

ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE
ENGINEERING AND TECHNOLOGY

EXPERIMENTAL COMBUSTION ANALYSIS OF DIESEL ENGINE



Ph.D. THESIS

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Department of Mechanical Engineering

Mechanical Engineering Programme

JUNE 2019

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ABBREVIATIONS

ACM	: Advanced Combustion Method
AFR	: Air-Fuel Ratio
APP	: Acceleration Pedal Position
aTDC	: After Top Dead Center
BD	: Burn Duration
BEV	: Battery Electric Vehicle
bTDC	: Before Top Dead Centre
CA	: Crank Angle
CAI	: Controlled Auto Ignition
CI	: Compression Ignition
DI	: Direct Injection
CO	: Carbon Monoxide
CO₂	: Carbon Dioxide
DOC	: Diesel Oxidation Catalyst
DPF	: Diesel Particulate Filter
ECU	: Engine Control Unit
EGR	: Exhaust Gas Recirculation
EGT	: Exhaust Gas Temperature
EOI	: End Of Injection
EPA	: Energy Protection Agency
EU	: European Union
EUDC	: Extra Urban Driving Cycle
FL	: Full Load
GCV	: Gross Calorific Value
HC	: Hydrocarbon
HCCI	: Homogeneous Charge Compression Ignition
HCLI	: Homogeneous Charge Late Injection
HPLI	: Highly Premixed Late Injection
HEV	: Hybrid Electric Vehicles
HiMICS	: Homogeneous Charge Intelligent Multiple Injection Combustion System
HRR	: Heat Release Rate
ICE	: Internal Combustion Engine
IDD	: Ignition Delay Duration
LCV	: Light Commercial Vehicles
LNT	: Lean NO _x Trap
LTC	: Low Temperature Combustion
MBF	: Mass Burned Fuel
MK	: Modulated Kinetics
MRC	: Marmara Research Center
MULDIC	: Multiple Stage Diesel Combustion
NADI	: Narrow Angle Direct Injection
NEDC	: New European Driving Cycle
NO_x	: Nitrogen Oxides

OBD	: On Board Diagnostics
OEM	: Original Equipment Manufacturers
PCI	: Premixed Compression Ignited Combustion
PFI	: Port Fuel Injection
PHEV	: Plug in Hybrid Electrical Vehicle
PLDV	: Passenger Light Duty Vehicle
PM	: Particulate Mass
PN	: Particulate Number
PPCI	: Partially Premixed Compression Ignition
PREDIC	: PREmixed lean Diesel Combustion
ROHR	: Rate Of Heat Release
SCR	: Selective Catalyst Reduction
SOC	: Start Of Combustion
SOI	: Start Of Injection
TA	: Type Approval
TDC	: Top Dead Center
UDC	: Urban Driving Cycle
UNIBUS	: Uniform Bulky Combustion System
WLTC	: Worldwide Harmonized Light Vehicles Test Cycle
ZEV	: Zero Emission Vehicle

SYMBOLS

c_p	: Specific heat at constant pressure
c_v	: Specific heat at constant volume
E_{wheel}	: Energy consumption of the vehicle
E_{rex}	: Energy consumption of the range extender unit
F_{wheel}	: Velocity of the vehicle
L	: Volumetric efficiency
$MBF5$	: Crank angle where 5% of the injected fuel was burned
$MBF10$	: Crank angle where 10% of the injected fuel was burned
$MBF50$	: Crank angle where 50% of the injected fuel was burned
$MBF90$	: Crank angle where 90% of the injected fuel was burned
P	: Pressure
P_i	: In-cylinder pressure according to crank angle
P_{max}	: The highest in-cylinder pressure
P_s	: Pressure in the intake manifold
P_{soc}	: Pressure in the cylinder at start of combustion
R	: Universal gas constant
Q	: Heat transfer
T	: Temperature
T_i	: In-cylinder temperature according to crank angle
T_{max}	: The highest in-cylinder temperature
T_{rex_work}	: Range extender working duration in seconds
T_s	: Temperature in the intake manifold
T_{soc}	: Average gas temperature in the cylinder at start of combustion
V	: Swept volume of the cylinder
V_i	: Volume of the cylinder according to crank angle
V_h	: Total engine volume
V_{wheel}	: Force at wheel of the vehicle
γ	: Specific heat ratio c_p/c_v
λ	: Lambda
Θ	: Equivalence ratio
κ	: Polytrophic coefficient
η_{trans}	: Efficiency of transmission
η_{motor}	: Efficiency of electric motor
$\eta_{battery}$	: Efficiency of battery
$\eta_{generator}$	: Efficiency of generator



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EXPERIMENTAL COMBUSTION ANALYSIS OF DIESEL ENGINE

SUMMARY

Diesel engines are quite popular in daily life as they have lower fuel consumption than gasoline engines. Because, diesel engines have higher compression ratio and no throttle loss that enable to achieve higher efficiency at both part-load and full load conditions compared to gasoline engines. However, additional exhaust components (DPF, SCR) need to be utilized to achieve the limits of NO_x and soot that were declared by the European Commission.

Researchers and automotive companies have been conducting studies on the combustion of diesel engines since the introduction of emission regulations by the governments. First homogenous diesel combustion were tried at 1957 to kill the smoke. Numerous scientists up to now have studied different type of diesel combustion methods that aim to combine the advantages of gasoline (homogeneity) and diesel (high compression ratio, no throttling loss) engines. On the other hand, definition of the conventional diesel combustion was changed from 90s to nowadays due to obligatory emission regulations and customer expectations. Previously, all fuel was injected into cylinder in single injection before top dead center (bTDC) to achieve the maximum in-cylinder pressure approximately 5 crank angle after top dead center (aTDC) and to burn all injected fuel in diffusion combustion state. However, nowadays application of Exhaust Gas Recirculation (EGR) and pilot injection are standard to reduce NO_x emission by reducing the combustion temperature. Besides, alternative combustion methods are also started to be studied to reduce the NO_x and soot emissions.

Homogenous Charge Compression Ignition (HCCI) and Low Temperature Combustion (LTC) are the most common naming for the advanced combustion methods. Researchers have been utilizing these naming according to their convention. The main goal of the advanced combustion studies under different naming was to prepare more homogenous mixture, which prevent soot formation, before the start of combustion and perform the combustion below the NO_x formation region. Although researchers achieved to reduce both soot and NO_x emission simultaneously by applying advanced combustion, some results of the engine was deteriorated. The common disadvantages of the tried advanced combustion methods were; high fuel consumption, HC and CO emissions, lack of control of start of ignition and only applicable at low and medium load conditions.

Researches have been performed to improve these drawbacks by modifying combustion chamber (UNIBUS), injector type (NADI), fuel type (RCCI), injection configuration (HCCI, PCCI, PREDIC MULDIC, MK, HCLI), and EGR rate (LTC). However, limited improvements were achieved in diesel applications of the advanced combustions. Although, there are more study for the application of advanced combustion methods for gasoline engines that aim to increase the compression ratio under naming Controlled Auto Ignition (CAI), the progress is also limited for gasoline engines.

Researchers that aim to develop clean diesel combustion have started to focus on expanding the premixed combustion phase by preparing more homogenous charge than conventional combustion instead of having fully homogenous charge before the start of combustion. Injection parameters were modified to increase the premixed combustion conditions in the overall combustion process. Promising results were achieved under partially premixed combustion method. These results strengthened motivation of the thesis to study advanced combustion methods that enable cleaner diesel combustion.

Purpose of the presented thesis is to propose fuel injection strategy by improving the understanding of the alternative diesel combustion methods available in the literature. A new injection configuration was developed with the help of the experimental evaluation that reduced the emissions without sacrificing efficiency of diesel engine. Five set of experiments were performed in the scope of the thesis to gain experience about the details of advanced combustion methods and propose applicable engine operation configuration for clean diesel combustion.

