, in % degree of filter paper blackening, or in FSN (Filter Smoke Number). The FSN indicates the degree of blackening of white filter paper on a scale ranging from 0 to 10. Exhaust gas is taken from the exhaust gas line with a probe and sucked through a filter paper. The blackening of the filter paper caused is measured with a reflectometer and indicates the soot content in the exhaust gas. The blackening of the exhaust gas depends on the soot concentration and the effective filter length (the length of the column of gas passed through the filter paper). The value '0' is assigned for clean filter paper and absolutely black paper is assigned to FSN value 10 [19]. A schematic of the measurement is shown in Figure 5.11.

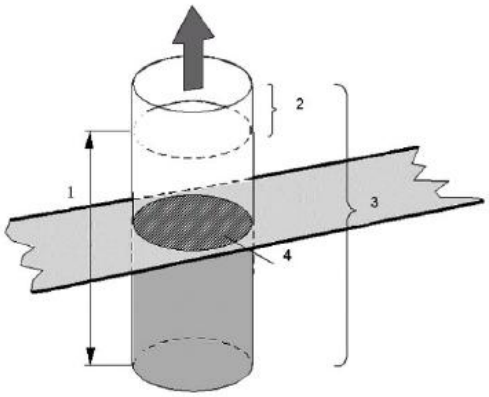

Figure 5.11 : Schematic of smoke measurement

5.5.3 Fuel mass flow meter and fuel conditioning system

AVL Fuel Mass Flow Meter 735S is used for fuel consumption measurements. It is capable of measuring fuel mass flow continuously in kg/h or l/h. It can measure fuel flows up to 125 kg/h or 165 l/h (at 0.75g/cm^3).

AVL Fuel Temperature Control system 753C which is integrated to the fuel mass flow meter is used to keep fuel temperature at 40 ^oC to prevent any variance on combustion due to changes in fuel inlet temperature. In this system, continuously recirculated chilled water is used for fuel cooling.

5.5.4 NO_x analyzer

HORIBA MEXA 1170HCLDA is used for NO_x measurements. This analyzer applies chemiluminescence (CLD) method for measurement. Chemiluminescence is a chemical reaction that produces light. When an NO reacts with ozone, it is oxidized to $NO_2[\Diamond]$ in an excited state. A small fraction of the molecules in this excited state decay by emitting a photon (i.e. giving off light) in the near infrared portion of the spectrum. Therefore, if one mixes a gas sample with ozone and measures the amount of light emitted, the concentration of NO_x in the sample can be determined.

$$NO+O_3 \rightarrow NO_2[\Diamond]+ O_2 \quad \textbf{(5.1)}$$

NO_2 in the sample is not activated by the ozone reaction, so any NO_2 in the sample is converted to NO in a converter before the ozone activation. The measurement range of the NO_x analyzer is between 0 to 10000 ppm with a NO_x converter efficiency of 95%.

5.5.5 Exhaust gas analyzer

Measurement of CO and HC emissions is done by MEXA 7100DEGR HORIBA exhaust gas analyzer. CO is measured by the non-dispersive infrared analyze method. A portion of the infrared radiation is absorbed by the component of interest in the gas sample. The quantity of infrared radiation that is absorbed is proportional to the CO concentration [20]. CO measurement range of the analyzer is between 0 to 5000 ppm.

HC is analyzed with a flame ionization detector (FID). FID analysis involves the detection of ions. The source of these ions is a hydrogen-air flame. The flame burns

at such a temperature as to pyrolyze the organic compounds, producing positively charged ions and electrons. In order to detect these ions, two electrodes are used to provide a potential difference. The ions thus are attracted to the collector plate and upon hitting the plate, induce a current. The current measured corresponds roughly to the proportion of reduced carbon atoms in the flame. The measurement range is between 0 to 20000 ppm.

5.6 Selection of Test Points

Two different test points are selected for the study. As a beginning, the load and rpm points corresponding to the NEDC (New European Driving Cycle) test cycle has been run in engine dynamometer. During the engine run, instantaneous NO_x emissions are recorded.

The two points which are contributing most to the NO_x emission are selected. These two points are found to be the 70 km/h and 120 km/h vehicle speed points, which are contained in EUCD part of the NEDC. The corresponding engine test points to these vehicle speeds are:

- 1250 rpm 22% load (vehicle speed 70 km/h)
- 1570 rpm 37% load (vehicle speed 120 km/h)

After the determination of the test points, for the set up of the design of experiments, engine is run at these two points in steady state conditions and base calibration values are recorded.

The test parameters were EGR ratio, fuel injection pressure, main injection timing and pilot injection volume. Based on the base calibration values, four different values of each parameter are selected. Those include the base calibration value itself and three other points, including values higher and lower than the base calibration. These are shown in Table 5.5 and Table 5.6 for the two test points respectively. In this study, the EGR ratio is defined as the percentage of recirculated exhaust gas in the total intake charge on a mass basis, and is calculated by measuring the oxygen and carbon dioxide concentrations in the intake and exhaust gases, assuming complete combustion of the injected fuel.

Table 5.5 : Design of experiment for test point 1

Test point 1		Base value	Test values
Vehicle speed	[km/h]	70	
Engine speed	[rpm]	1250	
Brake torque	[Nm]	143	
Load	[%]	22	
Main injection timing	[^{o}BTDC]	-5.65	(2) - (-1) - (-5.65) - (-8)
Fuel injection pressure	[MPa]	80	(60) – (80) – (100) – (120)
Pilot injection volume	[mg/stroke]	1.75	(1.25) – (1.50) – (1.75) – (2.00)
EGR ratio	[%]	37	(0) – (15) – (30) – (52)

Table 5.6 : Design of experiment for test point 2

Test point 2		Base value	Test values
Vehicle speed	[km/h]	120	
Engine speed	[rpm]	1570	
Brake torque	[Nm]	263	
Load	[%]	37	
Main injection timing	[^{o}BTDC]	-3.25	(2) - (-1) - (-3.25) - (-5)
Fuel injection pressure	[MPa]	120	(90) – (120) – (140) – (160)
Pilot injection volume	[mg/stroke]	1.25	(1.00) – (1.25) – (1.65) – (2.20)
EGR ratio	[%]	34	(0) – (15) – (30) – (45)

5.7 Test Procedure

During the emission tests, only one parameter with EGR ratio is changed while others kept at base calibration value. To prevent any variance, test room air is conditioned at 25 °C while engine clean air temperature was kept at 18 °C and fuel temperature at 40 °C.

An emulated ECU is used to change the engine control variables in real time. After changing of each variable and before taking the measurements, engine is stabilized for two minutes before 30 seconds of data recording for each test point.

5.8 Results and Discussion

5.8.1 Effect of main injection timing and EGR (1st point)

In this part of the study, start of main injection timings 2° BTDC, 1° ATDC, 5,65° ATDC (base calibration) and 8° ATDC with varying EGR ratios 0%, 15%, 30% and 52% for each injection timing are tested. The results obtained are shown in Figures A1, A2, A3, A4, A5, A6, A7 and A8.

As expected when the main injection is retarded, the NO_x emissions decrease due to the reduced peak cylinder pressure and burned gas temperatures. When the fuel injection is advanced, ignition delay is extended and greater portion of the fuel is injected during this period. This increases the premixed combustion and causes increase in NO_x emissions.

On the other hand, when the injection is retarded, the PM emissions are decreased. This is opposed to the expected trend. Normally when the start of injection is advanced before TDC, due to the low temperatures at the beginning of injection, ignition delay period will be longer and the amount of fuel burning at vapor phase will increase. This will decrease PM emissions. At around TDC, the temperatures will not change and advancing around TDC is expected not to effect PM emissions. With very late injection timings, due to the low temperatures PM emissions may decrease. The soot formation in this condition is suppressed because the combustion temperatures are at the lower side of the soot formation peninsula. Since this test point is a low load point and the main injection timings are late after TDC, as the injection timing is retarded, PM emissions decreased due to the reasons explained.

CO emissions are increased with the retardation of the injection. This is due to the effect of reduced oxidation resulting from the decreased in-cylinder pressure and temperatures. Another effect in increased CO may be the less time remaining for the oxidation of CO to CO_2, due to the retarded combustion.

HC emissions are also increased when the injection is retarded. HC was expected to increase with advancing due to the wider lean flame out region produced because of the longer ignition delay which allows more fuel vapor and smaller droplets to be carried away with swirling air. There will be more fuel in the cylinder before ignition and the air-fuel mixture will be too rich to ignite to support a flame. In this case, as

similar to the mechanism given for CO, due to the decreasing temperatures with retarded main injection timing, HC burning is decreased.

When the injection timing is retarded, the torque that can be obtained from a fixed amount of fuel decreases thus increasing the specific fuel consumption. The reason for the reduction in torque is shifting of combustion to the expansion period.

When the effects of EGR are considered, as expected PM increases with increased rate of EGR and NO_x decreases significantly. The increase in PM emissions is more significant at high (%50) EGR rates. This is due to the decreased oxidation of soot at low in cylinder temperature associated with increased rate of EGR.

The CO emissions stay relatively low until about 30% EGR rate .But it increases considerably at higher EGR rates due to decreased rate of O_2 thus obstructed oxidation of CO to CO_2. This trend is due to the decrease in the air/fuel ratio i.e. the formation of richer mixtures and the increased recirculation of CO from the previous engine cycle as a result of EGR.

HC emissions increase with increasing EGR, however, relatively constant levels are found up to 30%. At high EGR rates the combustion efficiency declines and hence the HC emissions increase; this deterioration of combustion is caused by the reduction of inlet O_2 concentration and also the downward travel of the piston causes cooling of the in-cylinder gases which tends to increase HC emissions.

The mass of injected fuel for each stroke was kept constant during the experiments. The specific fuel consumption which is the mass of fuel needed to obtain a given engine torque has been increased with increasing levels of EGR. This is because of the decreasing engine torque resulting from the decreased maximum cylinder pressure.

5.8.2 Effect of main injection timing and EGR (2nd point)

In this part of the study, start of main injection timings 2° BTDC, 1° ATDC, 3,25° ATDC (base calibration) and 5° ATDC with varying EGR ratios 0%, 15%, 30% and 45% for each injection timing are tested. The results obtained are shown in Figures A9, A10, A11, A12, A13, A14, A15 and A16.

All the emission and performance trends are similar to that of first test point. Only the PM emissions trend is different. This time with the retarded main injection

timing, PM emissions increase. The reason for the opposed trend compared to first test point is the increase of temperatures with increased engine load and disappearing of the low PM formation at low temperatures.

5.8.3 Effect of fuel injection pressure and EGR (1st point)

In this part of the study, fuel injection pressures of 60MPa, 80MPa (base calibration), 100MPa, 120MPa with varying EGR ratios 0%, 15%, 30% and 52% for each injection pressure are tested. The results obtained are shown in Figures A17, A18, A19, A20, A21, A22, A23 and A24.

As previously discussed, higher injection pressures produce faster fuel jets at the nozzle exit, increase spray penetration, reduce Sauter Mean Diameter and improve mixing. With smaller fuel particles, combustion efficiency and combustion temperatures are higher. These factors increase the premixed combustion fraction, which produces high NO_x and less soot. The reduction in PM at higher injection pressure is due to the improved mixing and homogeneity of the charge. Another factor in PM reduction could be the reduction of the fuel fraction injected into the flame. In other words, most of the fuel is injected during the premixed combustion period.

When the injection pressure is decreased, lower engine torque is obtained for a fixed amount of fuel. The reason for the decrease in torque with lower fuel injection pressures is shifting of combustion towards the expansion period resulting from the longer ignition delay.

At lower injection pressures, CO emissions are higher. Reduced spray penetration would reduce the mixture formation rate and suppress leaning of the mixture causing increase in CO emissions.

HC emissions on the other hand, decrease first with increasing the injection pressure but at the highest injection pressure, it is slightly increased. The first part can be explained such that the improved atomization lowers the HC emissions and another contributor to this characteristic can be the fuel remaining in the injector nozzle. (nozzle sac volume) As the expansion proceeds this fuel vaporizes and mixes with air slowly causing the fuel to escape cylinder without burning. At lower injection pressures this phenomena is more dominant. The reason for the increased HC emission at the highest injection pressure can be explained such that due to the high

injection pressure, the flame lean out region widens causing higher HC emissions or in other words, lean regions become even leaner. Another reason for this kind of trend can be wall wetting because of the high injection pressure. (Also at low injection pressures ignition delay is long and overleaning occurs.)

The effects of EGR are similar to that of discussed in previous section. As a general trend, EGR decreases NO_x emissions while increasing other emissions and specific fuel consumption. There is a slight increase in HC with increased EGR ratio while CO is increasing significantly at 45% EGR due to the O_2 deficiency.

5.8.4 Effect of fuel injection pressure and EGR (2nd point)

In this part of the study, fuel injection pressures of 90MPa, 120MPa (base calibration), 140MPa, 160MPa with varying EGR ratios 0%, 15%, 30% and 45% for each injection pressure are tested. The results obtained are shown in Figures A25, A26, A27, A28, A29, A30, A31 and A32.

At this point, NO_x, PM and CO characteristics are similar to the first test point. HC emissions this time do not increase at the highest fuel injection pressure. It seems that at this point, wall wetting does not occur.

5.8.5 Effect of pilot injection volume and EGR (1st point)

In this part of the study, pilot injection volumes of 1.25mg/str, 1.50mg/str, 1.75mg/str (base calibration), 2.00mg/str with varying EGR ratios of 0%, 15%, 30% and 52% for each pilot injection amount are tested. The results obtained are shown in Figures A33, A34, A35, A36, A37, A38, A39 and A40.

It is a well-known method that pilot injection has a reducing effect on NO_x emissions. The pilot injection shortens the ignition delay due to the increased gas temperature and the combustion of pilot fuel. In this part of the experiments, the effect of pilot injection quantity is investigated.

It is observed that increasing the pilot injection volume decreases NO_x emissions. This is expected to be due to the suppressing of the sharp pressure rise in the cylinder. PM emissions on the other hand are increased with increasing pilot injection amount. This is due to the extension of the diffusion controlled combustion duration.

The fuel consumption on the other hand is reduced. This might be due to a slow rate of pressure rise in the initial combustion, which resulted in a higher mechanical efficiency and a lower cooling loss of the engine.

HC and CO emissions are decreased with increasing pilot quantity. This can be due to the longer time available for the oxidation and higher temperatures at the beginning of main injection combustion.

The effects of EGR are similar to that of observed in previous tests.

5.8.6 Effect of pilot injection volume and EGR (2nd point)

In this part of the study, pilot injection volumes of 1.00mg/str, 1.25mg/str (base calibration), 1.65mg/str, 2.20mg/str with varying EGR ratios of 0%, 15%, 30% and 45% for each pilot injection amount are tested. The results obtained are shown in Figures A41, A42, A43, A44, A45, A46, A47 and A48.

For this point, PM, CO and HC emissions are different to that of observed for the first point. With the increasing amount of pilot injection, PM, CO and HC remained nearly constant. It can be concluded that the increase in amount of pilot injection was not sufficient to observe the effect on PM, CO and HC emissions.

CONCLUSIONS AND RECOMMENDATIONS

An experimental study has been conducted to examine the effect of EGR combined with variation of the main fuel injection timing, injection pressure and pilot injection amount. From the results, the following conclusions were derived.

- An increase in EGR rate results in a global decrease of NO_x emissions with penalty on SFC and PM emissions. The low oxygen available for combustion resulted in longer ignition delay periods that shifts the whole combustion process towards the expansion stroke. This results in combustion gases spending shorter periods at high temperatures leading to lower NO_x emissions and lower soot oxidation.
- Although EGR increases the ignition delay period, which leads to higher rates of premixed burning, the lower oxygen availability for combustion reduces the peak rate of premixed burning which reduces combustion temperatures. This consequently lowers NO_x and increase PM due to increase in diffusion burning resulting from the reduction in premixed burning.
- There is a slightly increasing trend for HC and CO with high rates of EGR. This is due to the decreased rate of O_2 thus obstructed oxidation of CO and HC.
- Drop in maximum pressures in cylinder resulting from the increased rate of EGR, leads to decrease in engine torque, which consequently increases the specific fuel consumption.
- Investigation on the effects of main injection timing showed that very low NO_x emissions are possible with retarded injection timing, which in turn increases specific fuel consumption. PM emissions on the other hand are mostly dependent on the cylinder temperatures, which effect the formation and oxidation of soot. Opposite trends for soot are observed in this study, which suggest that even there are general trends for emissions characteristics, the best approach is to test to ensure the characteristic at a specific point.

- Retarded injection timing lead to an increase in HC and CO for both points which is thought to be resulted from the reduced in-cylinder temperatures.
- It is observed that the effect of injection timing on NO_x and PM is less observable at high EGR rates since a significant amount is reduced by means of EGR.
- Combination of timing retard with EGR lead to significant deterioration in engine torque.
- Investigation on fuel injection pressure showed that reduction in NO_x is possible with lower injection pressure, which on the other hand increases PM emissions due to worse mixture formation.
- Decreased fuel injection pressure leads to deterioration in engine torque due to shifting of combustion to expansion stroke.
- As a general trend, lower HC and CO emissions are observed with higher injection pressure, which is brought by the improved mixing. Only for the low load test point, HC was increased at the maximum injection pressure point. The reason behind this is thought to be the increase in LFOR or in other words, leaner regions became even leaner with high injection pressures.
- Pilot injection is effective in shortening the ignition delay, which reduces the portion of premixed combustion, thus lowers the NO_x emissions. Conversely, PM emissions increased which is caused by the increase in diffusion-controlled combustion. The trend of emissions was not observable at the 37% load point. It is thought that, for this test point, the increase in amount of pilot injection was not sufficient for observable changes in emissions.
- HC and CO emissions showed decreasing trend with an increase in pilot injection amount.
- The specific fuel consumption is reduced when the pilot injection volume is increased. This might be due to a slow rate of pressure rise in the initial combustion, which resulted in a higher mechanical efficiency, and a lower cooling loss of the engine.
- The optimized fuel injection strategy brings impressive advances in engine performance and emissions profile. However, further improvements are

required to achieve very low NO_x emissions together with low PM emissions. Combining various after-treatment techniques such as LNT, SCR and DPF can further enhance the chance for a future low emission diesel engine.