First four set of experiments were performed at chassis dynamometer on a Ford Fiesta that satisfy EURO-5 emission limit with its 1.5-liter diesel engine. Experiments were conducted with pure diesel and biodiesel blended diesel without modifying injection parameters at; full load conditions, part-load conditions, New European Driving Cycle (NEDC), and Worldwide Harmonized Light Vehicles Test Cycle (WLTC). Evaluation of the results of these experiments enable to determine the effects of different fuel on emission, performance, and combustion. Besides, engine operation points under regulation cycles, total consumed fuel and energy, total emitted exhaust gases, and differences of the NEDC and WLTC were evaluated. According to the investigated results, it was evaluated that the use of 10% biodiesel has the most balanced improvement among the blended volumes 5, 10, 20, 30 and 50%. The use of 10% biodiesel enable to reduce CO, HC and PN respectively 30%, 20% and 25% while 5% and 2% increase was observed at NO_x emission and fuel consumption. On the other hand, it was evaluated that the WLTC has more disturbed and higher engine load condition compared to the NEDC. This led 3% higher fuel consumption under the WLTC than the NEDC. Moreover, exhaust gas temperature was higher under the WLTC compared to the NEDC due to lower idling period of the WLTC. This resulted in high Diesel Oxidation Catalyst (DOC) conversion efficiency and therefore more than 60 percent lower CO and HC emission was observed under the WLTC compared to the NEDC. On the other hand, the WLTC was more sensitive to use of biodiesel than the NEDC, because emissions and fuel consumption of the WTLC was more affected from biodiesel blended diesel compared to the NEDC. Besides, outcomes of the first four set of experiments enabled to define appropriate engine operation conditions for advanced combustion experiments that can be widely utilized during the operational life of the engines.

Fifth set of experiments were performed by modifying fuel injection parameters at selected engine operation conditions. Optimization study was performed at 2000 rpm, which is the engine load condition of the cruise speed of the NEDC and common operation point of the WLTC. Fuel injection configuration was modified to investigate alternative advanced combustion methods. More than 300 different experiments were performed at 2000 rpm while injecting 6.6 mg diesel per stroke. Partially premixed combustion was achieved among conducted experiments that improved both emissions and efficiency. Results of the pure diesel experiments showed that it was possible to reduce fuel consumption 30% while reducing the NO_x and opacity 35% and 60%

respectively by injecting the fuel very early under heavy EGR compared to the original working condition of the engine. Similar results were also obtained for double and triple injection configurations. Besides, contribution of biodiesel to advanced combustion method was investigated in the single injection strategy. Use of biodiesel for early injection also improved the power and soot compared to the original working condition of the engine. However, higher oxygen content of the biodiesel worsen the NO_x emission and reduce the ignition delay effect of EGR compared to pure diesel. This caused less power and NO_x improvement with biodiesel blended diesel compared to the pure diesel. On the other hand, it is found that optimization study for biodiesel blended fuel should be performed separately as start of combustion is affected from biodiesel.

In addition to 6.6 mg injection tests, a moderate load condition was also studied to investigate the applicability of heavy EGR under high load conditions. Optimization study that aims to achieve partially premixed combustion with more homogenous fuel-air mixture was conducted. More than 50 experiments were executed to generate the map of emissions, performance, and combustion parameters according to start of injection and EGR rate for single injection of 12 mg diesel. The optimum condition among 50 experiments had 30% higher power, 32% less NO_x and 20% less opacity compared to the original working condition. Therefore, promising emission and performance was achieved that can be utilized for high load conditions. Moreover, evaluated results enable to distinguish different advanced combustion methods.

In the last section of the thesis, a range extended hybrid electric vehicle model of the tested Ford Fiesta was proposed by adapting the new advanced combustion method. Energy consumption of the range extended vehicle was calculated by implementing the energy conversion efficiencies of the components that are necessary to convert the conventional diesel vehicle to hybrid electric vehicle. Proposed combustion method enable to achieve EURO-6 emission with EURO-5 exhaust layout. Original vehicle consumed 1125 ml diesel and had 8607 mg NO_x emission under the WTLC, while proposed range extended vehicle that run with advanced combustion had 861 ml diesel consumption and 703 mg NO_x emission according to model. The results showed that fuel consumption and emissions of the vehicle was improved compared to the conventional vehicle by applying advanced combustion methods.

The originality of this thesis arise from proposing a partially premixed combustion in which utilization of EGR improve efficiency while reducing NO_x . It is discovered that the effect of EGR enabled to achieve more efficient operation compared with conventional operation point of the tested engine while reducing both NO_x and soot emissions. Moreover, prepared range extended electrical vehicle power consumption analysis showed that it is possible to satisfy EURO-6 emission limits by reducing fuel consumption by using diesel engine that has EURO-5 exhaust configuration. This may attract the OEMs to utilize diesel engines for hybrid electric vehicle applications and regain the popularity of the diesel engines for future applications.



DİZEL İÇTEN YANMALI MOTORLARDA YANMANIN DENEYSEL İNCELENMESİ

ÖZET

Dizel motorlar, benzinli motorlara göre daha az yakıt harcadıkları için günlük yaşamda oldukça popülerdir. Çünkü dizel motorlar, benzinli motorlara kıyasla hem kısmi hem de tam yük koşullarında daha yüksek verim elde etmeyi sağlayan daha yüksek sıkıştırma oranına sahiptir ve kısılma kaybı yoktur. Ancak, Avrupa Komisyonu tarafından beyan edilen NO_x ve kurum limitlerini sağlayabilmek için ek egzoz bileşenlerinin (DPF, SCR) kullanılması gerekmektedir.

Öte yandan, elektrikli araçların satışları, içten yanmalı motorların sera gazı etkisini azaltmak ve petrol yakıtlarına bağımlılığı azaltmak için neredeyse dünyadaki tüm ülkeler tarafından teşvik edilmektedir. Örneğin, Avrupa Komisyonu, Paris Anlaşması kapsamındaki taahhütnamelere ulaşmak için binek otomobillerinden kaynaklanan CO_2 emisyonlarını azaltmak için bir yasa önerisi getirmiştir. Avrupa Birliği, sıfır araç filolarının 2030 yılında 2021'e kıyasla ortalama emisyonlarında %30 azalma önermiştir. AB, hedefi gerçekleştirebilmek için üreticilerin sattığı araç filolarındaki hibrid elektrikli araç payını öngörülen gösterge seviyesinden (2025'te %15 ve 2030'da %30) daha fazla arttırmayı sağlayan üreticilere teşvik verecektir. Ayrıca, Almanya ve Fransa gibi birçok Avrupa ülkesi, elektrikli ve hibrit araçların pazardaki payını artırmak için yalnızca içten yanmalı motoru bulunan araçların doğrudan satışlarını sınırlamak için yeni düzenlemeler yaptığını açıklamıştır. Ayrıca, AB, 2009/28/EC direktifini açıkladıktan sonra, standart dizel yakıtına hacimce %10 oranına kadar ilave alternatif yakıt eklenmesine izin vererek, petrol yakıtının sera etkisini azaltmak için biyodizel kullanımını teşvik etmektedir.

Çoğu araç üreticisi filolarına tamamen elektrikli, hibrid ve / veya menzili genişletilmiş elektrikli araçlar ekleyerek yeni çıkan düzenlemeleri karşılamayı planlamaktadır. Bununla birlikte, benzinli motorların kullanımı, hibrid uygulamalar için dizel motorlardan daha popülerdir, çünkü benzinli motorların egzoz sistemi dizel motorların egzoz sisteminden daha ucuz ve daha az karmaşıktır.

Araştırmacılar ve otomotiv şirketleri, emisyon düzenlemelerinin yürürlüğe girmesinden bu yana dizel motorlarda yanma sürecinin detaylarını araştırmaktadırlar. İlk homojen dizel yanma, dumanı öldürmek için 1957'de denendi. Şimdiye kadar çok sayıda bilim insanı, benzinli (homojen) ve dizel (yüksek sıkıştırma oranı, kısılma kayıpsız) motorlarının avantajlarını birleştirmeyi amaçlayan farklı türde dizel yanma yöntemleri geliştirdi.

Öte yandan, konvansiyonel dizel yanmasının tanımı yasal emisyon limitleri ve müşteri beklentileri nedeniyle 90 yıllardan günümüze değişmiştir. Önceden, tüm yakıt, üst ölü noktadan (aTDC) sonra yaklaşık 5 krank mili açısında (KMA) maksimum silindir içi basıncını elde etmek için üst ölü noktadan (bTDC) 10-15 KMA önce bir seferde enjekte edilip ve neredeyse enjekte edilen yakıtın tamamı difüzyon yanma durumunda yanmaktaydı. Ancak, günümüzde NO_x emisyonunu azaltmak için EGR akışı ve parçalı

püskürtme ile ön karışimli yanma safhası elde etmek standart uygulama haline gelmiştir. Ayrıca, NO_x ve kurum emisyonlarının üstesinden gelmek için alternatif yanma yöntemleri de incelenmeye başlanmıştır.

Homojen Dolgulu Sıkıştırma ile Ateşlemeli (HCCI) ve Düşük Sıcaklıkta Yanma Metotları (LTC), gelişmiş yanma yöntemleri için kullanılan en yaygın isimlendirmelerdir. Birçok akademik çalışma kendi kabul ettikleri konvansiyonlara göre bu isimlendirmeleri kullanmıştır. Farklı isimler altında yapılan ileri yanma çalışmalarının temel amacı, yanma başlamadan önce daha homojen bir karışım hazırlayarak kurum oluşumunu engellemek ve NO_x oluşum sıcaklık bölgesinin altında yanma işlemi gerçekleştirmektir. Araştırmacılar, gelişmiş yanma uygulayarak hem kurum hem de NO_x emisyonunu aynı anda azaltmayı başarsa da, motorun bazı performans ve emisyon sonuçları bozulmuştur. Farklı adlandırmalar altında denenmiş yanma yöntemlerinin ortak dezavantajları; yüksek yakıt tüketimi, HC ve CO emisyonları, ateşleme başlangıç anının kontrolünün eksikliği ve sadece düşük ve orta yük koşullarında uygulanabilir olmasıdır.

Bilim insanları bu dezavantajları iyileştirmek için yanma odasını (UNIBUS), enjektör tipini (NADI), yakıt tipini (RCCI), enjeksiyon konfigürasyonunu (HCCI, PCCI, PREDIC MULDIC, MK, HCLI) ve EGR oranını (LTC) değiştirerek araştırmalar yapmıştır. Ancak, gelişmiş yanma yöntemlerinin dizel uygulamalarında sınırlı gelişme sağlanabilmiştir. Benzinli motorlarda, Kontrollü Otomatik Ateşleme (CAI) adı altında sıkıştırma oranını arttırarak verimi arttırmayı amaçlayan ileri yanma yöntemi denemeleri dizele göre daha popülerdir ancak benzinli motorlar içinde ilerleme sınırlı olmuştur.

Temiz dizel yanmasını geliştirmeyi amaçlayan araştırmacılar, yanmaya başlamadan önce tamamen homojen bir karışım yerine, geleneksel yanmaya göre daha homojen bir karışım hazırlayarak ön karışimli yanma fazını genişletmeye odaklanmıştır. Dizel yanma sürecinde ön karışimli dizel yanma miktarını arttırmak için enjeksiyon parametreleri değiştirilmiştir. Kısmen önceden karıştırılmış yanma yöntemiyle umut verici sonuçlar elde edilmiştir. Bu sonuçlar, gelişmiş yanma yöntemi geliştirerek daha temiz dizel yanması sağlamayı hedefleyen tezin motivasyonunu güçlendirmiştir.

Sunulan tezin amacı, literatürde mevcut olan alternatif dizel yanma yöntemlerini detayını anlamak ve yeni bir yakıt enjeksiyon stratejisi önermektir. Bu amaç doğrultusunda deneysel değerlendirmelerin yardımı ile dizel motorun verimliliğinden ödün vermeden emisyonları azaltan yeni bir enjeksiyon konfigürasyonu geliştirilmiştir.

Gelişmiş yanma yöntemlerinin detayları hakkında deneyim kazanmak ve temiz dizel yanması için uygulanabilir bir yanma yöntemini önermek üzere beş farklı deney seti yapılmıştır

İlk dört deney seti, 1.5 litrelik dizel motoru olan EURO-5 emisyon şartlarını karşılayan bir Ford Fiesta ile şasi dinamometresinde gerçekleştirilmiştir. Enjeksiyon parametrelerinde değişiklik yapmadan hem saf dizel hem de biyodizel karışimli dizel ile yapılan ilk deneyler; tam yük koşulları, kısmi yük koşulları, Yeni Avrupa Sürüş Döngüsü (NEDC) ve Dünya Çapında Uyumlu Hafif Araçlar Test Döngüsüdür (WLTC). Bu deneylerin sonuçlarını değerlendirerek farklı yakıtların emisyon, performans ve yanma üzerindeki etkileri belirlenmiştir. Ayrıca, çevrim testlerindeki motor çalışma noktaları, toplam tüketilen yakıt, harcanan enerji ve egzoz gazı emisyonları hesaplanmıştır. NEDC ve WLTC çevrimlerinin farklılıkları ortaya konulmuştur. İncelenen sonuçlara göre hacimce %10, %20, %30 ve %50 oranında

harmanlanmış biyodizel dizel karışımları arasında en dengeli iyileştirmenin %10 biyodizel kullanımı ile elde edilmiştir. %10 biyodizel kullanımının, CO, HC ve is emisyonlarını sırasıyla %30, %20 ve %25 oranında azalttığı, NO_x emisyonu ve yakıt tüketiminde ise sırasıyla %5 ve %2 artışa sebep olduğu gözlemlenmiştir. Öte yandan, WLTC'nin NEDC'ye kıyasla daha değişken ve yüksek motor çalışma noktalarına sahip olduğu değerlendirilmiştir. Bu sebepten, WLTC testindeki yakıt tüketimi NEDC'den %3 daha fazla olduğu ölçülmüştür. Ayrıca, egzoz gazı sıcaklığının NEDC'ye kıyasla WLTC çevriminde daha yüksek olduğu gözlemlenmiştir çünkü WLTC'de daha az rölantide çalışma süresi bulunmaktadır. Bu sayede yüksek Dizel Oksidasyon Katalizörü (DOC) dönüşüm verimliliği sağlanmıştır ve WLTC testinde NEDC'ye kıyasla %60 daha düşük CO ve HC emisyonu oluşmuştur. Öte yandan, biyodizelin kullanımının WLTC testindeki emisyon ve performansla olan yüzdesel etkisi, NEDC testinin emisyon ve performansına olan etkisine kıyasla daha yüksek olduğu gözlemlenmiştir. Ayrıca, ilk dört deney grubunun sonuçları, motorların kullanım ömrü boyunca yaygın olarak kullanılabileceği ve gelişmiş yanma deneyleri için uygun olan motor çalışma koşullarını belirlemeyi sağlamıştır.

NEDC ve WLTC çevrimlerinde motorun sık çalışma koşulundan seçilen motor çalışma devrinde yakıt enjeksiyon parametreleri değiştirilerek beş farklı grup deney yapılmıştır. NEDC testindeki sabit seyir hızına denk gelen ve WLTC testinde de yaygın çalışma noktası olan 2000 devir/dakikada optimizasyon çalışması yapılmıştır. Konvansiyonel dizel yanmasına alternatif gelişmiş yanma yöntemleri önerebilmek için yakıt enjeksiyon konfigürasyonu değiştirilerek deneyler yapılmıştır. 2000 devir/dakikada enjekte edilen yakıtın (çevrim başına 6.6 mg) sabit tutulduğu 300'den fazla deney yapılmıştır. Yapılan deneyler sonucu hem emisyonların azaldığı hem de verimin arttığı, yakıtın çoğunun ön karışimli yanma fazında yandığı dizel yanması elde edilebilmiştir. Konvansiyonel dizel çalışma koşullarına kıyasla yüksek EGR bulunan yanma odasına yakıtın çok erken püskürtülmesi yöntemi ile yakıt tüketiminin %30'dan fazla azaldığı, NO_x ve kurum emisyonlarının da sırasıyla %35 ve %60 oranında azaldığı gözlemlenmiştir. Tüm enjeksiyon konfigürasyonları için benzer sonuçlar elde edilebilmiştir. Saf dizel çalışmalarına ek olarak, tek püskürtmeli enjeksiyon stratejisinde biyodizel katkılı yakıtın gelişmiş yanma yöntemine olan etkisi de incelenmiştir. Yakıtın tek seferde yüksek EGR bulunan yanma odasına çok erken püskürtüldüğü konfigürasyonda biyodizel kullanımının da motorun orijinal çalışma koşullarına kıyasla gücü ve kurum emisyonunu iyileştirdiği gözlemlenmiştir. Ancak biyodizelin içerdiği yüksek oksijen konsantrasyonundan dolayı NO_x emisyonunu artırdığı ve EGR kullanımı ile sağlanabilen uzun tutuşma gecikmesi süresini azalttığı gözlemlenmiştir. Biyodizel ile de motorun orijinal çalışma durumuna göre iyileştirme sağlanabildiği ancak tutuşma gecikmesi ve NO_x emisyonuna olan olumsuz etkisinden dolayı gelişmiş yanma yöntemi çalışmasında saf dizele kıyasla daha az iyileştirme sağlayabildiği anlaşılmıştır. Ayrıca biyodizelin tutuşma gecikmesini etkilediğinden saf dizel ile yapılan optimizasyon çalışması sonucu elde edilen enjeksiyon parametrelerinin birebir uygulanmaması gerektiği, biyodizel harmanlanmış yakıt ile ayrı bir optimizasyon çalışması yapılması gerektiği anlaşılmıştır.