- Due to time constraints during experimentation, all measurements could not be conducted at once. It is best to complete the tests in a short time period to prevent any variance, which consequently may affect measurement results.

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APPENDIX

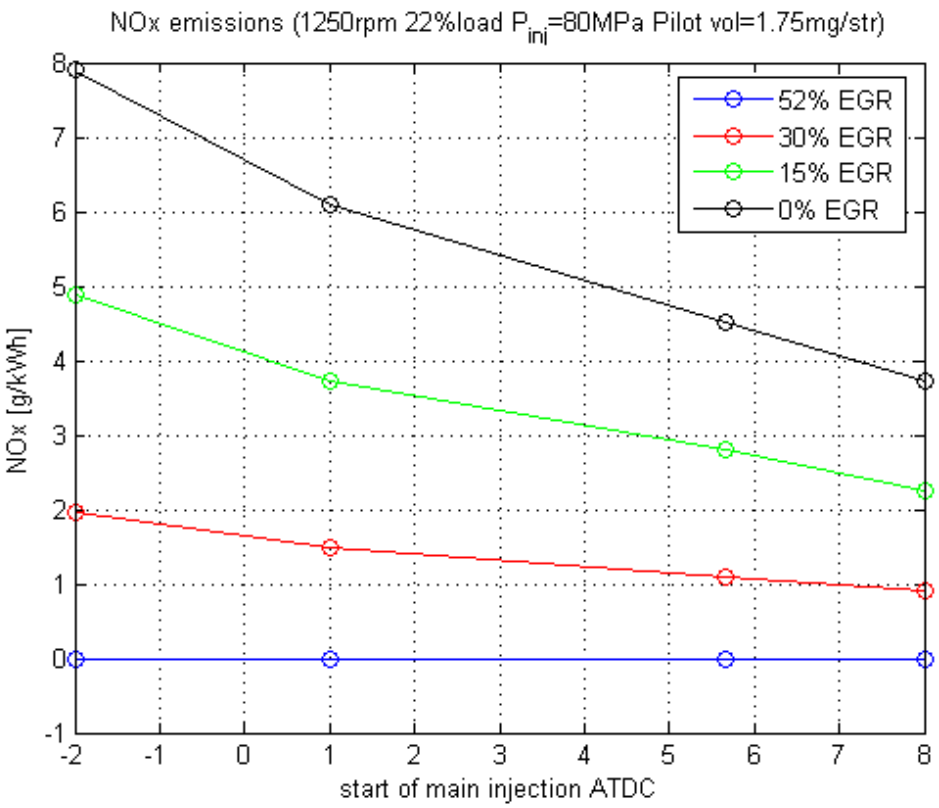


Figure A.1 : NO_x emissions – Effect of main injection timing – Test point 1

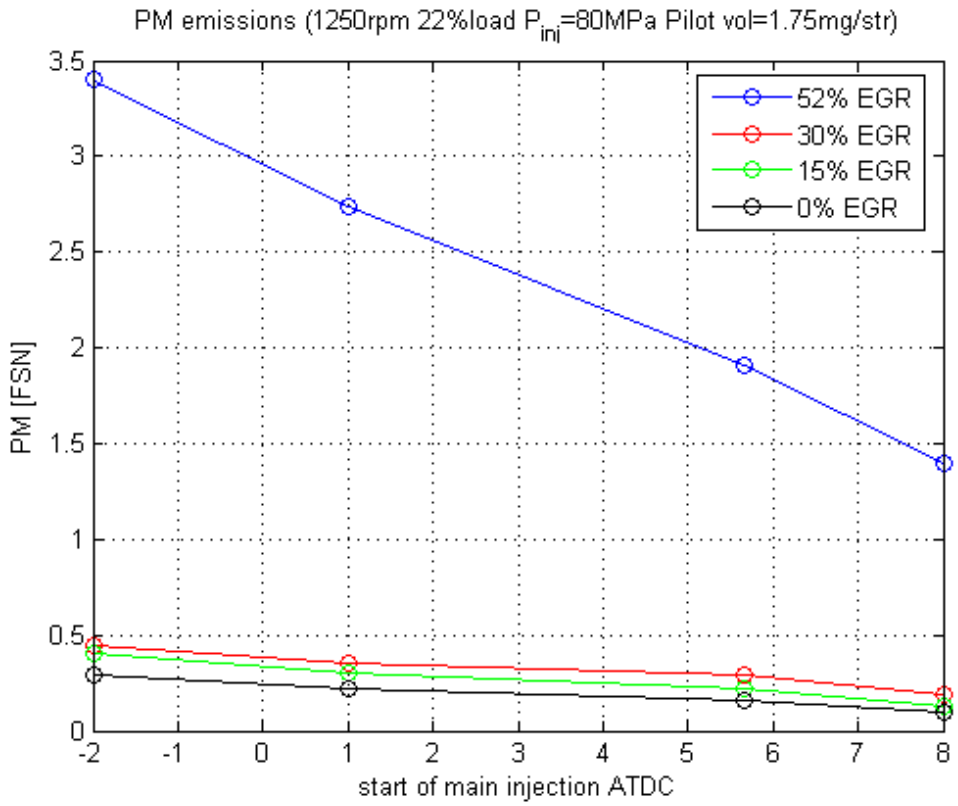


Figure A.2 : PM emissions – Effect of main injection timing – Test point 1

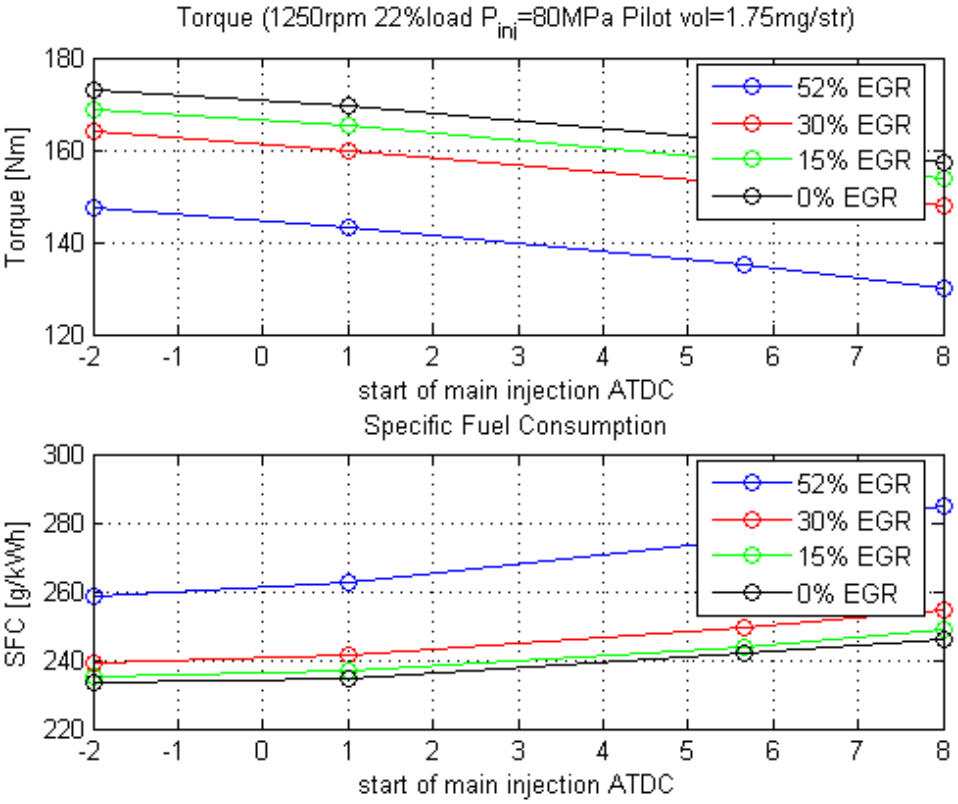


Figure A.3 : Torque and SFC – Effect of main injection timing – Test point 1

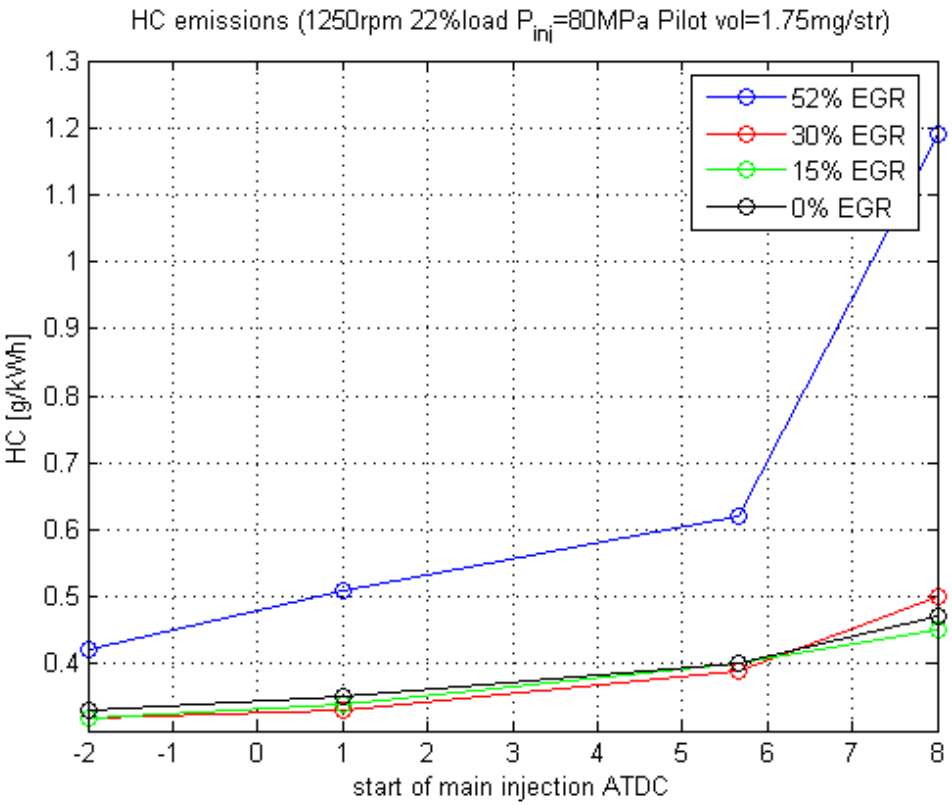


Figure A.4 : HC emissions – Effect of main injection timing – Test point 1

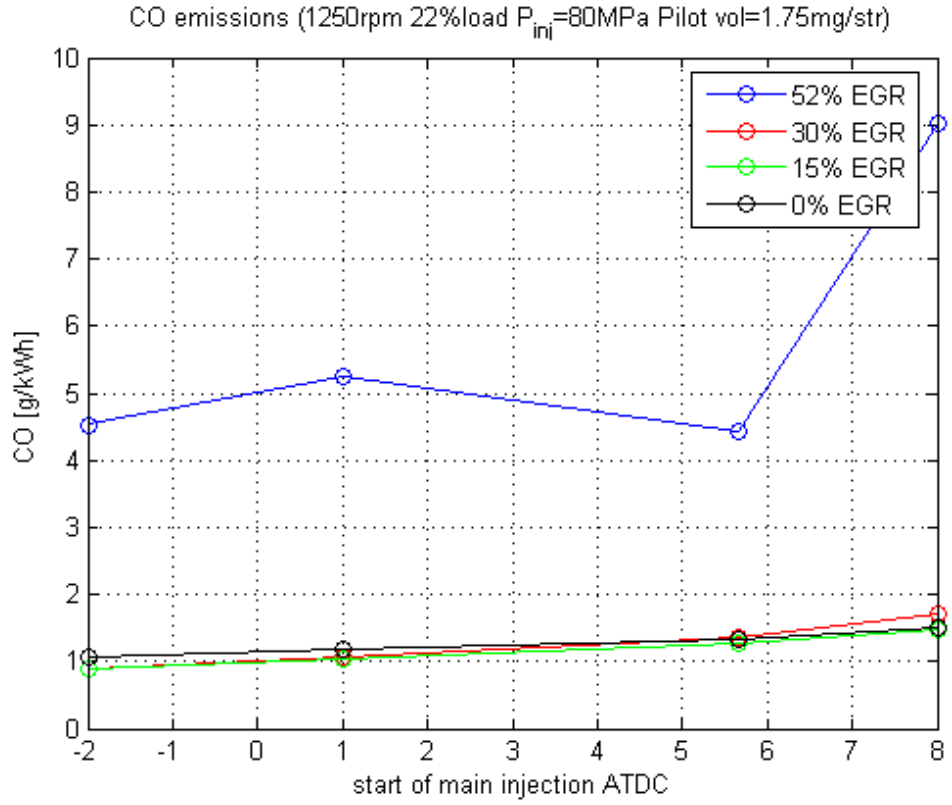