2000 devir/dakikada çevrim başına 6.6 mg dizel enjekte edilerek yapılan testlere ek olarak, yüksek miktarda EGR bulunan yanma odasına erken püskürtme yapılarak elde edilen iyileştirmenin daha yüksek motor gücü koşullarında da uygulanabilir olduğunu göstermek için 2000 devir/dakikada çevrim başına 12 mg yakıt enjekte edilerek deneyler yapılmıştır. Deneylerde püskürtülen yakıtın büyük kısmının ön karışimli yanma fazında gerçekleştirilebilmesi için optimizasyon çalışması yapılmıştır. Farklı

EGR oranı ve püskürtme zamanında 50'den fazla deney yapılarak 2000 devir/dakika, 12 mg çalışma noktasının EGR ve püskürtme zamanına göre emisyonların, performansın ve yanma parametrelerinin haritasını çıkartılmıştır. Deneyler sonucu elde edilen en optimum çalışma noktasının motorun orijinal çalışma noktasına kıyasla %30 daha yüksek güce, %32 daha az NO_x ve %20 daha az opasiteye sahip olduğu gözlemlenmiştir. Böylece, önerilen enjeksiyon konfigürasyonunun orta-yüksek yük (10kW) koşullarında da emisyonu azaltıp ve performansı artırabileceği görülmüştür. Ayrıca değerlendirilen sonuçlar, yaygın olarak bilinen gelişmiş yanma yöntemlerinin farklarının anlaşılmasını da sağlamıştır.

Tezin son bölümünde, test edilen Ford Fiesta'nın menzil uzatıcılı hibrit elektrikli taşıt modeli oluşturulmuştur. Modelin emisyon ve yakıt tüketim sonuçları dizel motorun geliştirilen yanma yöntemi ile çalıştığı varsayımı ile hesaplanmıştır. Konvansiyonel bir aracın hibrit elektrikli araca dönüştürülebilmesi için gerekli bileşenlerin (elektrik motoru, motor sürücü, batarya) enerji dönüşüm verimleri hesaba katılarak menzil uzatılmış aracın enerji tüketimi hesaplanmıştır. Model sonuçları önerilen yanma yöntemi ile çalışan hibrid aracın, EURO-5 egzoz sistemine sahipken EURO-6 emisyonunu sağlayabileceğini göstermiştir. Orijinal aracın WTLC testindeki toplam yakıt tüketimi ve NO_x emisyonu salınımı sırasıyla 1125 ml 8607 mg'dır. Önerilen gelişmiş yanma yöntemi ile çalışan menzili uzatılmış elektrikli aracın ise aynı çevrimi 861 ml dizel tüketip ve 703 mg NO_x emisyonu salarak tamamlayabileceği hesaplanmıştır. Elde edilen sonuçlara göre, yakıt tüketimi ve emisyonların geleneksel araçlara kıyasla daha iyi hale getirilebileceği görülmüştür.

Tezin özgünlüğü, EGR kullanımının dizel motorunun verimini artırırken, NO_x emisyonunun azalttığı bir dizel yanması önermesinden kaynaklanmaktadır. Yüksek oranda EGR kullanımı yakıtın ön karışım yanma fazında yanmasını sağlayarak motorun orijinal çalışma durumuna kıyasla daha verimli bir çalışma sağlarken, NO_x ve kurum emisyonlarını da azalttığı keşfedilmiştir. Dahası, menzili uzatılmış elektrikli araç modeli sonuçları; EURO-5 egzoz sistemine sahip bir dizel motorun önerilen yöntem ile çalıştığında EURO-6 emisyon limitlerini sağlayabildiğini ve yakıt tüketimini azaltabildiğini göstermiştir. Bu imkân araç üreticilerinin hibrit elektrikli araç uygulamaları için dizel motorları kullanmalarının önünü açabilir ve gelecekteki uygulamalar için dizel motorların kullanımını artırabilir.

1. INTRODUCTION

Internal combustion engines are being utilized more than 100 years as the power source of the transportation. As internal combustion engines consume the limited fuel resources, fuel prices are continuously increasing because of the insufficient resources of the fossil fuels. On the other hand, exhaust gases that are emitted from internal combustion engines cause global warming and environmental pollution, which are serious problems for all human being.

Governments started to apply emission limits for the passenger cars to reduce the harmful effects of the exhaust emissions on the environment since 1990s [1]. Besides, automotive companies have been conducting researches on internal combustion engines for lowering the fuel consumption and exhaust emissions mainly to reduce the impact of the rises in fuel price and to achieve the mandated exhaust emission limits.

However, sales of the vehicles that have internal combustion engine increasing day by day and emission limits that applied by the governments cannot preserve the environment from the harmful effects of the exhaust emissions. In order to overcome this global environmental problem, 195 countries accepted the first-ever universal, officially binding global climate deal in December 2015 at the Paris climate conference (COP21). The United States Energy Division release very strict emission rules for light duty passenger cars. Additionally, the European Commission introduced a legislative proposal to reduce CO₂ emissions from passenger cars to achieve the European Union's (EU) commitments under Paris Agreement [2]. These progresses indicate that most of the vehicles will be hybrid until 2050 [3]. Moreover, many European countries, such as Germany and France, declared new regulations to limit the direct sales of the vehicles that only have internal combustion engine to fulfill commitments of the Paris Agreement [2].

Although, automotive companies, research centers and universities continue to study on the development of alternative solutions such as battery electric vehicle and fuel cell electric vehicles, proposed vehicles cannot provide long ranges as conventional vehicles that have internal combustion engines [4]. Necessity of complex and

expensive after treatment systems reduce the popularity of diesel engines. Therefore, reducing exhaust emissions via achieving clean combustion, especially for the diesel engines is still one of the major research topic.

Diesel engines have higher compression ratio than gasoline engines and they do not have gas exchange losses at part load conditions. These two key parameters are the reasons of the higher thermal efficiency of the diesel engine compared to gasoline engines. Moreover, fuel is directly injected to the combustion chamber of the diesel engines, therefore the air-fuel mixture is heterogeneous and combustion of the some portion of the injected fuel occurs in non-stoichiometric condition. Due to heterogenous and non-stoichiometric combustion, additional exhaust after treatment components are necessary for the removal of the NO_x and soot. On the other hand, combustion of the gasoline engines occurs with homogenous and stoichiometric air-fuel mixture. This enables removal of the NO_x emission with three-way catalyst, which also removes CO and HC. Besides, no soot emission is emitted at gasoline engines due to homogenous air-fuel charge in the combustion chamber.

Experiments were performed to combine the benefits of gasoline (homogeneity) and diesel (high compression ratio, no throttling loss) engines. In this way, conventional combustion mechanism of the diesel fuel was modified that enable to reduce NO_x and soot emissions. Researchers achieved to reduce both NO_x and soot emission via combining the homogeneity of gasoline engine and the compression ignition property of diesel engine. These combustion methods were named as Homogenous Charge Compression Ignition (HCCI) and Low Temperature Combustion (LTC)[5,6]. However, studied advanced combustion methods by modifying the conventional diesel combustion worsen the fuel consumption, HC and CO emissions. Furthermore, developed combustions methods cannot be applied under high load and transient conditions, because the high thermal energy of the early-injected fuel causes uncontrolled start of the combustion during the engine's compression stroke [7].

Researchers studied on these drawbacks of the HCCI and LTC and proposed Premixed Compression Ignited Combustion (PCI), in which early pilot injection and high EGR rates were utilized, to reduce NO_x and soot by increasing premixed charge in the chamber [8]. Another method to improve drawbacks of HCCI and LTC is Partially Premixed Compression ignition (PPCI), in which all fuel was injected before start of combustion [9]. Besides, fuel injection parameters, rate of exhaust gas recirculation,

inlet air temperature and pressure, and pistons were modified under advanced combustion methods to be able to achieve diesel combustion without mentioned drawbacks. However, conventional commercial vehicles that has diesel engine can satisfy the mandated emission limits still by utilizing complex exhaust systems.

Improvements on clean diesel combustion may regain the popularity of the diesel engine for future applications. Even accomplishing to run engine under advanced combustion condition for short periods by shifting from conventional diesel combustion can improve an engine's total emissions under a regulation cycle. At least, this enables to reduce the size of the exhaust after treatment equipment that are necessary for the removal of NO_x and soot.

1.1 Purpose of Thesis

Purpose of this thesis study was to investigate the combustion of compression ignition engine of a passenger car by modifying fuel injection parameters and EGR to improve the understanding of the principles of the combustion and pollutant formation mechanism of diesel engine. The research was planned to study different injection configurations with high levels of EGR to evaluate different advanced combustion methods that were mentioned in the literature and to examine the existing combustion process of the engine. Besides, experiments were performed, which are not available in the literature, via combining high EGR rate with split injection strategy and modifying start of injections' time in a wide range. In this way, a novel combustion method was studied on light duty diesel engine to improve the emissions while keeping the advantages of the diesel engine. The ultimate purpose of the experimental study on the 1.5-liter diesel engine of the passenger vehicle is to propose an alternative injection configuration, which enable to achieve homogenous low temperature combustion, that emit lower NO_x and soot compared to the original configuration without increasing fuel consumption.

In order to achieve the target, a diesel vehicle was purchased from the market and its existing engine control unit was replaced with a controllable engine control unit that allows manipulating and monitoring all engine actuators. In-cylinder pressure sensor and other necessary pressure and temperature sensors were instrumented on the engine to get useful data.

Once experimental set-up was prepared, advanced combustion methods were studied with different fuel injection configurations under heavy EGR at part load conditions of the engine, that were selected from the operation points of the regulation cycles (NEDC, WLTC). Besides, effects of biodiesel on combustion were also investigated to understand the improvement opportunities of biodiesel at advanced combustion applications. At the end, diesel range extended hybrid vehicle application model was simulated to indicate the potential of advanced diesel combustion methods for future hybrid applications

1.2 Literature Review

1.2.1 Conventional diesel combustion

Compression Ignition engines, that have 4-stroke cycle, burn the injected fuel through a heterogeneous process. This heterogeneous combustion process follows respectively ignition delay, pre-mixed combustion, diffusion burning and late combustion phases according to heat release diagram shown in Figure 1.1. The combustion process shown in Figure 1.1 has also been accepted as conventional diesel combustion process.

When the liquid fuel is atomized into the high-density combustion chamber environment, it begins to break down into small droplets and the transition from the liquid phase to the gas phase increases. In addition, the momentum of injected fuel spray has an important role in the preparation of the fuel-air mixture. As a result, the temperature of the liquid fuel increases and liquid fuel droplets begin to evaporate, especially at the spray boundaries. No chemical reactions occur in this period. After evaporation of some portion of the injected fuel, the first stage of the self-ignition process starts around 750 K. At this stage, temperature of the combustion chamber is relatively low compared to temperatures at the diffusion burning stage. Slow oxidation of the fuel during the first stage, together with the production of free chemical radicals, increases the pressure inside the cylinder by a small amount. In parallel with this chemical mobility, the fuel, which is mixed with hot gases, continues to increase its temperature. The time elapsed from the start of injection to the start of combustion is defined as the phase of ignition delay [10–12]. After the ignition delay phase, self-ignition phase starts. This step is usually referred to as preliminary mixing phase. The amount of fuel burned at this stage is directly correlated to the interval of the ignition delay phase. The observed heat release peak at premixed combustion, which is shown

in Figure 1.1, is due to the rapid (in a few crank angle degree) burn of the fuel in its flammability limits [10–13]. Diffusion combustion (controlled combustion) starts after premixed combustion phase. During this phase in-cylinder temperatures reaches to very high levels. Lack of time depending on high in-cylinder temperature prevent evaporating the fuel and mixing it with air for fully oxidation. Therefore, soot emission formation starts under these circumstances. However, during the fourth combustion phase (late combustion), soot and unburned fuels that formed during the diffusion phase on the cylinder walls is burned.

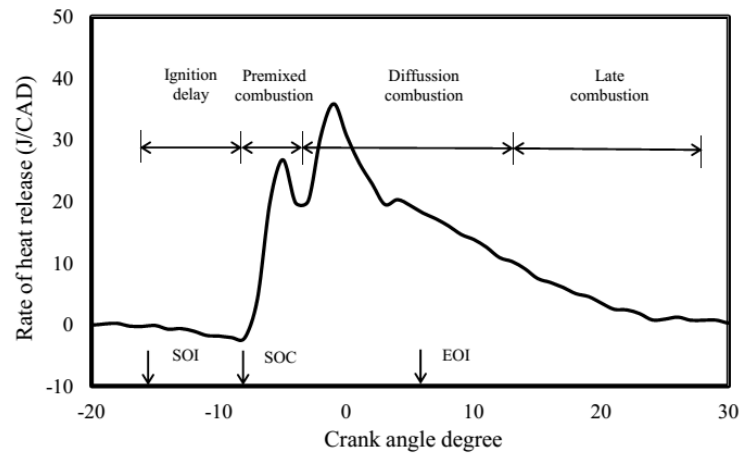


Figure 1.1 : Heat release rate and phases during diesel combustion [13].

Figure 1.2 demonstrates the NO_x and soot formation conditions according to the temperature (T) and equivalence ratio (Θ). The soot and NO_x formation regions were sketched according to the numerical results of Akihama et al. and Kitamura et al. [14,15]. The path of T and Θ were adopted by Kook et al. for the conventional combustion [16]. In the conventional diesel combustion, at the beginning of the cycle, fuel is not spread in the chamber and reactions start in a rich air-fuel zone, as can be seen in Figure 1.3. The fuel and air first react in a fuel-rich mixture and then this rich mixture burns out at the jet periphery in diffusion flame [17]. Besides, Figure 1.3 shows an overview of a diesel fuel jet and its approximate temperature distributions and the occurrence of pollutant emissions. Once fuel is spread in the combustion chamber, local equivalence ratio drops almost to the stoichiometric ratio. The progress of the combustion completed after passing diffusion flame region. Consequently, the air-fuel mixture and the temperature distribution in the combustion chamber during the conventional combustion lead NO_x and soot formation as shown in Figure 1.2 and Figure 1.3.

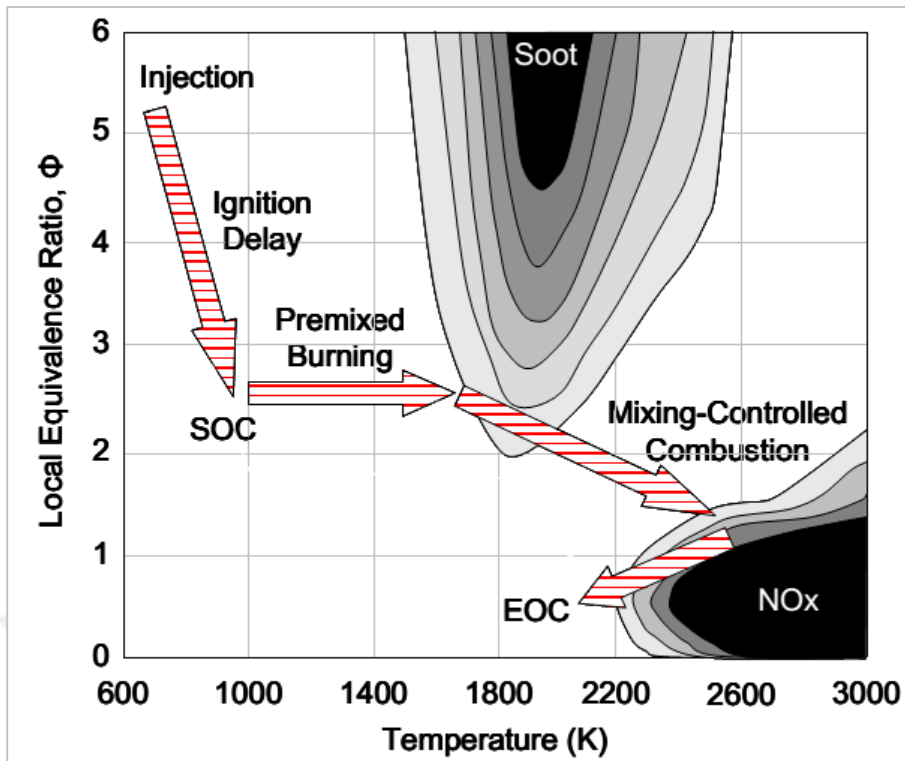


Figure 1.2 : Soot and NO_x formation and combustion regions at T- Φ [18].