Figure A.5 : CO emissions – Effect of main injection timing – Test point 1

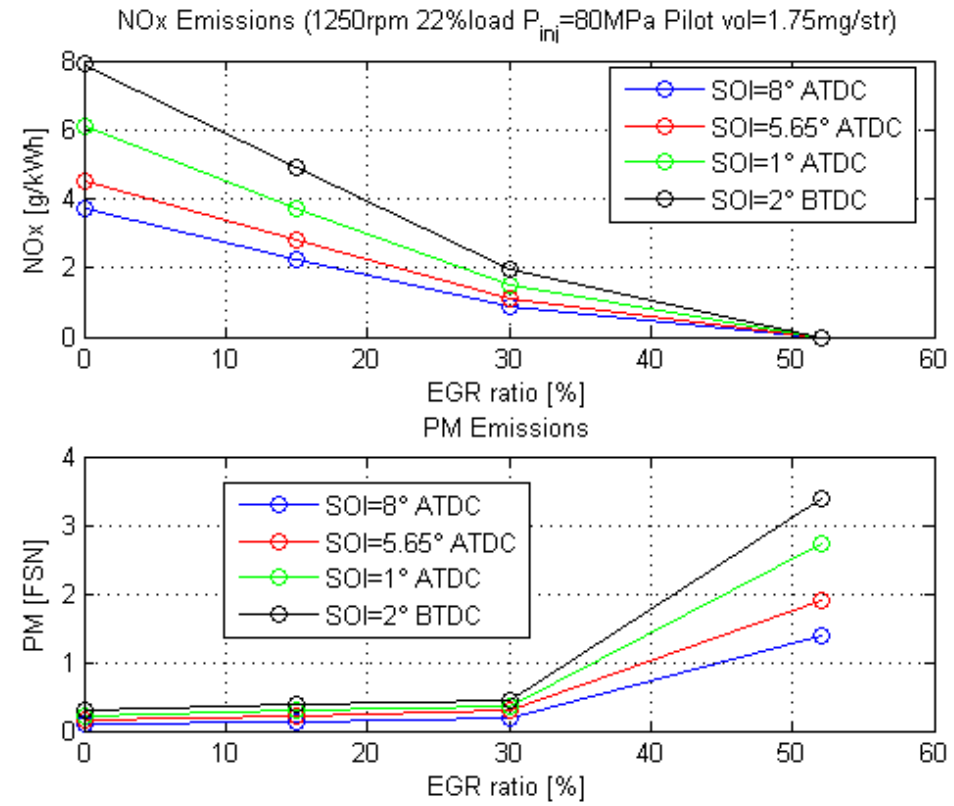


Figure A.6 : NO_x and PM emissions – Effect of EGR – Test point 1

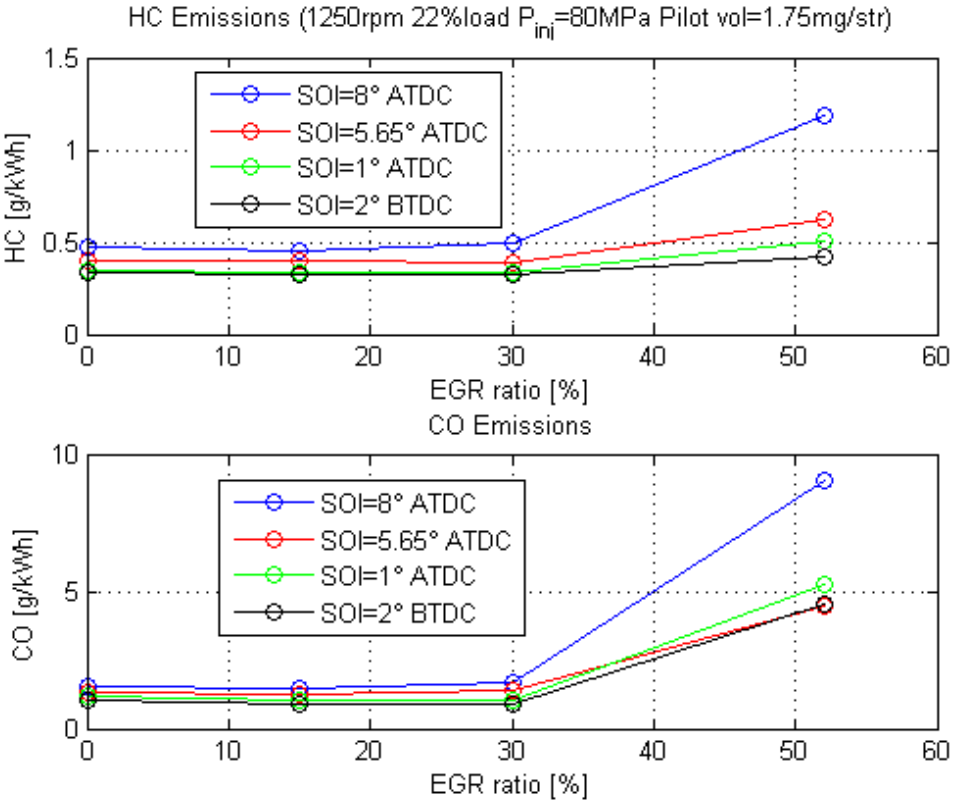


Figure A.7 : HC and CO emissions – Effect of EGR – Test point 1

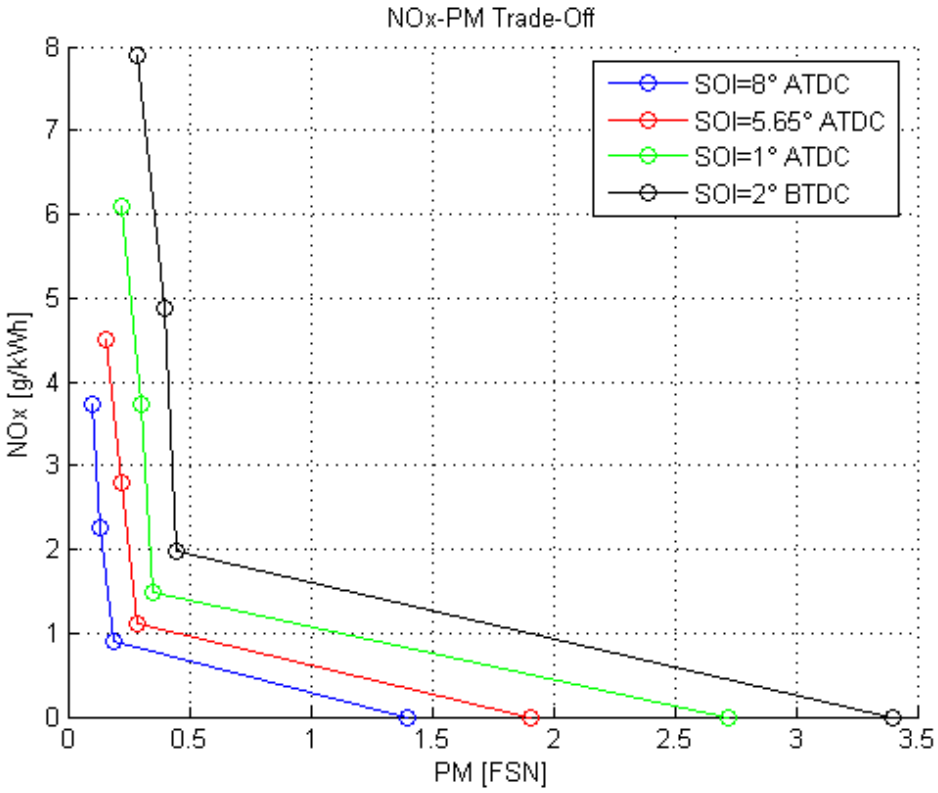


Figure A.8 : NO_x - PM trade off - Test point 1

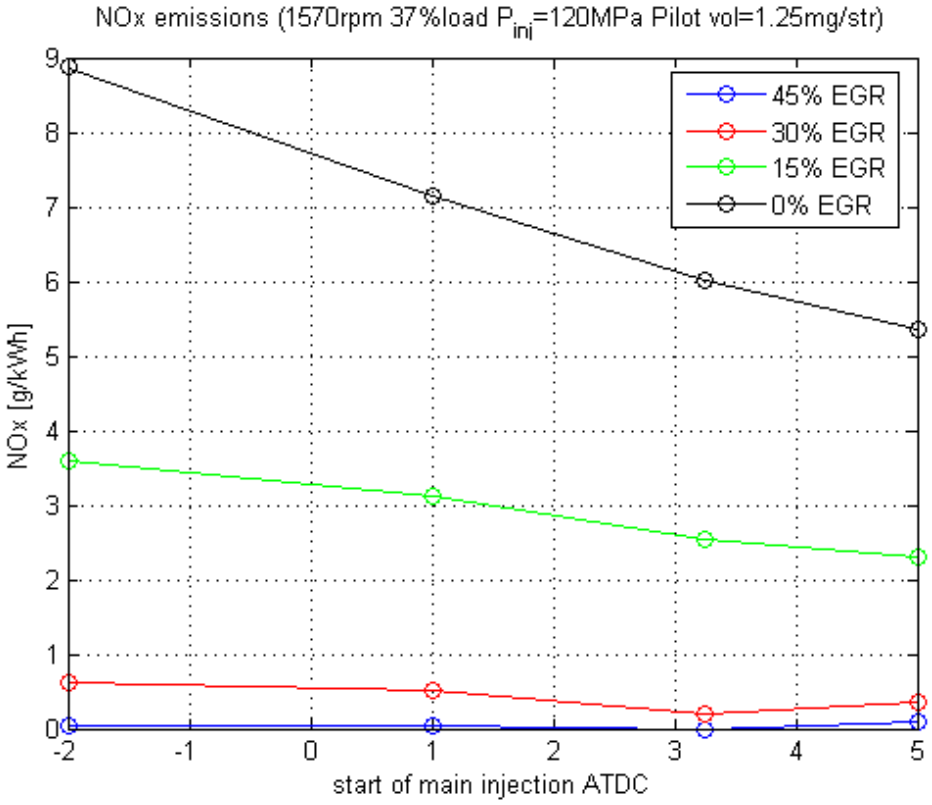


Figure A.9 : NO_x emissions – Effect of main injection timing – Test point 2

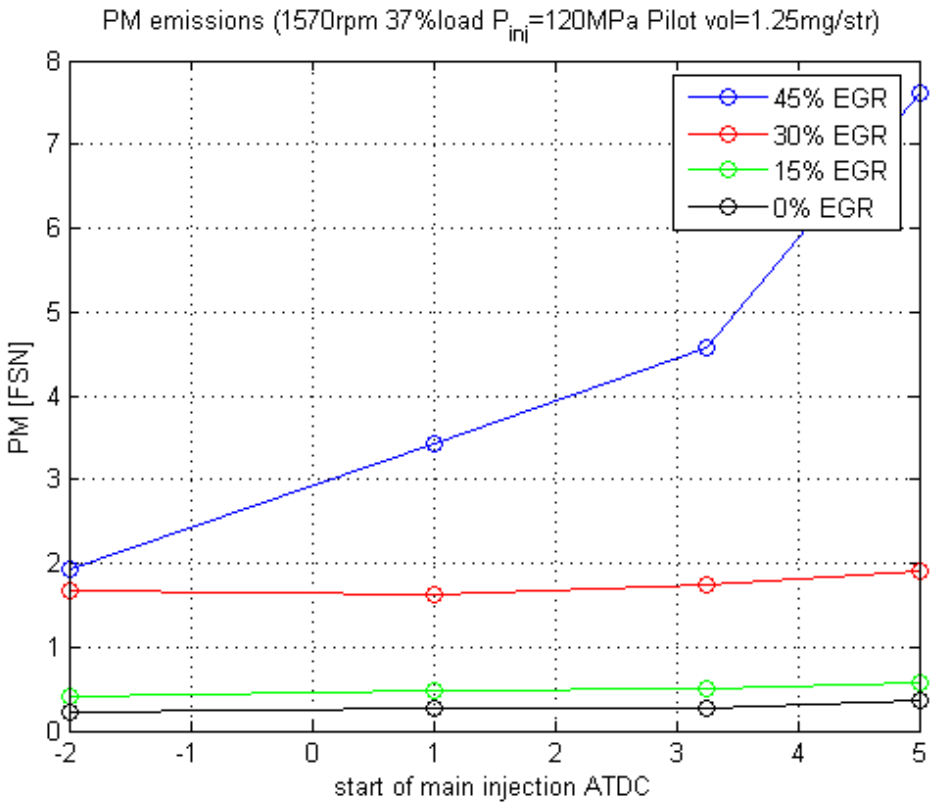


Figure A.10 : PM emissions – Effect of main injection timing – Test point 2

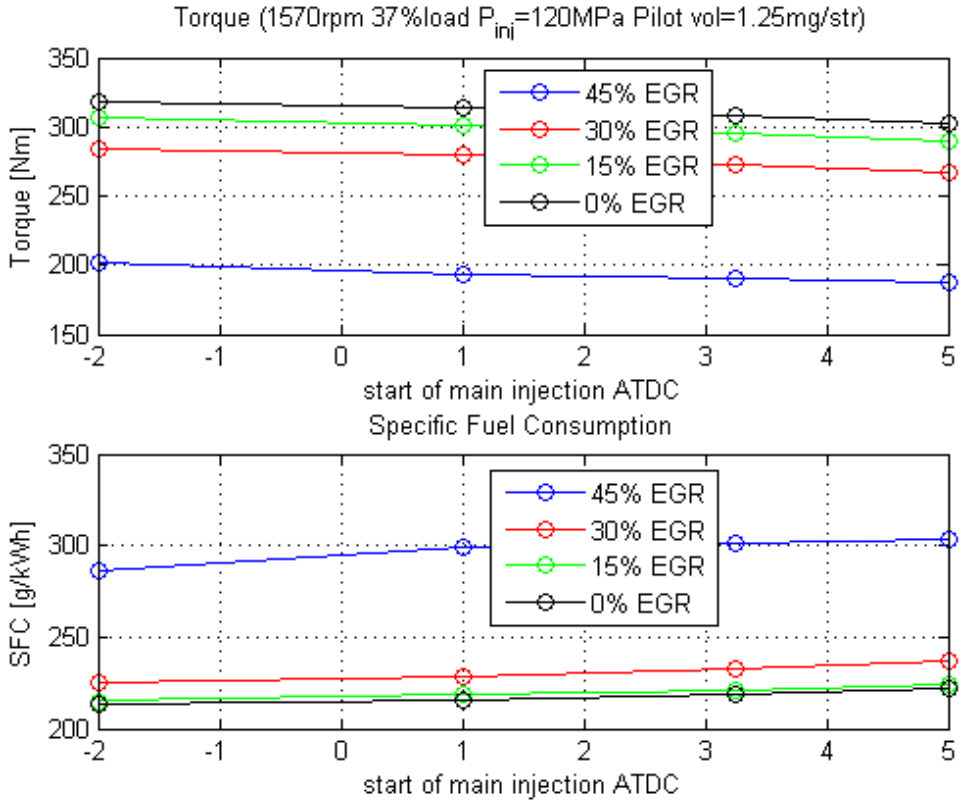
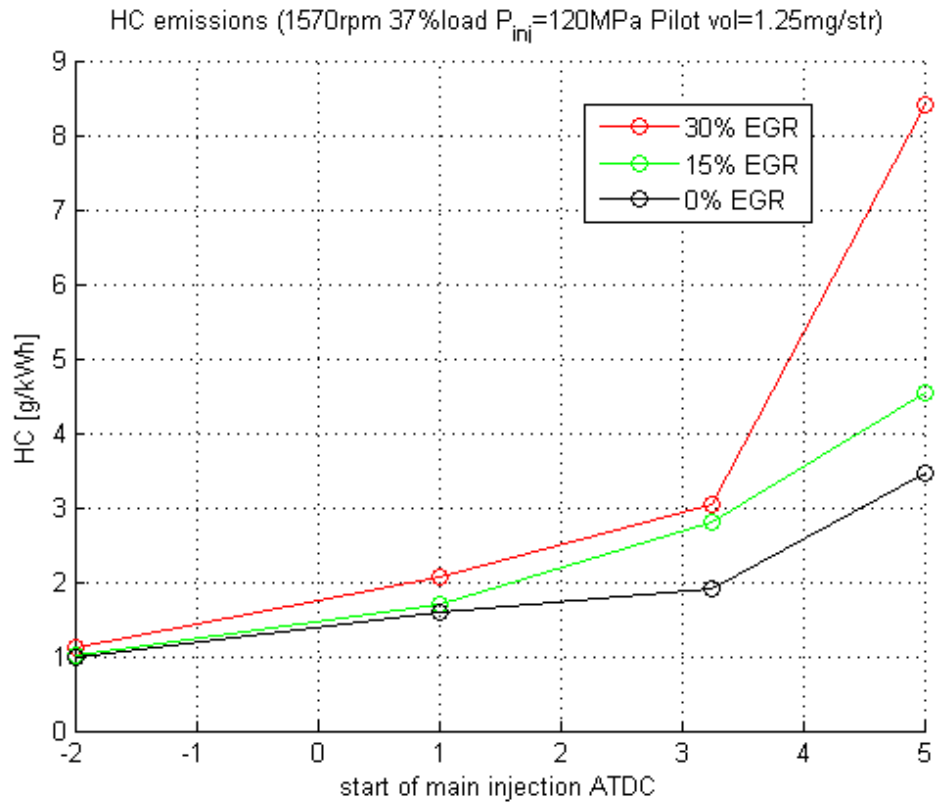


Figure A.11 : Torque and SFC – Effect of main injection timing – Test point 2



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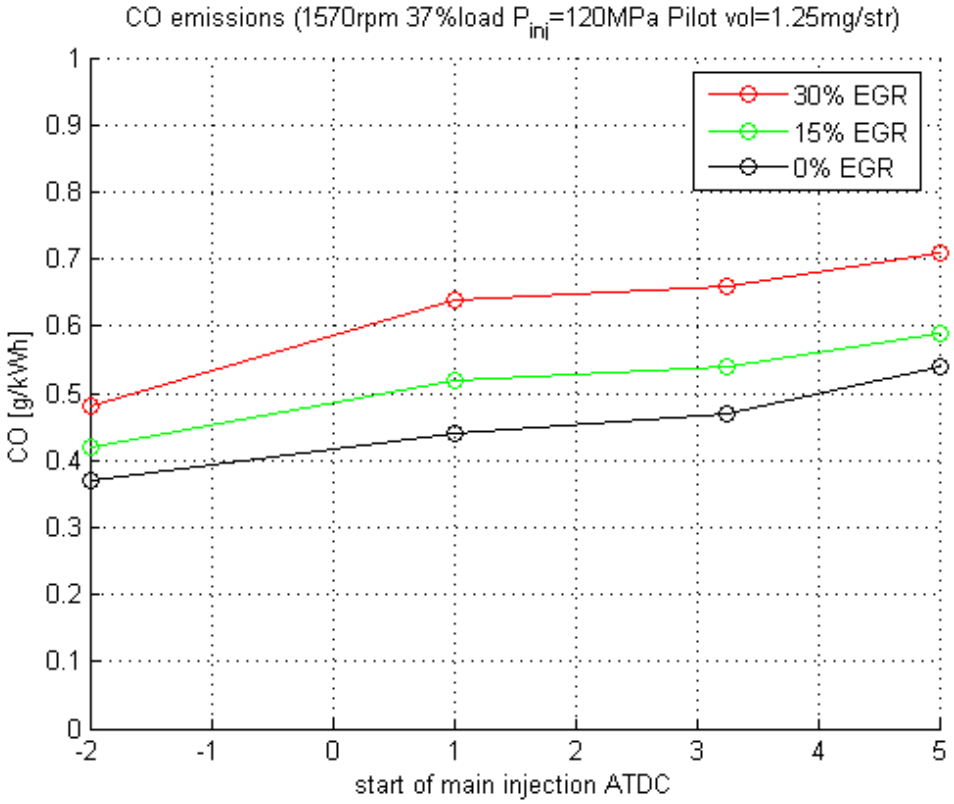