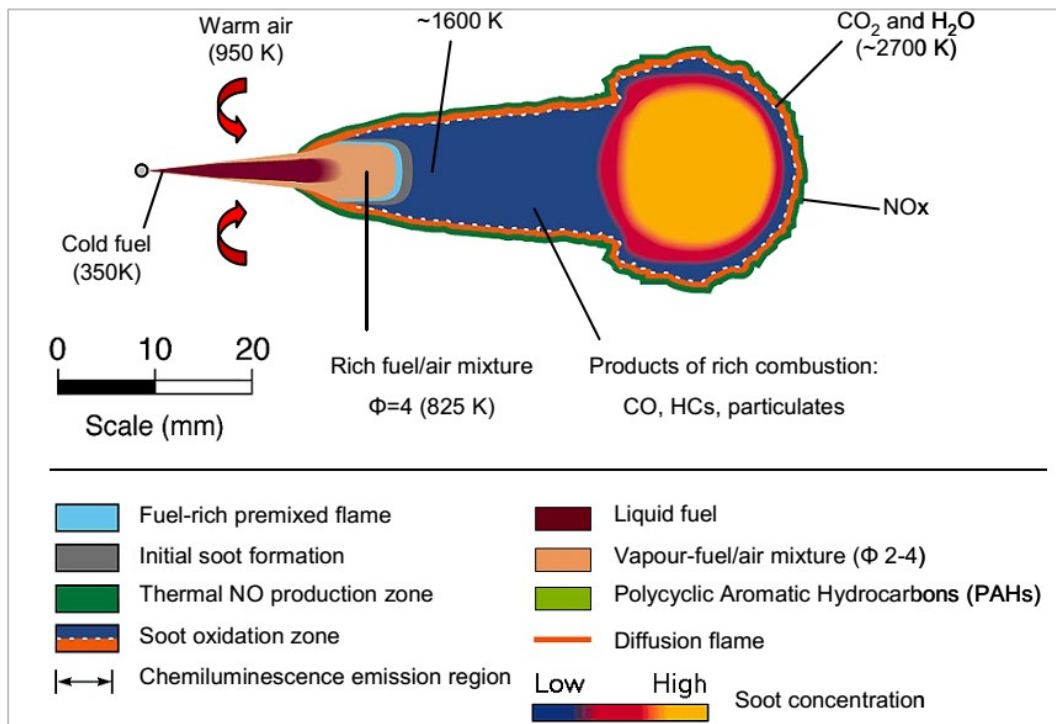


Figure 1.3 : Schematic of quasi-steady burning jet from laser measurements [18,19].

1.2.2 Alternative diesel combustions

The development of cleaner diesel combustion process have gained importance since the introduction of diesel engine emission standards. Advanced combustion strategies have either tried to find in-cylinder solutions in order to fulfill the emission standards and therefore eliminate the necessity of the after treatment systems or at least to reduce the performance requirements of the after treatment systems and thus reduce cost and complexities of them. Although reducing NO_x emission is the focus of the combustion system, there is also a significant interest in reducing soot emissions [20].

M. Alpertstain et al. performed the first homogenous diesel combustion study by utilizing diesel fumigation in the intake port in 1958 to reduce smoke emission [21]. Later, researchers studied on alternative diesel combustion methods under different naming to reduce exhaust emissions. The most known and accepted alternative low temperature and homogenous combustion methods are listed in Table 1.1.

Table 1.1 : List of early advanced combustion methods.

Advance Combustion Method	Reference
Active thermo-atmosphere combustion (ATAC)	[22]
Toyota-Soken combustion (TS)	[23]
Compression-ignited homogeneous charge (CIHC)	[24]
Homogeneous charge compression ignition (HCCI)	[5]
Active radical combustion (ARC)	[25]
Premixed lean diesel combustion (PREDIC)	[26]
Homogeneous charge compression ignition (HCCI)	[27]
Uniform bulky combustion system (UNIBUS)	[28]
Homogeneous charge intelligent multiple injection combustion system (HiMICS)	[29,30]
Multiple stage diesel combustion (MULDIC)	[31]
Premixed compression ignited combustion (PCI)	[8]
Modulated kinetics (MK)	[6,32]
Narrow Angle Direct Injection (NADI)	[33]
Partially Premixed Compression Ignition (PPCI)	[34]

Homogenous Charge Compression Ignition (HCCI) which is named as the main cluster for advanced diesel combustion [7,35] was first tried in 1989 by Thring [5]. Main idea of the HCCI combustion strategy is to prepare homogenous air-fuel mixture as possible as before the start of combustion like in gasoline engines. PREDIC, HiMICS, UNIBUS, MULDIC, PCI, and NADI are the developed different early

injection strategies and available in literature under HCCI technique. Late injection strategies were also studied to propose alternative diesel combustion with the name of MK, HCLI and HPLI [36,37]. On the other hand, Low Temperature Combustion (LTC) is also being used as a general term for advanced combustion method instead of HCCI [20,38–40]. First, Kimura et al. defined their alternative diesel combustion study as low temperature combustion [32]. Many researches have been performed under the naming of LTC on both light-duty and heavy-duty engines [32,41–43]. Although various researches were conducted under different names, the ultimate aim was to change the chemical chain reactions to prevent NO_x and soot formation by reducing combustion temperature while improving the homogeneity level of the air-fuel mixture.

In a similar manner; Akihama et al. simulated the formation mechanism of the NO_x and soot at 3D-CFD KIVA2 to clarify the alternative combustion paths that can be proposed instead of conventional diesel combustion. They plotted the time history of the injected fuel for smokeless (low temperature) combustion and high smoke (high temperature) combustion in Figure 1.4 at the scales of temperature (T) and equivalence ratio (Θ). They stated that smokeless combustion can be achieved at low temperature and equivalence ratios below the formation region of NO_x and soot [15]. Overall combustion temperature can be reduced by utilizing high EGR. Imtenan et al. stated that the use of high EGR reduces NO_x and soot formation, however has a penalty of higher CO and un-burnt HC emissions [44].

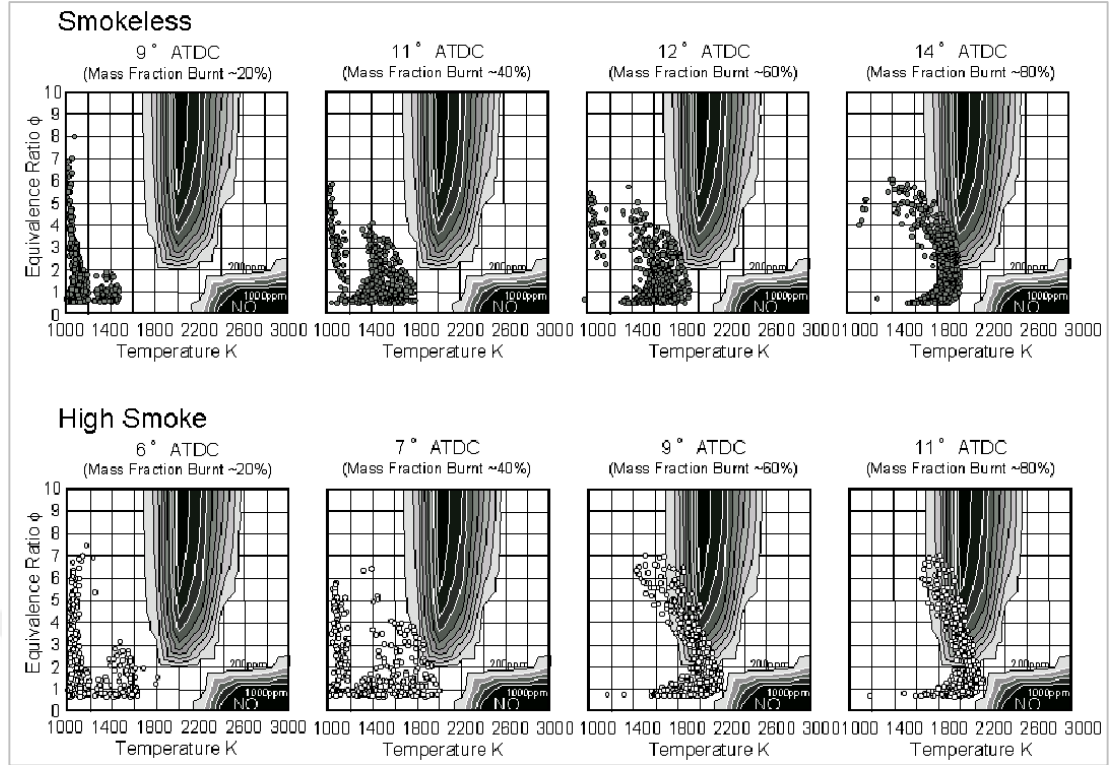


Figure 1.4 : Time histories of equivalence ratio (Θ) and temperature conditions for the cases of high and low smoke [15].