Figure A.13 : CO emissions – Effect of main injection timing – Test point 2

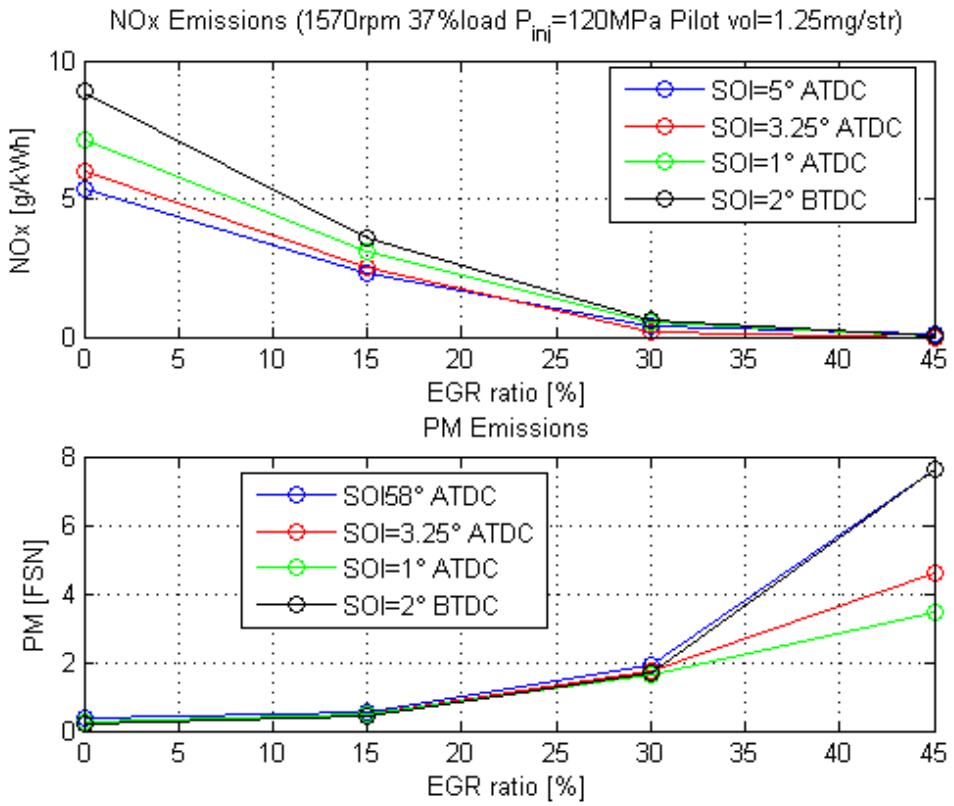


Figure A.14 : NO_x and PM emissions – Effect of EGR – Test point 2

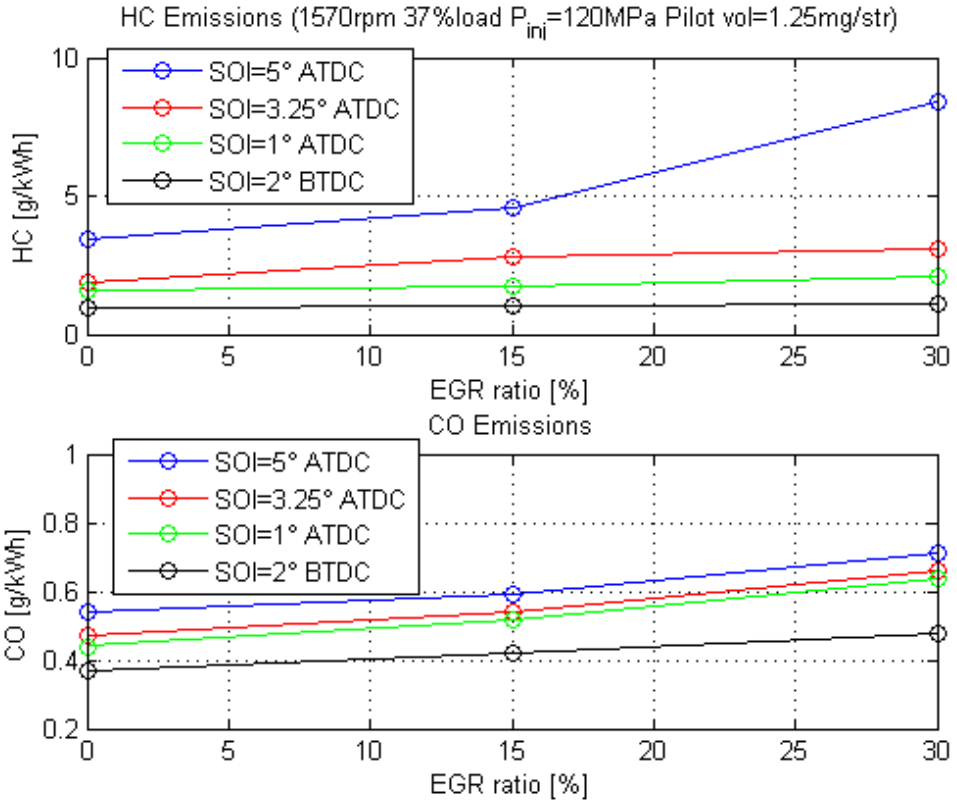


Figure A.15 : HC and CO emissions – Effect of EGR – Test point 2

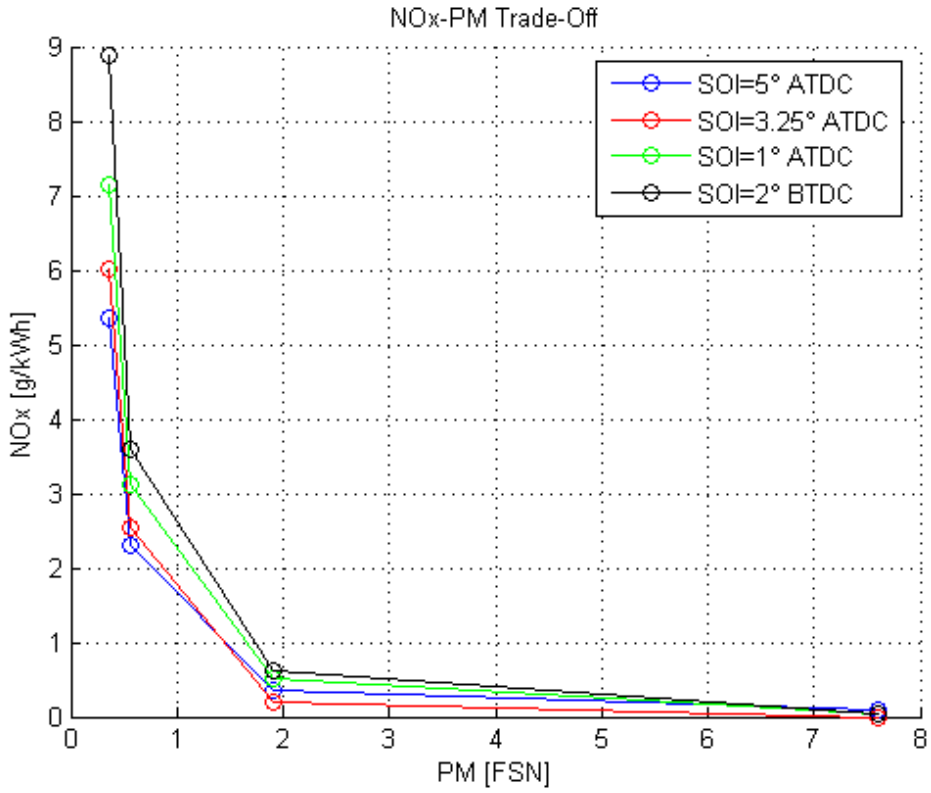


Figure A.16 : NO_x - PM trade off - Test point 2

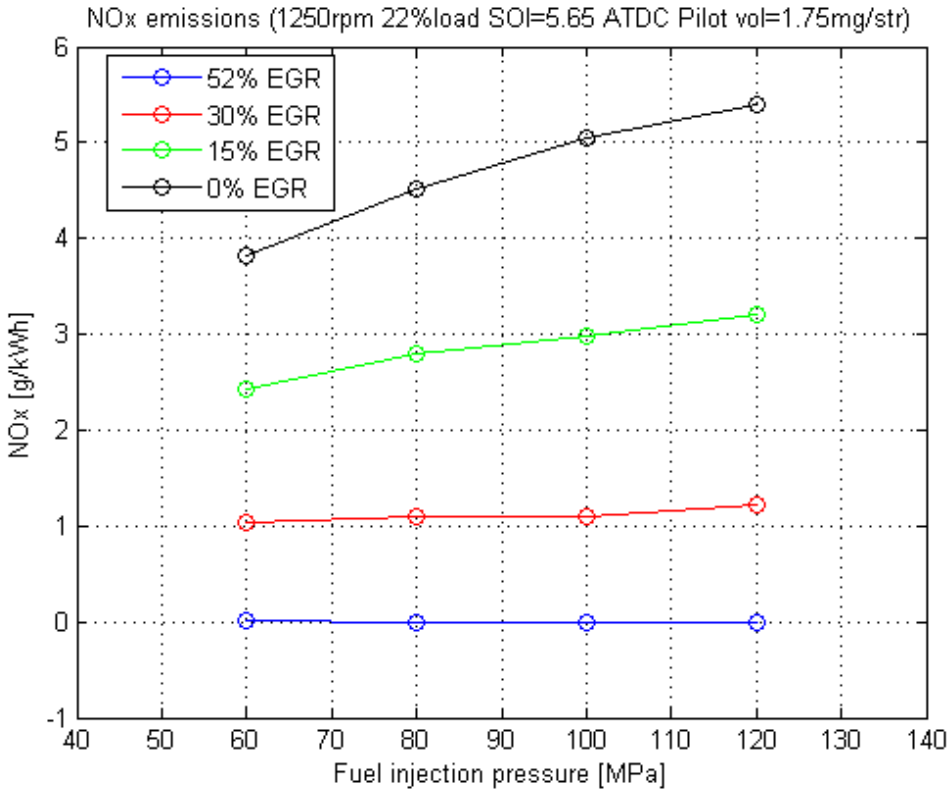


Figure A.17 : NO_x emissions – Effect of fuel injection pressure – Test point 1

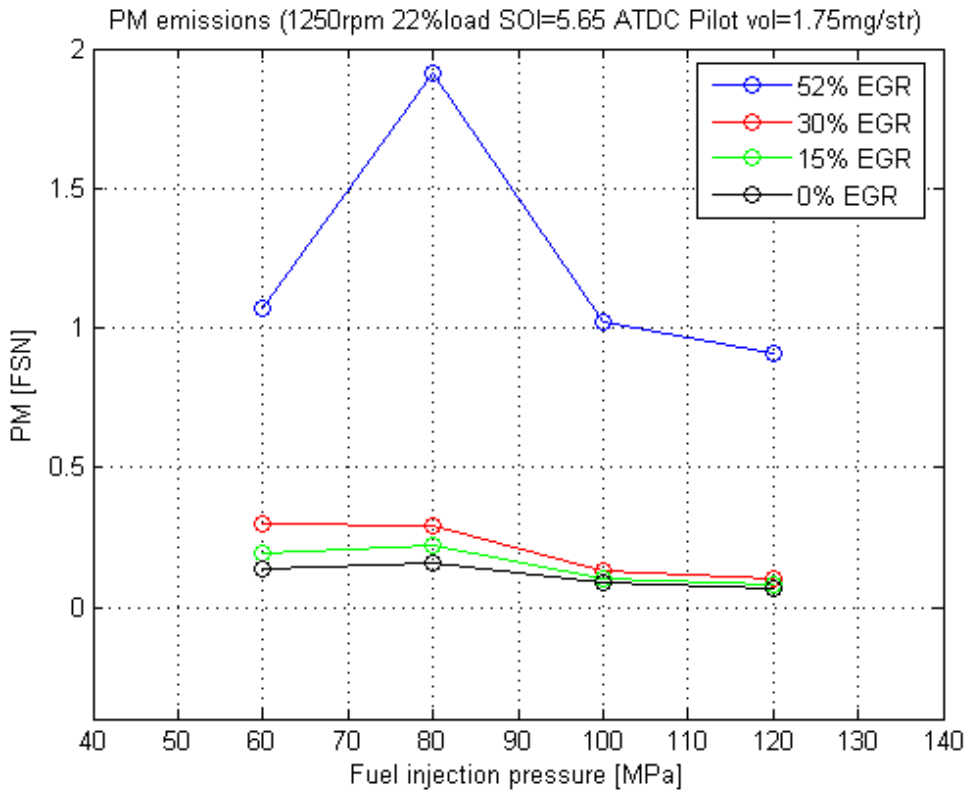


Figure A.18 : PM emissions – Effect of fuel injection pressure – Test point 1

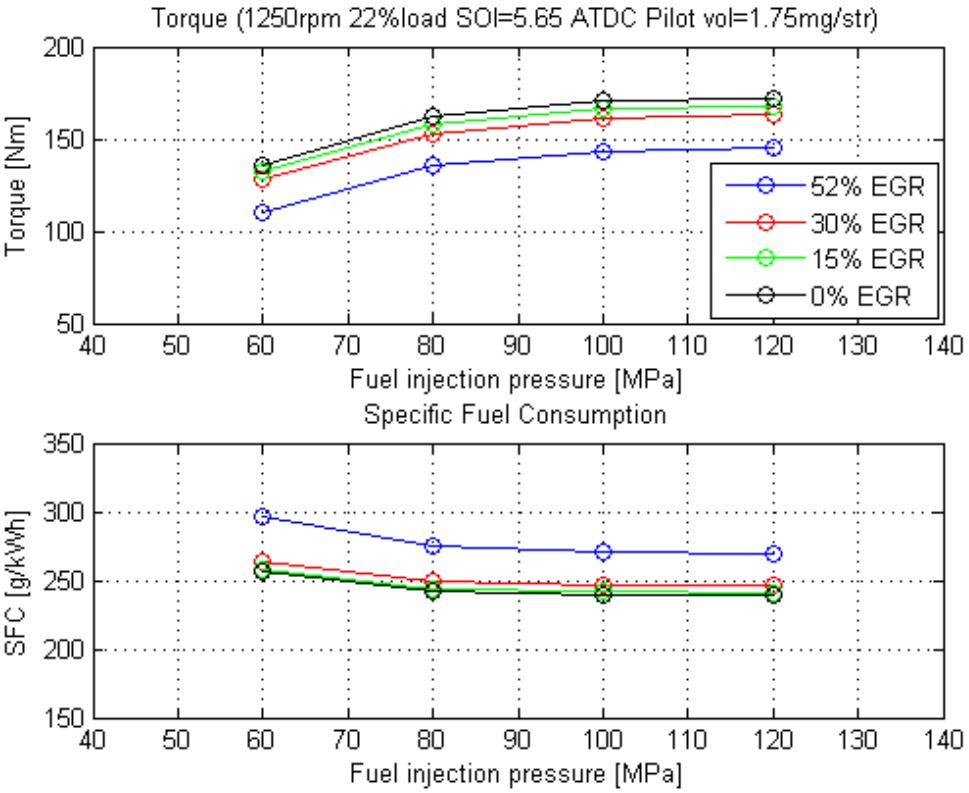


Figure A.19 : Torque and SFC – Effect of fuel injection pressure – Test point 1

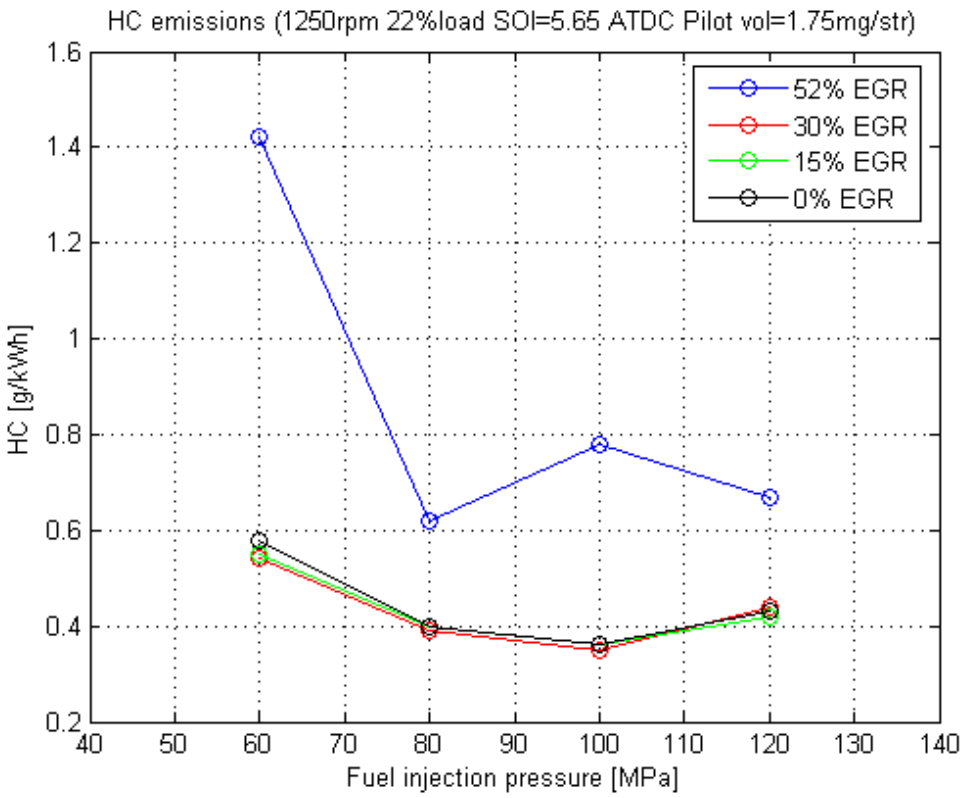
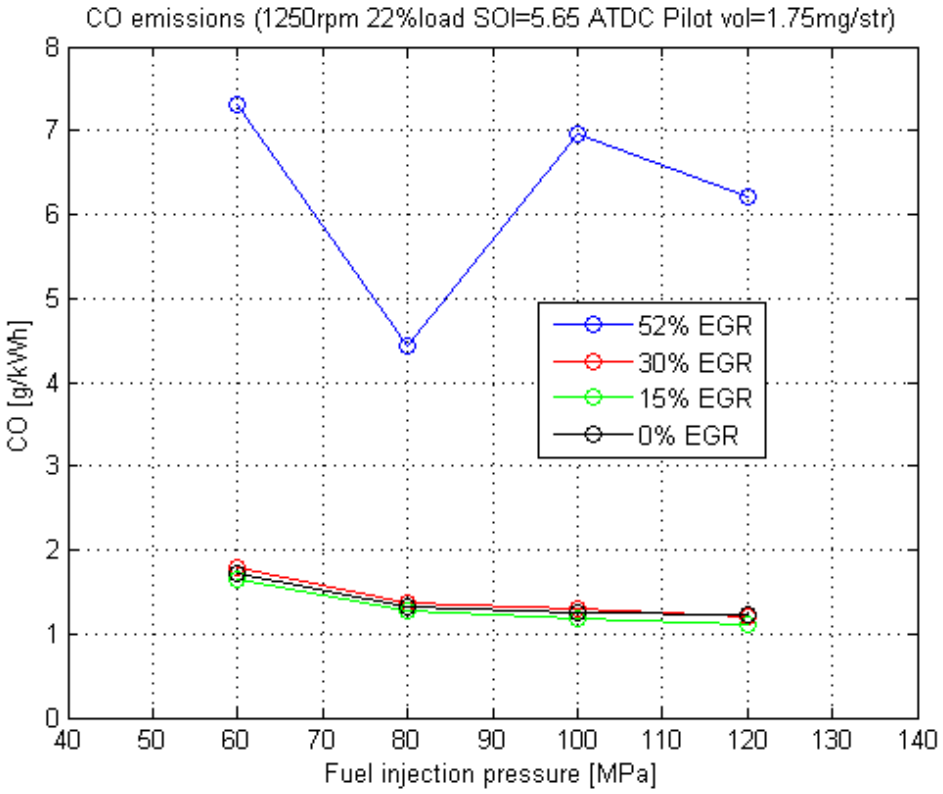