After researches identified the soot and NO_x formation regions with conventional diesel combustion path, alternative diesel combustion paths were also sketched on same plot as shown in Figure 1.5 [16,45]. The black arrows are denoted for HCCI combustion process. Kinetics of the chemical reactions dominate the phases of the combustion under HCCI method. Homogenous mixture of the air-fuel is completed before the start of combustion that reduces the equivalence ratio below one as shown in Figure 1.5. HCCI combustion depends on the auto-ignition of the charge in the cylinder at multiple locations. De Ojeda et al. presented the other alternative path with the gray arrows as LTC method, however Kook et al. and Keeler denoted the grey arrows as PCI [16,18,45]. Under this combustion method, injection parameters dominate the combustion phasing. The equivalence ratio of the mixture was rich at the beginning of the combustion and lean equivalence ratio was achieved after the premixed combustion [16].

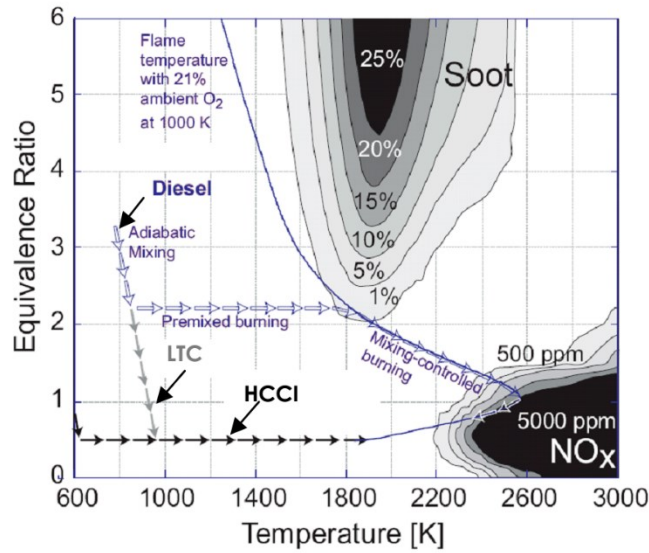


Figure 1.5 : Demonstration of different combustions on equivalence ratio-temperature plot [45].

In addition to alternative combustion paths, researchers studied on flame temperature of the diesel combustion [16,46,47]. Kook et al. calculated the flame temperature of the stoichiometric mixture that was plotted with solid blue line in Figure 1.5. They stated that the flame temperature reduced when EGR rate was increased in the combustion chamber due to less available oxygen ratio in the combustion chamber [16]. Keeler adapted this statement to the contours in Figure 1.6, which represent the peak temperatures of the air-fuel mixtures at 1000 K for different ambient oxygen concentration. EGR reduced the peak temperature in the combustion chamber as show in Figure 1.6 [18].

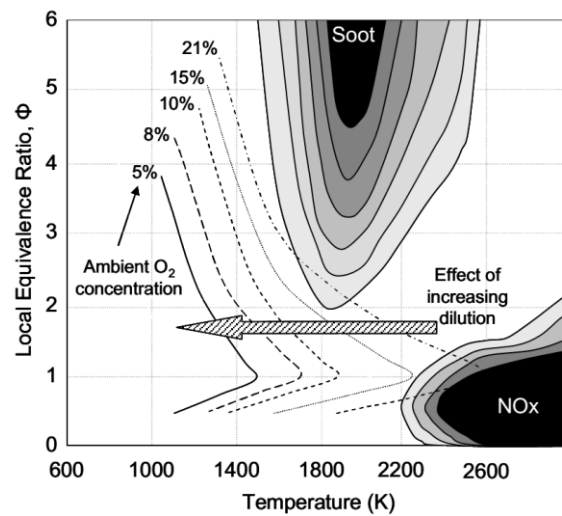


Figure 1.6 : Representative Θ -T diagrams for a fuel element mixed to equilibrium with the ambient air initially at 1000 K for varying ambient oxygen concentrations [18].

The mentioned studies, which identified the formation of NO_x and soot and the paths of diesel combustion process, have drawn the road map for the researches to reduce the emissions of NO_x and soot. Numerous scientists and companies performed modification on the injection configuration, EGR rate, boost temperature, injector type and combustion chamber to achieve a path that did not cross the emission formation regions.

1.2.2.1 Naming convention for alternative diesel combustion methods

Alternative diesel combustion methods were studied to achieve low NO_x and soot and they utilized different naming for their approaches as given in previous section in Table 1.1. The utilized naming and details of these methods are described in this section. Figure 1.7 shows the injection configuration of the different methods that are widely known in the literature. Although injection configuration of these methods are given in Figure 1.7, researches conducted additional modifications on injector, combustion chamber or EGR rate to achieve low NO_x and soot simultaneously.

The blue rectangular for each method shows the crank angle range of the injection. Yellow icon was utilized to indicate the crank angle of the start of combustion. Different methods are positioned from conventional to port fuel injection (PFI), going from bottom to top, temperature of the combustion is decreasing both with the effect of early injection and utilized high EGR. Besides, homogeneity of the air-fuel mixture is increasing from bottom to top due to early fuel injection and longer ignition delay duration. Therefore, both NO_x and soot can be reduced by going upper methods with the effect of low in-cylinder temperature and higher homogenous mixture. However, power, CO and HC drawbacks of the methods are not shown in the plot and these disadvantages are reported for all these methods.

Homogenous charge compression ignition (HCCI) was the first defined naming for alternative combustion that was shown with fumigation, PFI and HCCI in the Figure 1.7. Once researchers revealed that fuel penalty of the HCCI was too high, they focused on other alternative methods. The Low Temperature Combustion (LTC) is the general name for the other methods (PREDIC, MULDIC, UNIBUS, PCI, MK, PPCI) that shown in Figure 1.7. PCI and PREDIC are similar methods that have early injection to achieve start of combustion before TDC. Whereas, late injection were performed at MULDIC, UNIBUS and MK to achieve start of combustion after TDC.

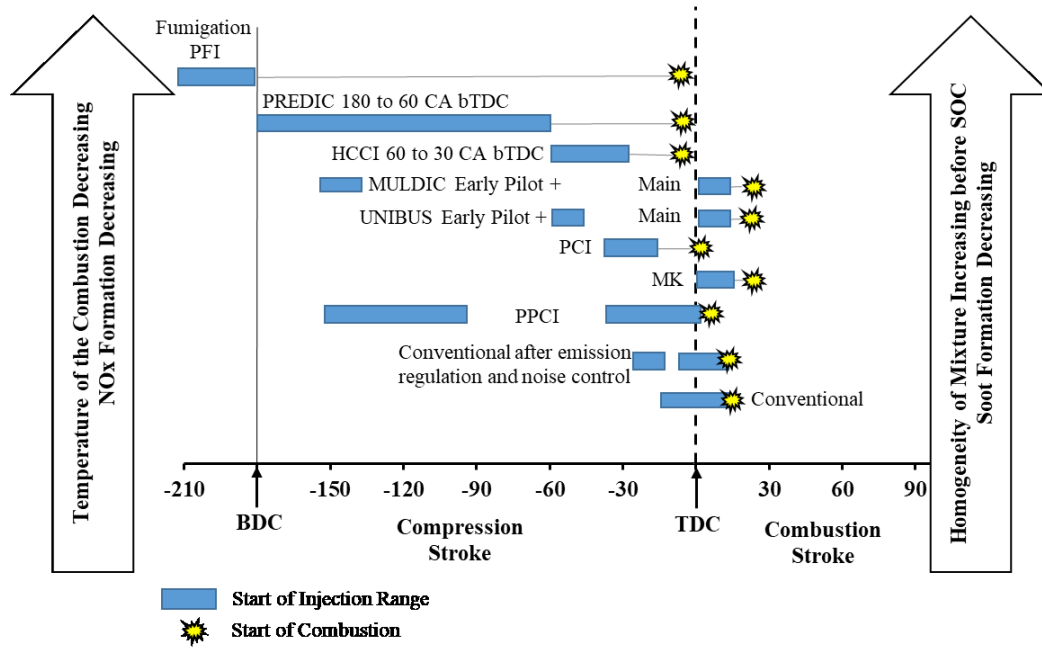


Figure 1.7 : Injection configuration of different alternative combustion methods.