Figure A.20 : HC emissions – Effect of fuel injection pressure – Test point 1



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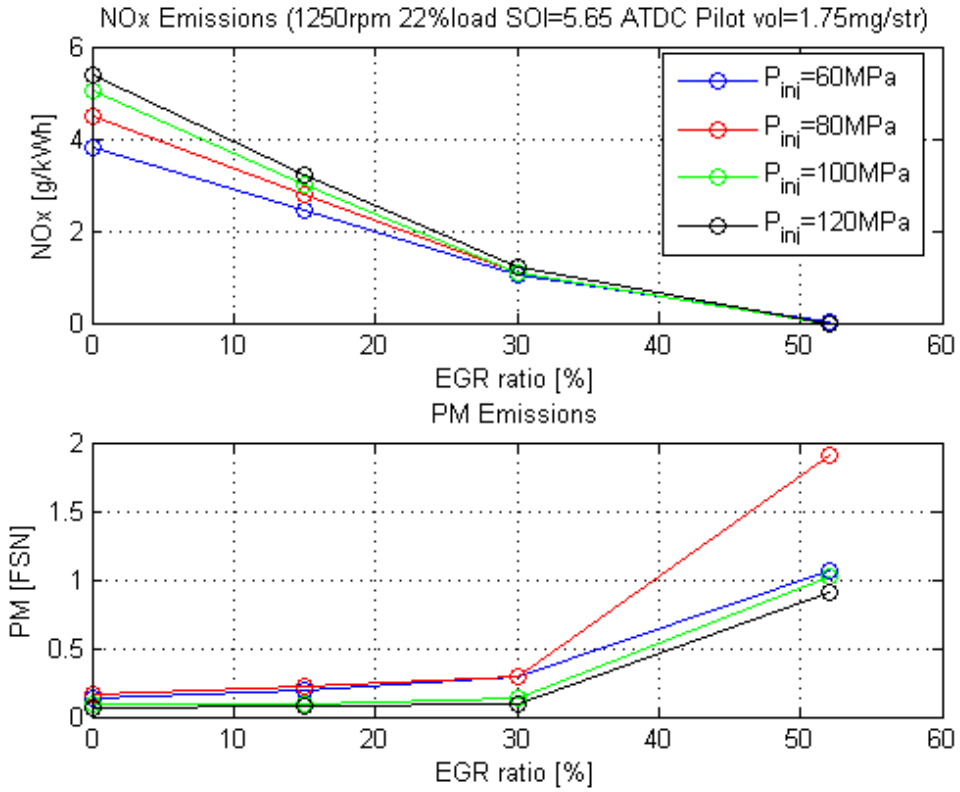


Figure A.22 : NO_x and PM emissions – Effect of EGR – Test point 1

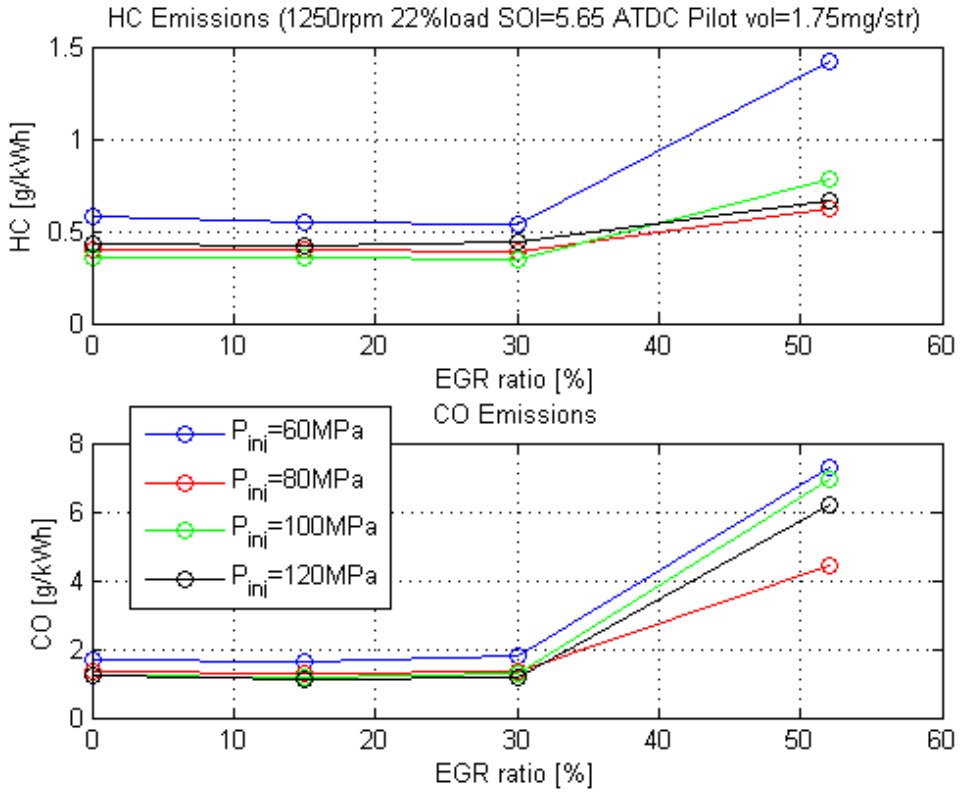


Figure A.23 : HC and CO emissions – Effect of EGR – Test point 1

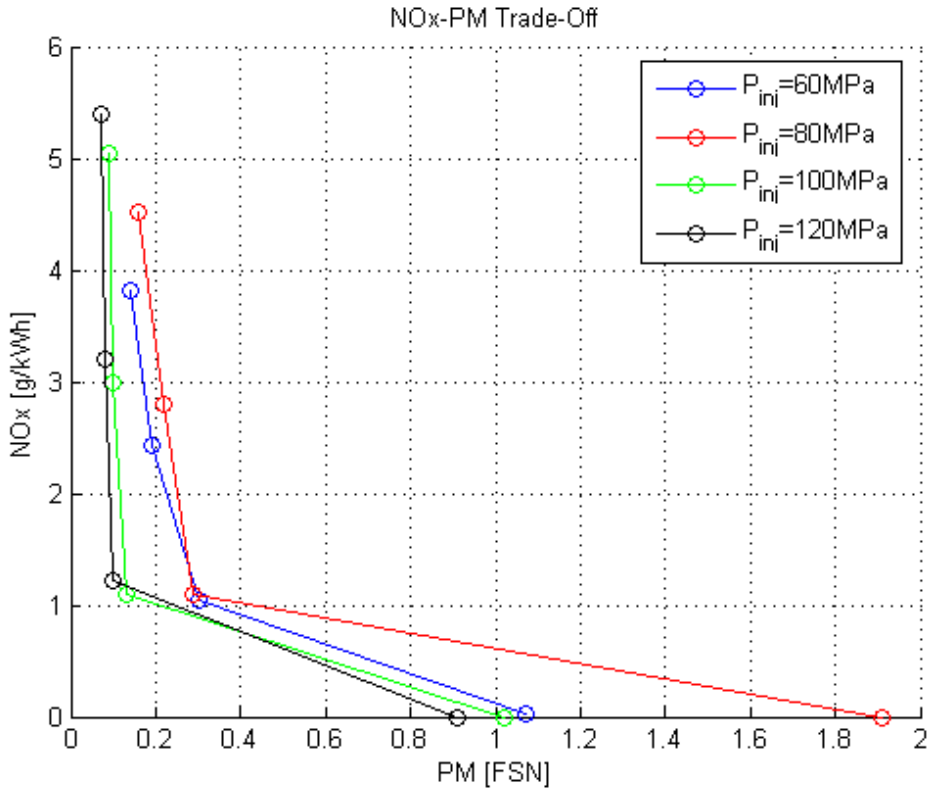


Figure A.24 : NO_x - PM trade off - Test point 1

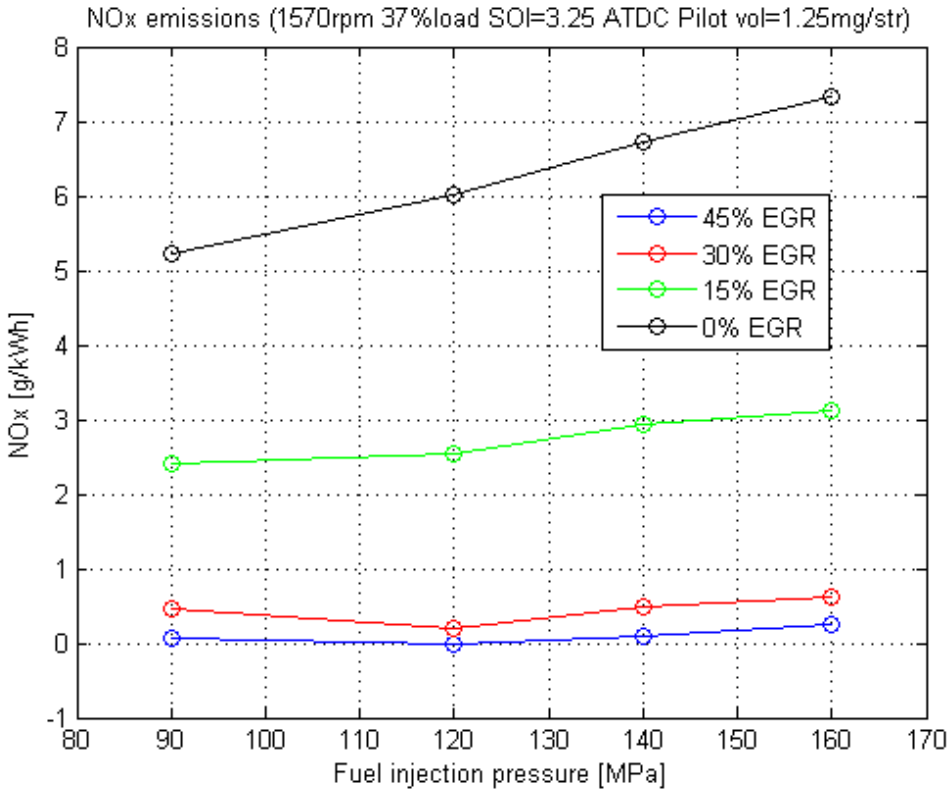


Figure A.25 : NO_x emissions – Effect of fuel injection pressure – Test point 2

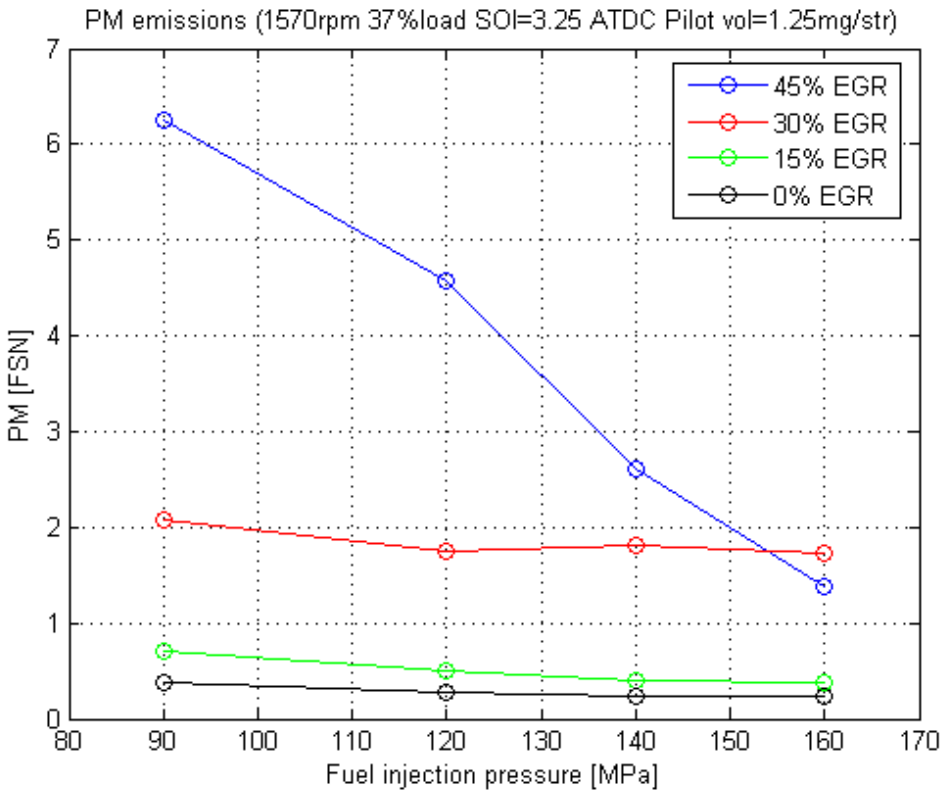


Figure A.26 : PM emissions – Effect of fuel injection pressure – Test point 2

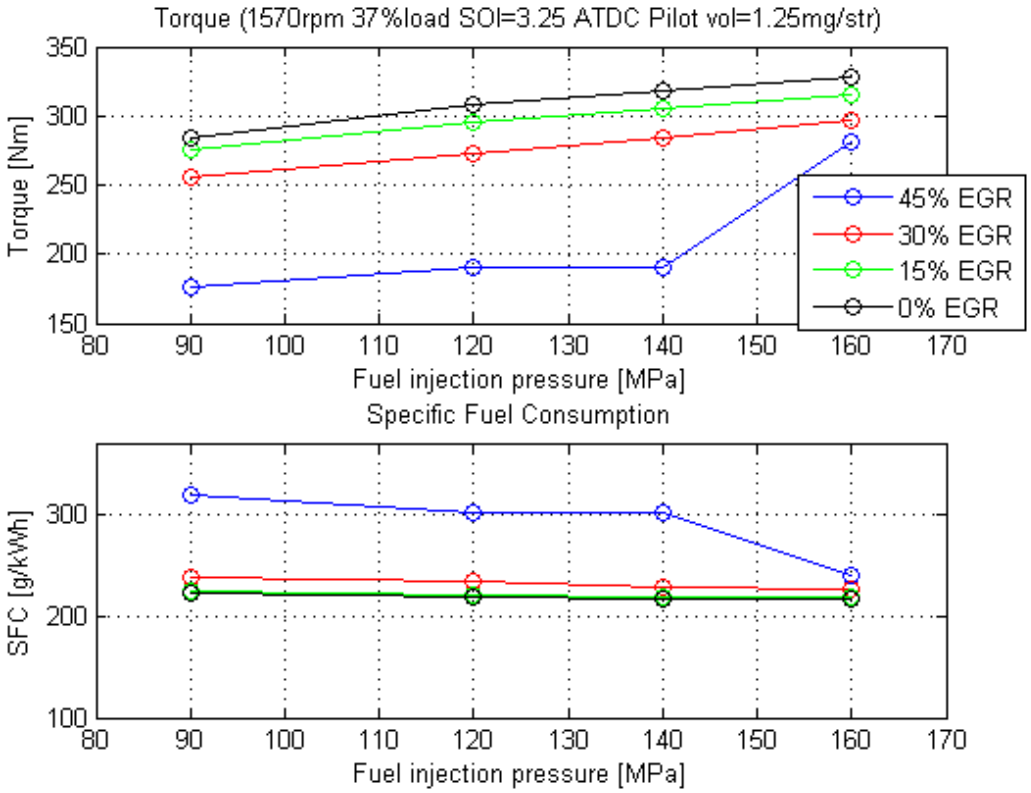


Figure A.27 : Torque and SFC – Effect of fuel injection pressure – Test point 2

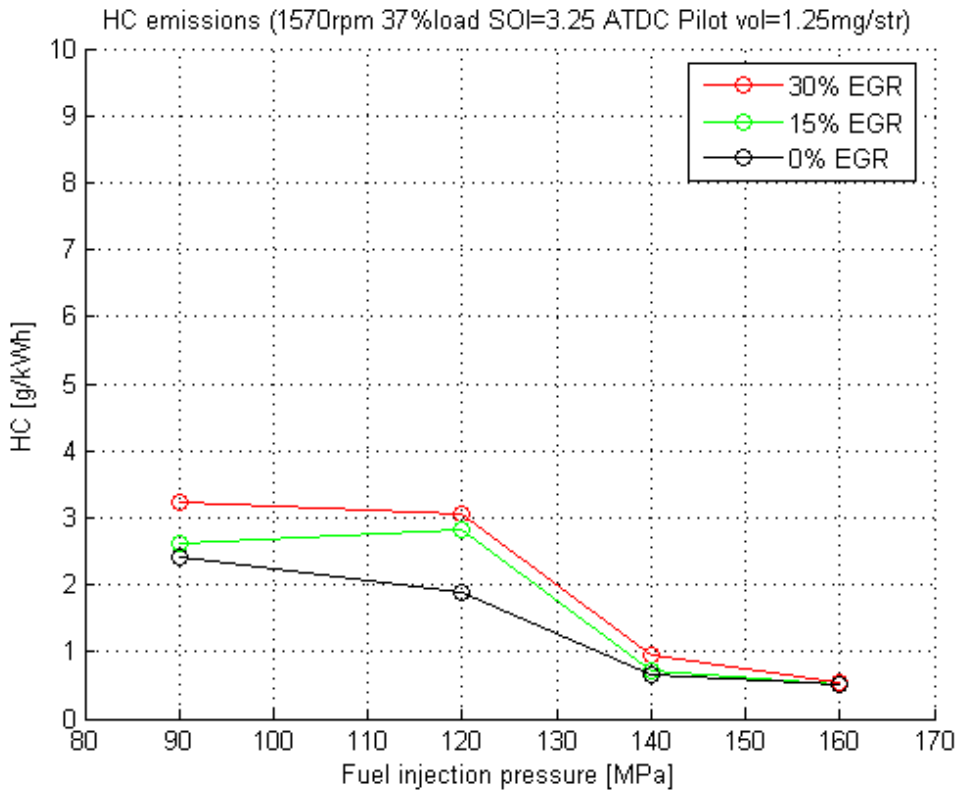


Figure A.28 : HC emissions – Effect of fuel injection pressure – Test point 2

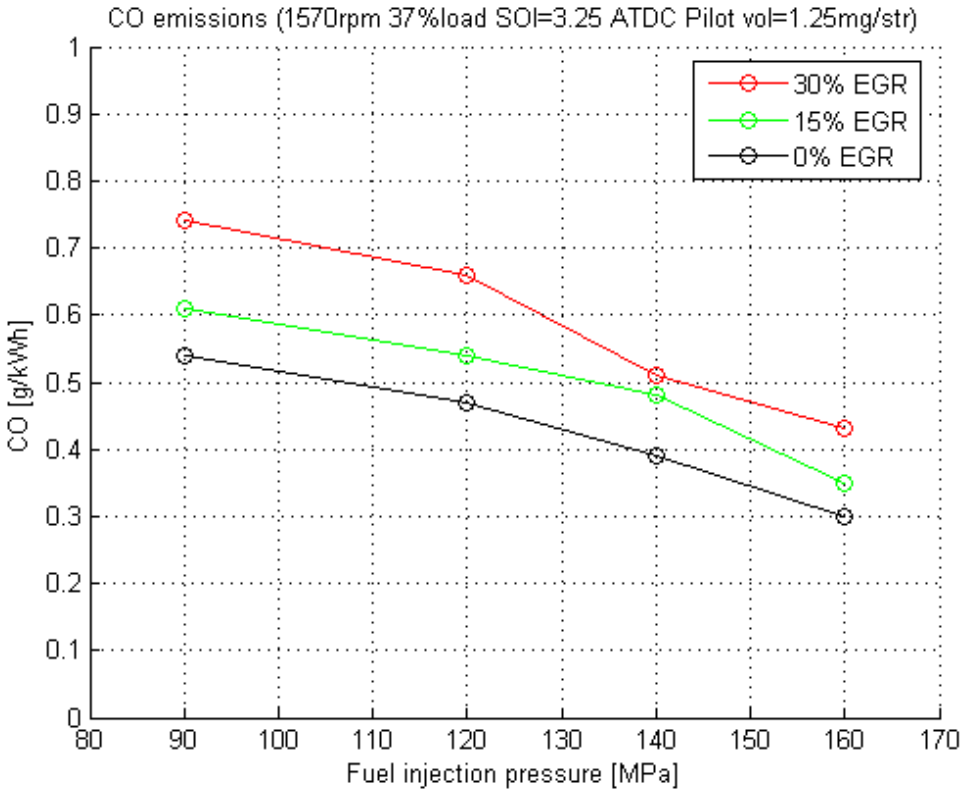


Figure A.29 : CO emissions – Effect of fuel injection pressure – Test point 2

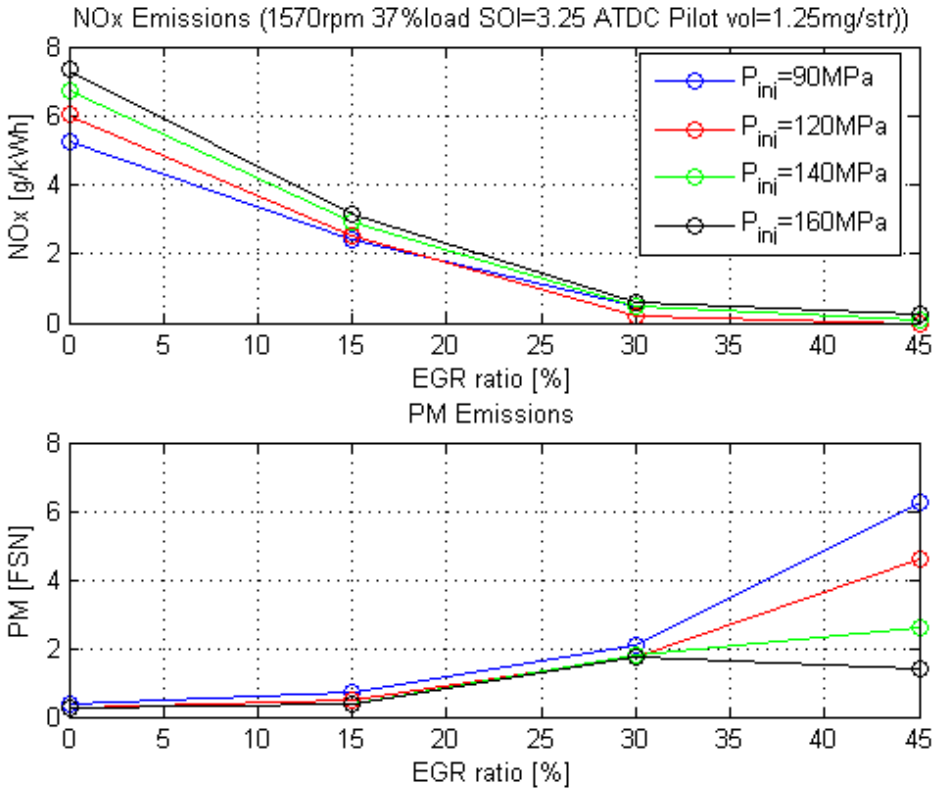
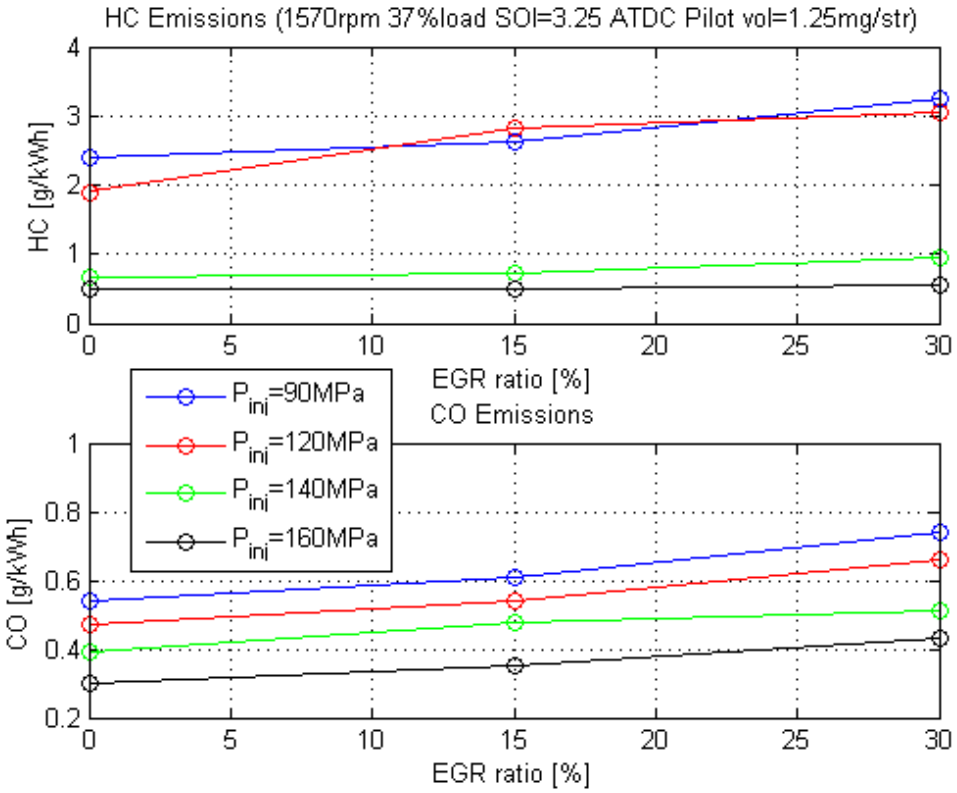


Figure A.30 : NO_x and PM emissions – Effect of EGR – Test point 2



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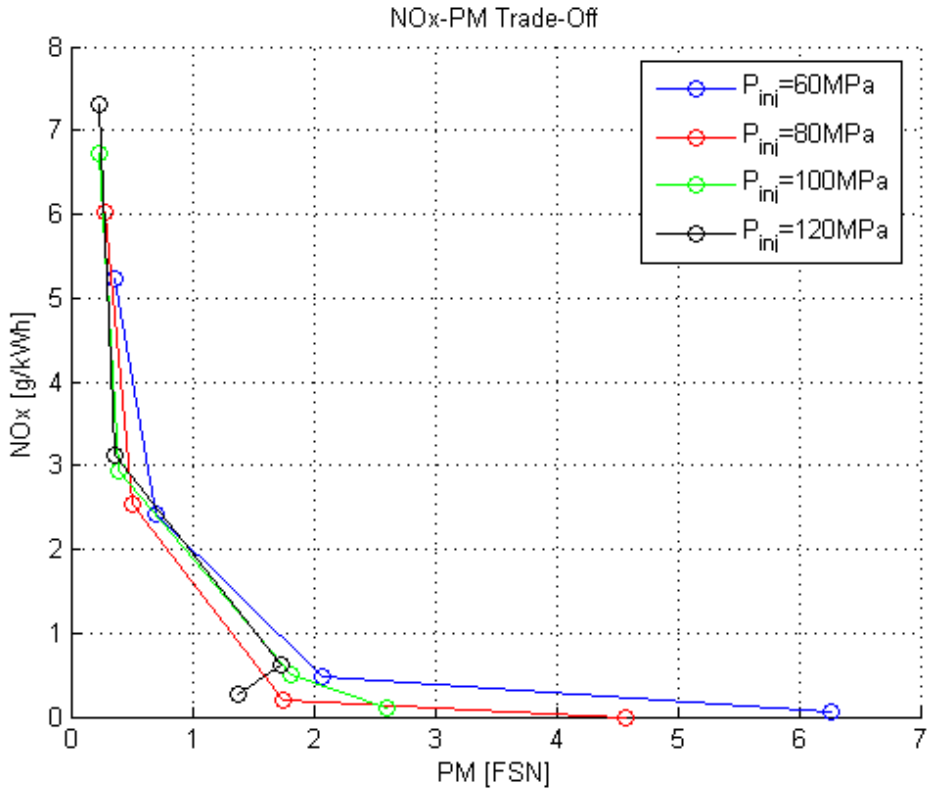

Figure A.32 : NO_x - PM trade off - Test point 2

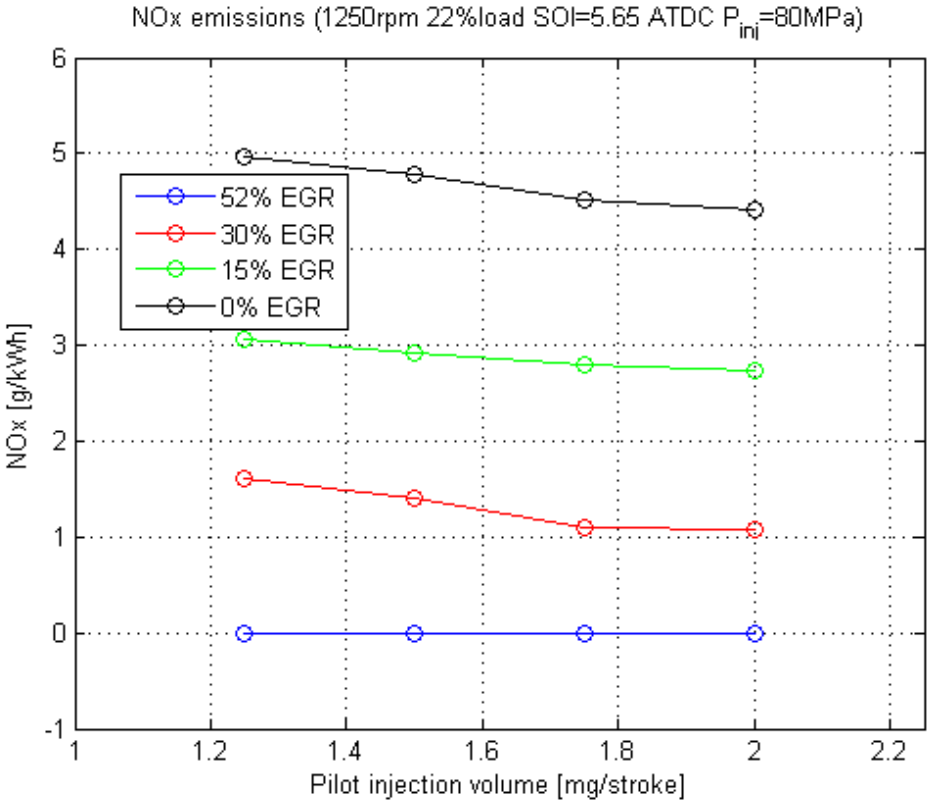

Figure A.33 : NO_x emissions – Effect of pilot injection volume – Test point 1

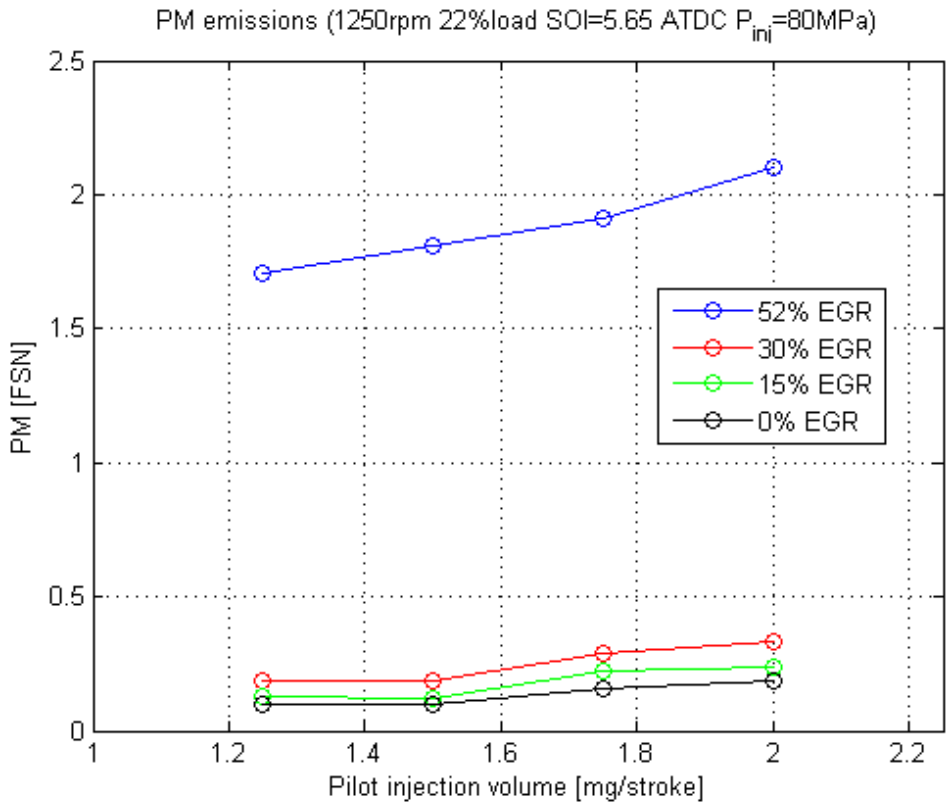

Figure A.34 : PM emissions – Effect of pilot injection volume – Test point 1

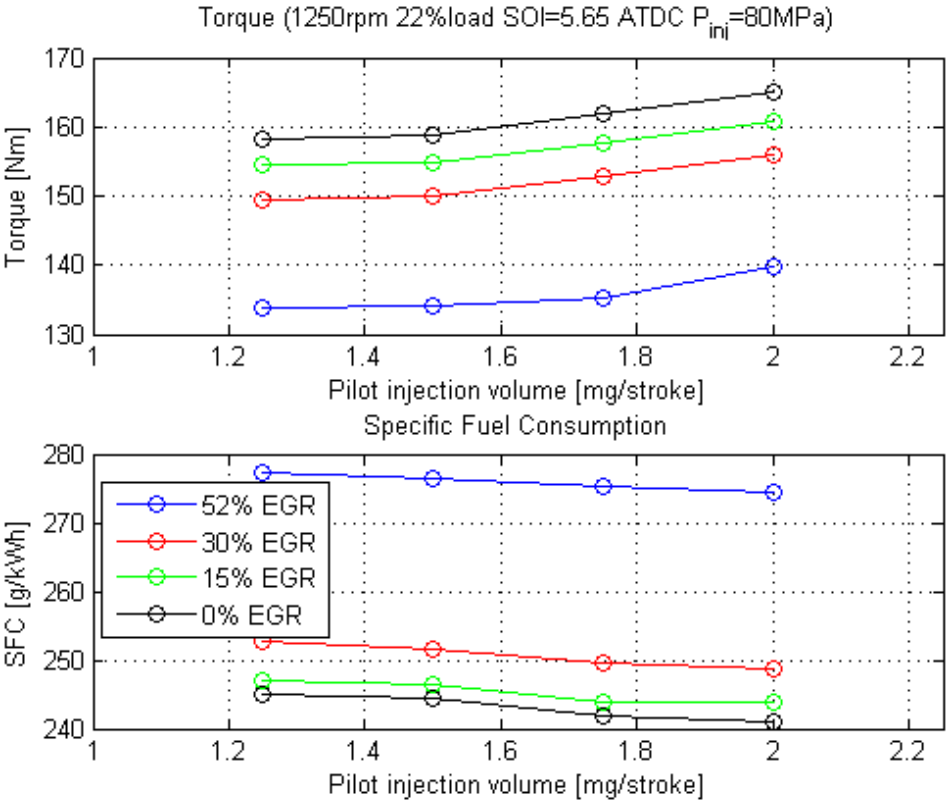

Figure A.35 : Torque and SFC – Effect of pilot injection volume – Test point 1

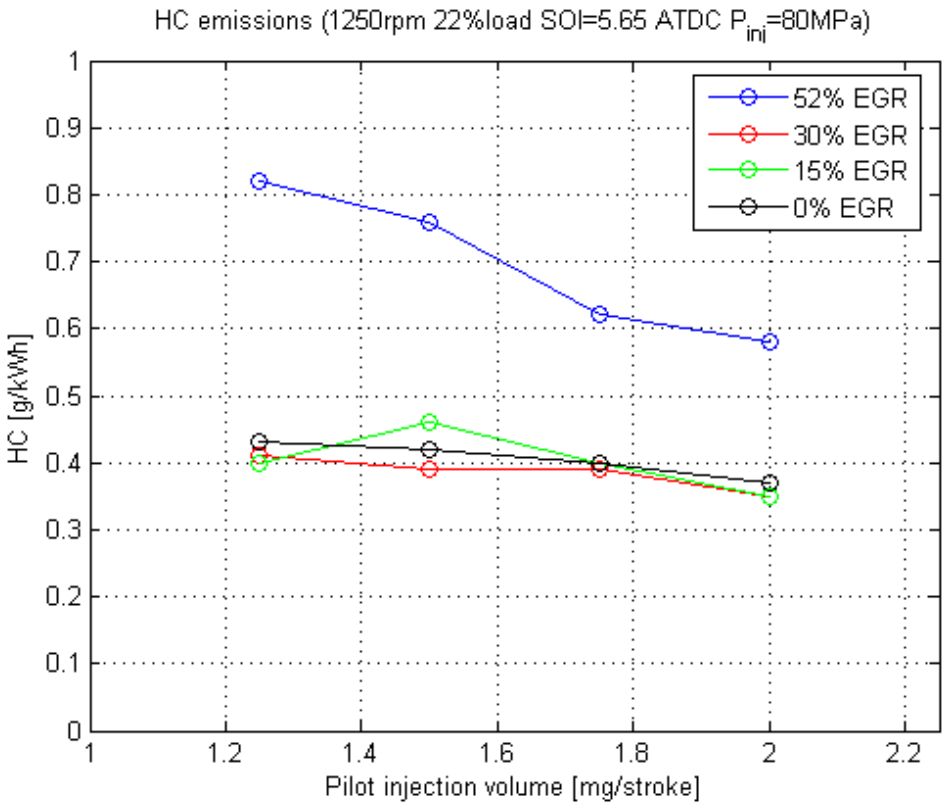

Figure A.36 : HC emissions – Effect of pilot injection volume – Test point 1

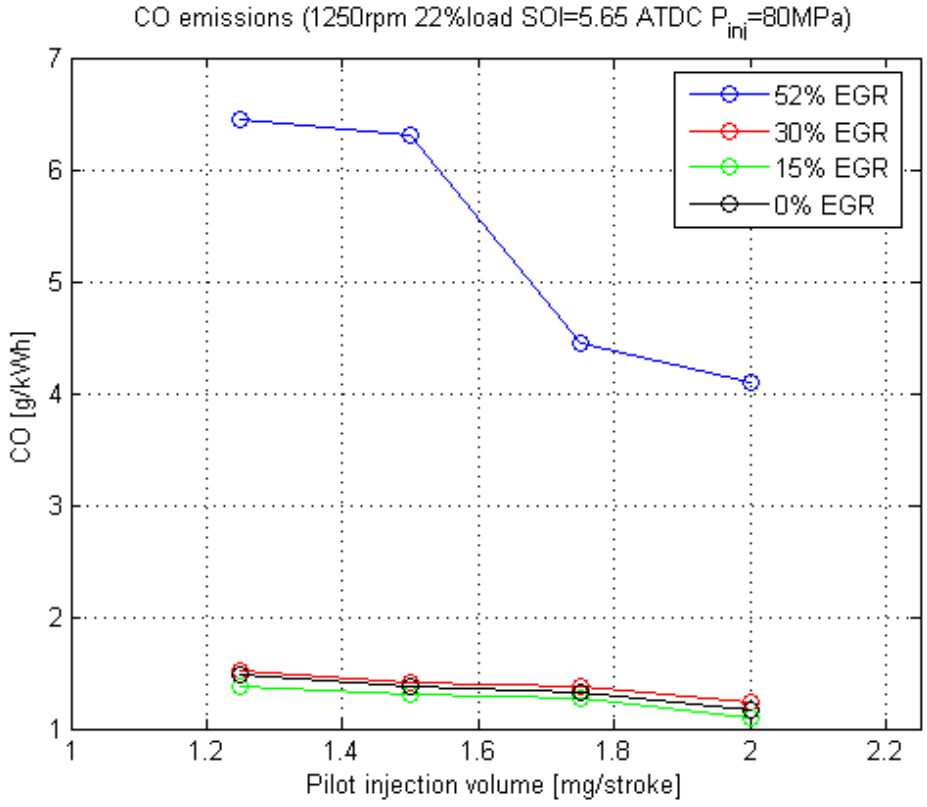

Figure A.37 : CO emissions – Effect of pilot injection volume – Test point 1

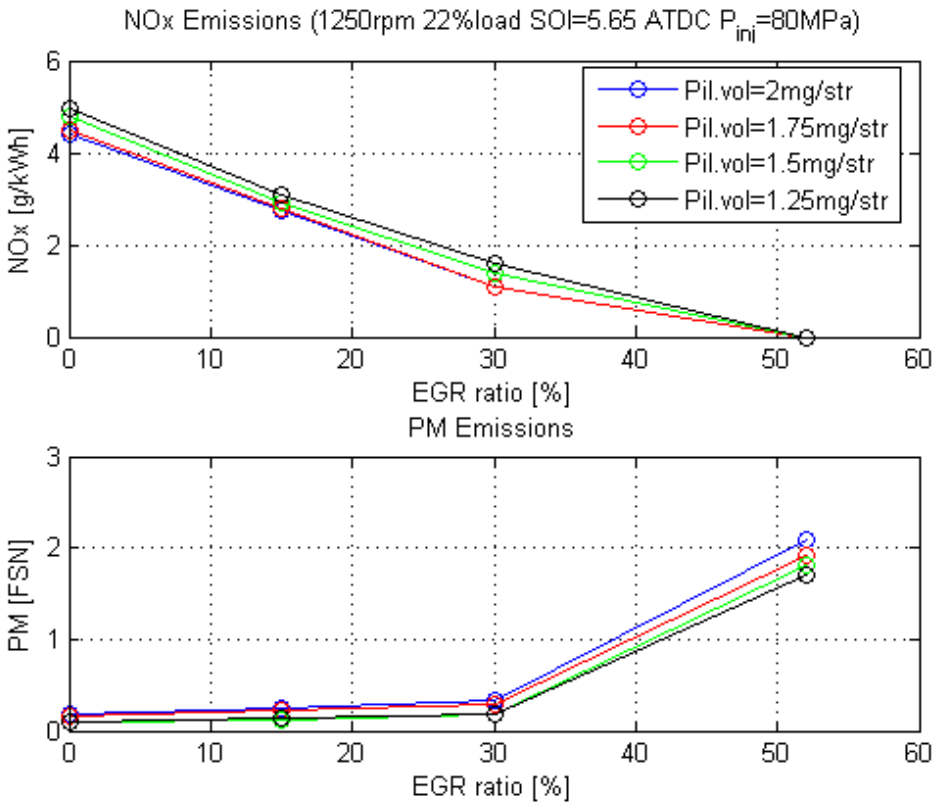

Figure A.38 : NO_x and PM emissions – Effect of EGR – Test point 1

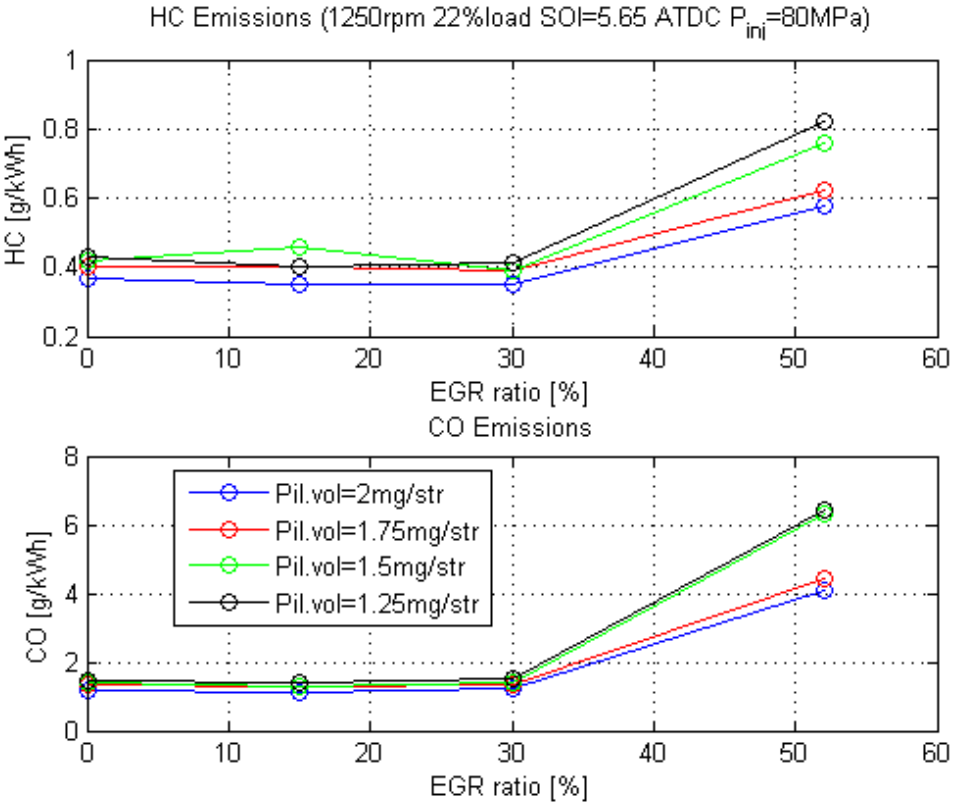

Figure A.39 : HC and CO emissions – Effect of EGR – Test point 1

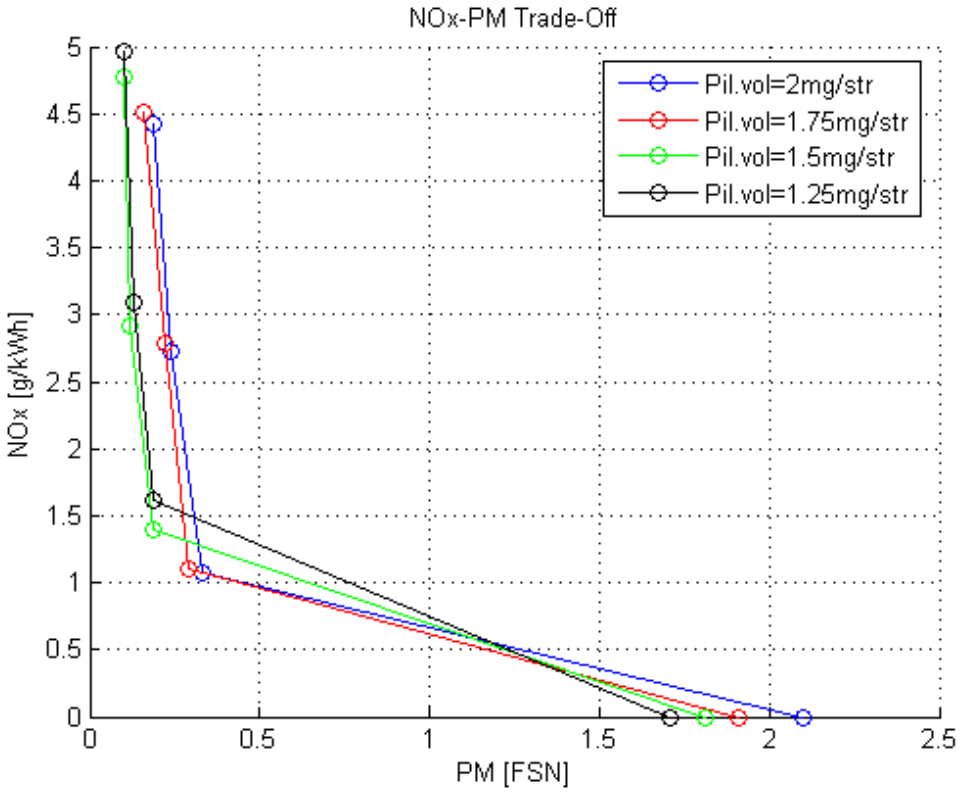

Figure A.40 : NO_x - PM trade off - Test point 1

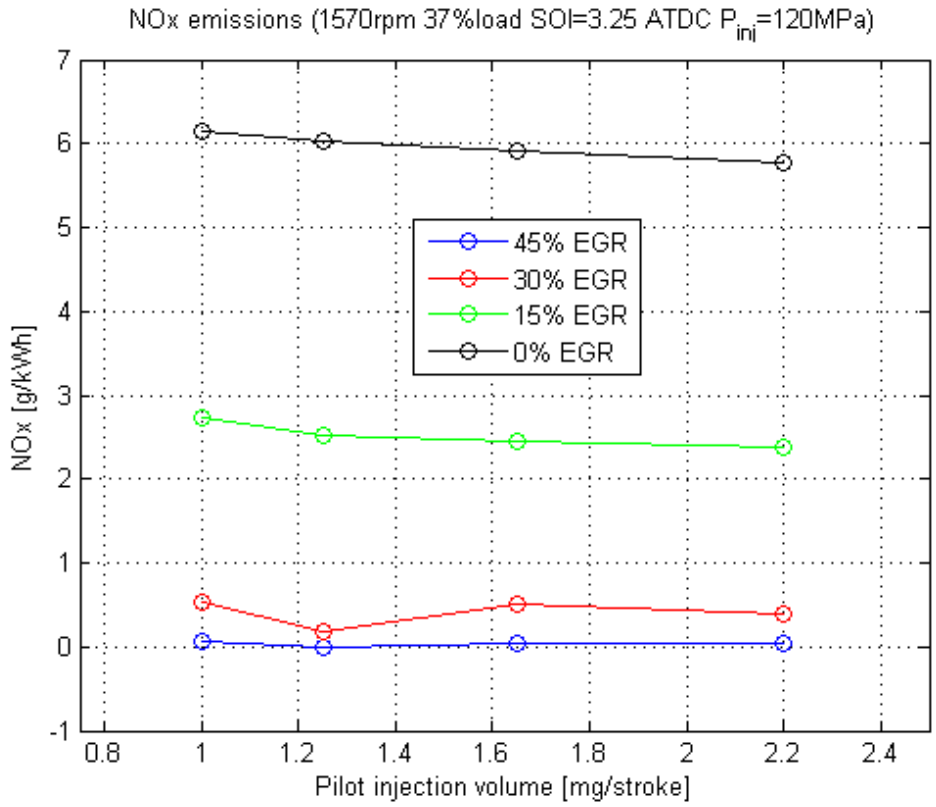

Figure A.41 : NO_x emissions – Effect of pilot injection volume – Test point 2

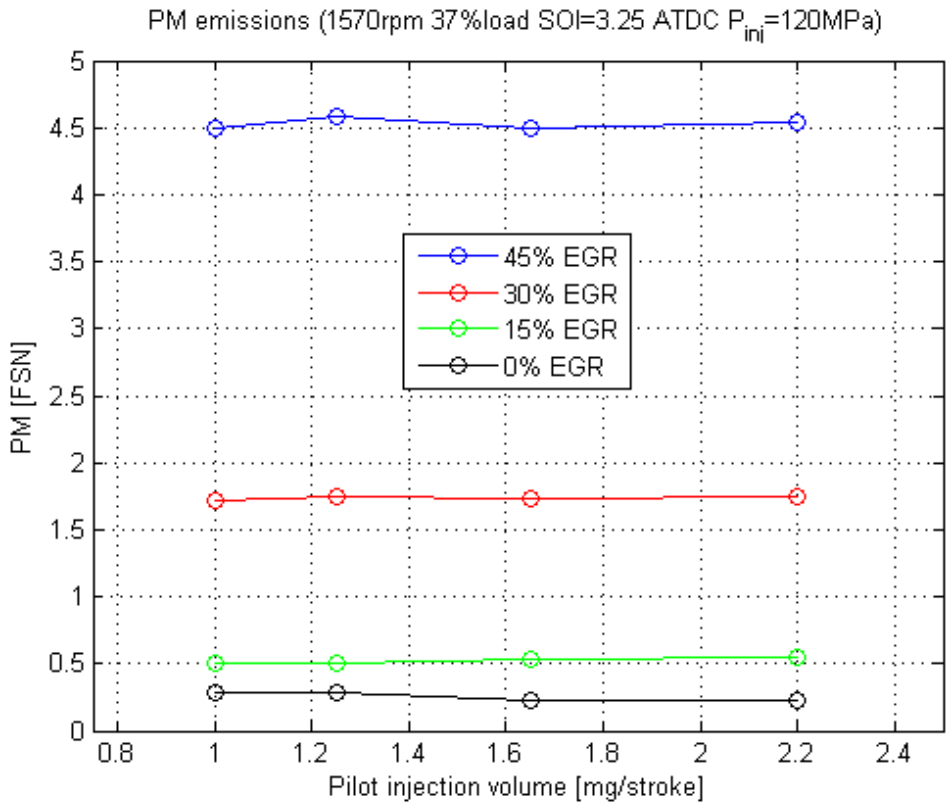

Figure A.42 : PM emissions – Effect of pilot injection volume – Test point 2

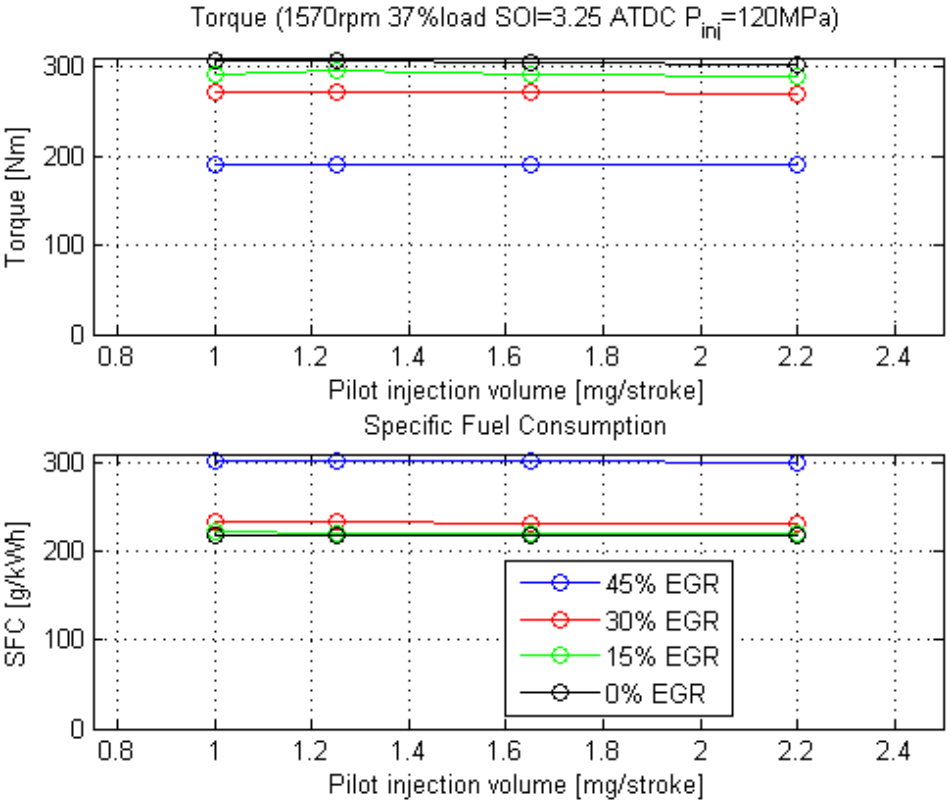

Figure A.43 : Torque and SFC – Effect of pilot injection volume – Test point 2

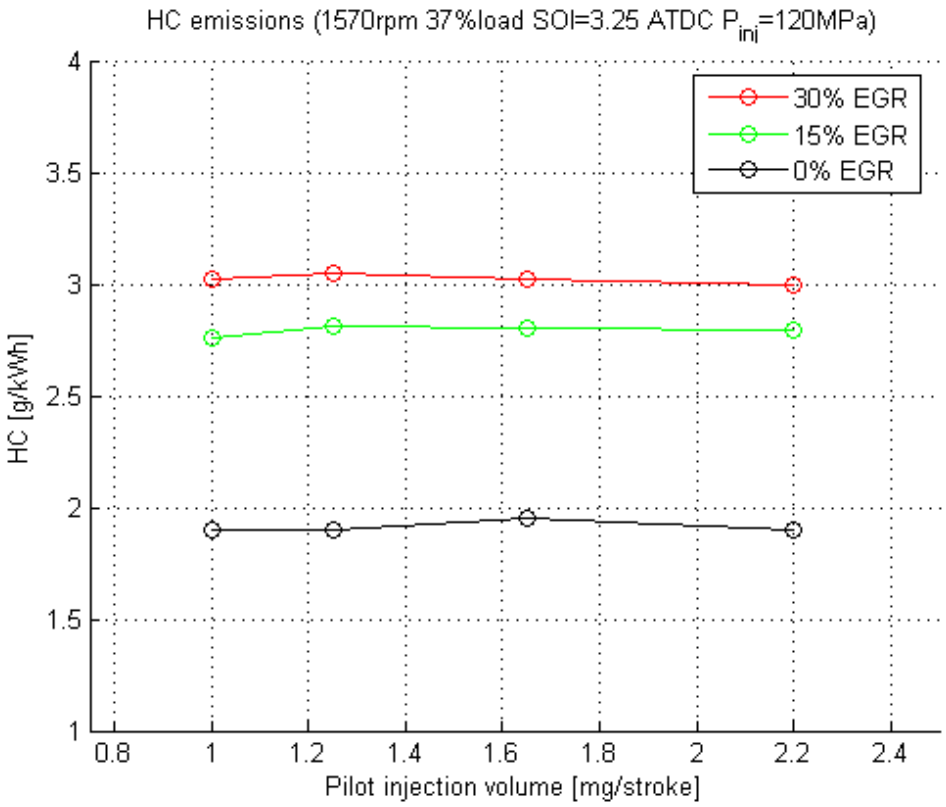

Figure A.44 : HC emissions – Effect of pilot injection volume – Test point 2

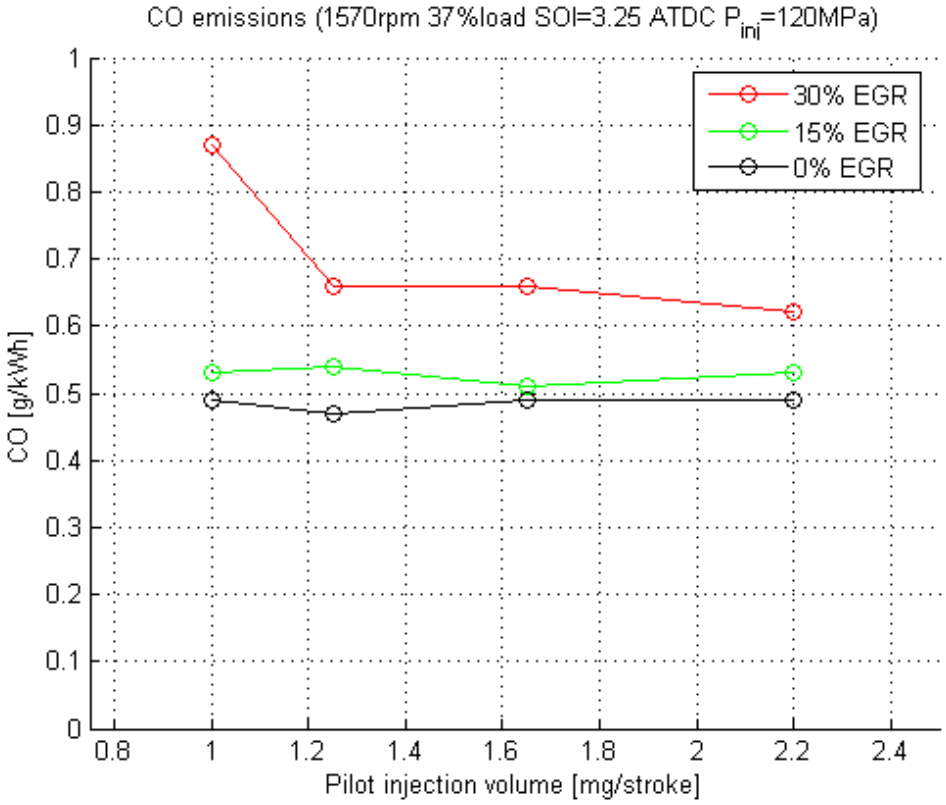

Figure A.45 : CO emissions – Effect of pilot injection volume – Test point 2

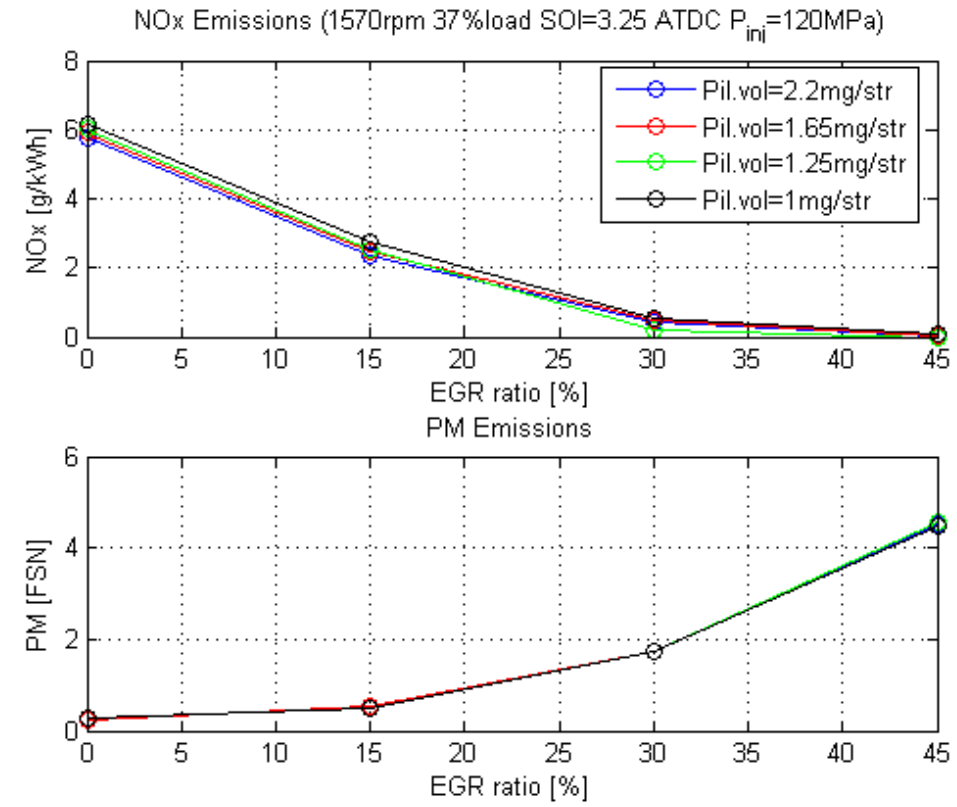

Figure A.46 : NO_x and PM emissions – Effect of EGR – Test point 2

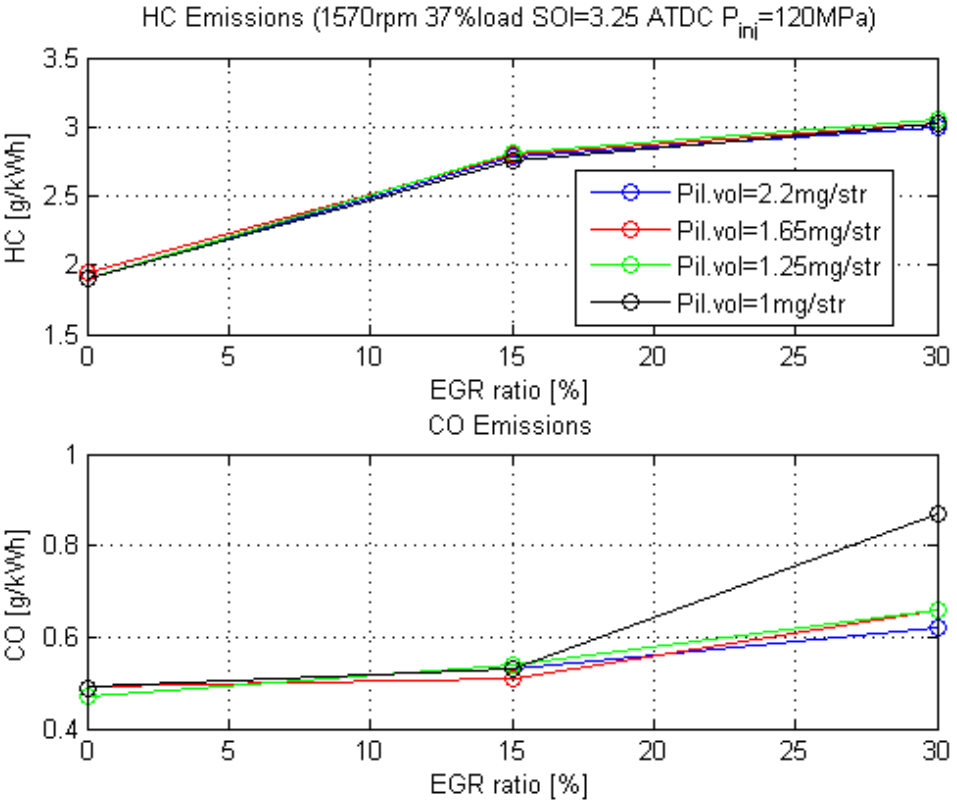

Figure A.47 : HC and CO emissions – Effect of EGR – Test point 2

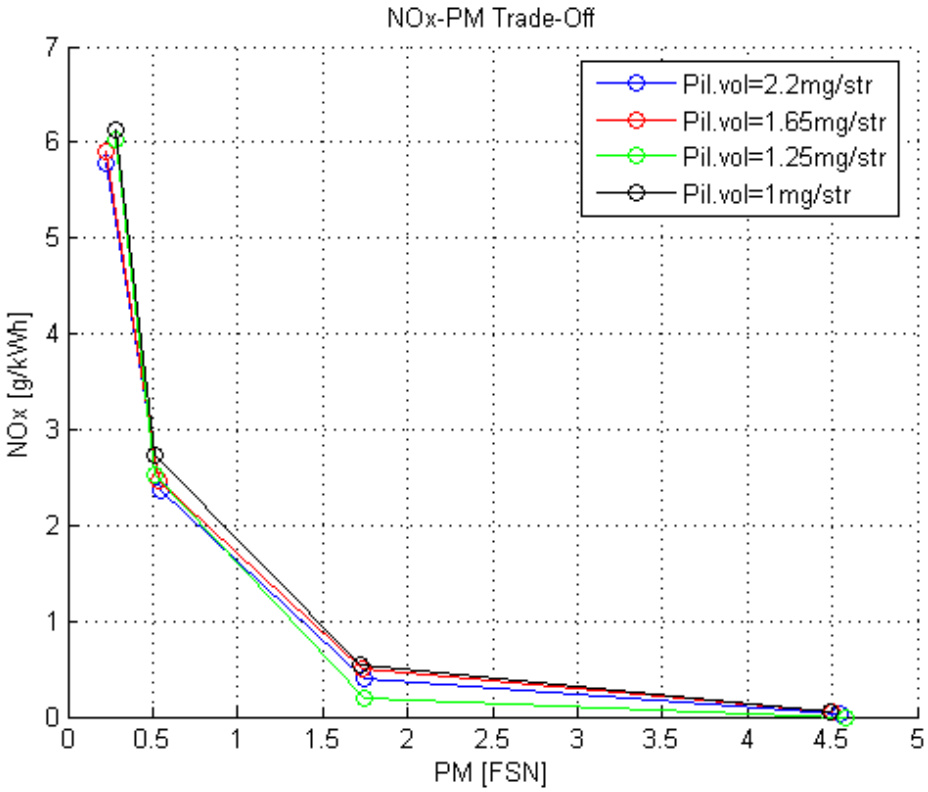

Figure A.48 : NO_x - PM trade off - Test point 2

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