1.2.2.2 HCCI

When a homogeneous mixture of air-fuel is created instead of the mixture of the classic heterogeneous combustion in the diesel engine chamber, the pressure and temperature increase in the compression cycle will create a different ignition mechanism than in the conventional combustion. The ignition will not start in one part of the fuel spray, it will spontaneously occur in the entire combustion chamber at the same time. In this way, the flame front is not visible at high temperature as is the case with spark ignition engines because automatic ignition will occur at the same time in the entire cylinder. The absence of a high temperature flame prevent formation nitrogen oxides. In addition, the homogeneous mixture in the cylinder prevents the formation of rich regions and therefore prevents the formation of soot.

The common label Homogeneous Charge Compression Ignition (HCCI) combustion was utilized via many researchers that investigated similar combustion concepts for gasoline and diesel engines [5,48–50]. HCCI concept aims to combine the homogenous mixture advantage of Otto cycle and auto ignition advantage of the diesel engine. In this way, both soot and NO_x emissions can be reduced while keeping high efficiency cycle [10,51].

Christensen tried HCCI via port fuel injection (PFI) in 1997 [48]. In their study, they achieved to reduce the NO_x around 100 times than conventional combustion in

addition to very low soot emission. However, there was fuel evaporation problem due to low temperature at the inlet manifold. As fuel cannot evaporate, it adheres to the inlet port, cylinder wall that increase the HC and CO emissions. Researchers concluded that, it was not possible to evaporate fuel properly in the port injection, so PFI studies did not continue [48].

Garcia et al. studied the difference of the HCCI and conventional diesel in their paper [52]. Some of the outcomes of this study are summarized as follows. In HCCI method, the start of fuel injection timing at high loads can be advanced to an ideal timing with the utilization of cold EGR. When the EGR ratio is constant, the increase in the inlet temperature leads to an earlier start of combustion. Regardless of the rate of utilized EGR, in the HCCI method, the soot emissions are low enough to be ignored. Very low NO_x emissions can be achieved in engines operating under HCCI combustion with EGR. In general, HC and CO emissions are higher compared conventional combustion because early injection in HCCI combustion mode lead wetting of fuel to cylinder walls [52]. In addition, the high level of HC and CO emissions was the result of incomplete HCCI combustion, resulting in higher fuel consumption compared to conventional combustion [17].

1.2.2.3 LTC

In conventional diesel combustion, increasing EGR will decrease NO_x and increase soot. However when high EGR rate is applied, combustion temperature decrease below 1400K and thus both NO_x and soot can be reduced simultaneously. Formation of NO_x emission can be suppressed by reducing the temperature inside the combustion chamber via utilizing high EGR rate. Once the temperature of the combustion reduced below 1700K, combustion do not occur in the NO_x and soot formation region that was defined in Figure 1.5, which is defined as LTC [53,54].

Low temperature combustion was developed to maximize the operation load of advanced combustion methods like HCCI. Engine run at much richer level fueling at LTC compared to HCCI, with a local equivalence ratio of up to 4, which is below 0.45 for HCCI. This is achieved by applying very high rates of EGR, usually up to 55% and 65% by volume [53].

Additionally, combustion can also be performed without formation of NO_x and soot by keeping local equivalence ratio and flame temperature below 2 and 2200K

respectively, which is also defined as LTC. The local equivalence ratio and combustion temperature are moved into different regions in the ϕ -T map via adjusting EGR and fuel injection, in this way formation of NO_x and soot can be decreased at the same time [55,56].

LTC which is also defined as dilution-controlled combustion, is different from conventional (controlled via air-fuel mixture) and HCCI combustion (dominated via air-fuel chemical kinetics), as it was performed under high EGR rate (more than 60%) and uniform air-fuel mixture was formed before SOC. Besides, injection strategy controls the diesel combustion that is dissimilar from the air-fuel chemical kinetics in HCCI combustion [55].

Many researches were also performed experiments under the naming LTC. High level of EGR were utilized in these studies to develop uniform charge in the cylinder. Although researchers achieved LTC and reduced NO_x and soot simultaneously, fuel penalty were observed in their studies [57–60]. Usman Asad et al. studied LTC for compression ignition engines to discover the operability limits and challenges. They performed experiment by varying environmental and combustion related parameters without making hardware modification on a 2 liter diesel engine. They found that, mixing time of the charge is crucial for LTC and high levels of EGR and boost pressure are necessary to reduce the soot and to increase combustion efficiency [61].

1.2.2.4 PCI – PREDIC

Premixed compression ignited combustion (PCI) and Premixed lean diesel combustion (PREDIC) are very early fuel injection strategies that was tried by Iwabuchi et al. [8] and Yoshinaka Takeda et al. [26] respectively. They utilized different start of injection (SOI) to mix the fuel and air before start of combustion (SOC) and they intended to achieve SOC at TDC.

In PREDIC method of Yoshinaka Takeda et al. [26], fuel was injected between approximately 80 Crank Angle (CA) Before Top Dead Centre (bTDC) and 60 CA bTDC for lambda 2.7. They utilized side injectors to accomplish early injection. They achieved to reduce NO_x ten times lower than the lowest concentrations of a standard diesel operation. However, they reported that HC and CO emissions increased due to lean mixture of PREDIC. In addition to high CO and HC emissions, it was stated that

combustion can only be performed at low loads and ignition-timing control is very hard [26].

Iwabuhci et al. from Mitsubishi Motor Company developed Premixed Compression Ignited (PCI) method, where fuel was injected at 45 CA bTDC that was closer to TDC compared to PREDIC. Impinged-spray nozzle injector was utilized in their study to achieve low spray penetration and high fuel diffusion. They accomplished to reduce both NO_x and smoke emissions via utilizing early fuel injection strategy. Unfortunately, like other alternative combustion methods, HC emission was higher compared to conventional diesel and the method could not be applied under high load conditions [8].

1.2.2.5 MULDIC - MK - UNIBUS

MULDIC, MK and UNIBUS are introduced in this section as all these strategies have late injection that aim to postpone the SOC after top dead center (aTDC) as shown in Figure 1.7.

Hashizum et al. developed MULTIPLE stage Diesel Combustion (MULDIC) method by splitting the injection into two parts. First injection was almost half of the total injection amount and at 150 CA bTDC. Second injection was executed within the range of 2 bTDC to 30 aTDC. Premixed lean combustion observed for the first injection and this reduces NO_x emissions. On the other hand, diffusion combustion occurred under high temperature and low oxygen condition during the second stage. Late injection of the second part of the fuel allowed extending the operation range of the MULDIC to higher loads than PREDIC method. Furthermore, simultaneous reduction of NO_x and PM was possible with retarding the second stage injection timing with MULDIC operation. However, fuel consumption was higher compared to the conventional combustion [31].

Uniform Bulky Combustion System (UNIBUS) developed by Toyota Motor Corporation as a novel combustion method by Yanagihara et al at 1996 [28]. First, they tried single injection strategy. Later in 2003, Hasegawa and Yanagihara developed a double injection technique with a common rail injection system as shown in Figure 1.8. In order to create homogenous mixture, they utilized early injection. Late fuel injection was performed to activate combustion of all injected fuel. Hasegawa and Yanagira reported that the timing and quantity of the early injection have to be between

54 bTDC and 36 bTDC with 5 mm³/st and second injection (25 mm³/st) was fixed to the 3 CA aTDC according to their optimization study. They reported that combustion occurred suddenly and at the same time throughout the combustion chamber. Besides, they explained the achieved zero soot and low NO_x emissions with Non-luminous flame combustion and low combustion field temperature [62].

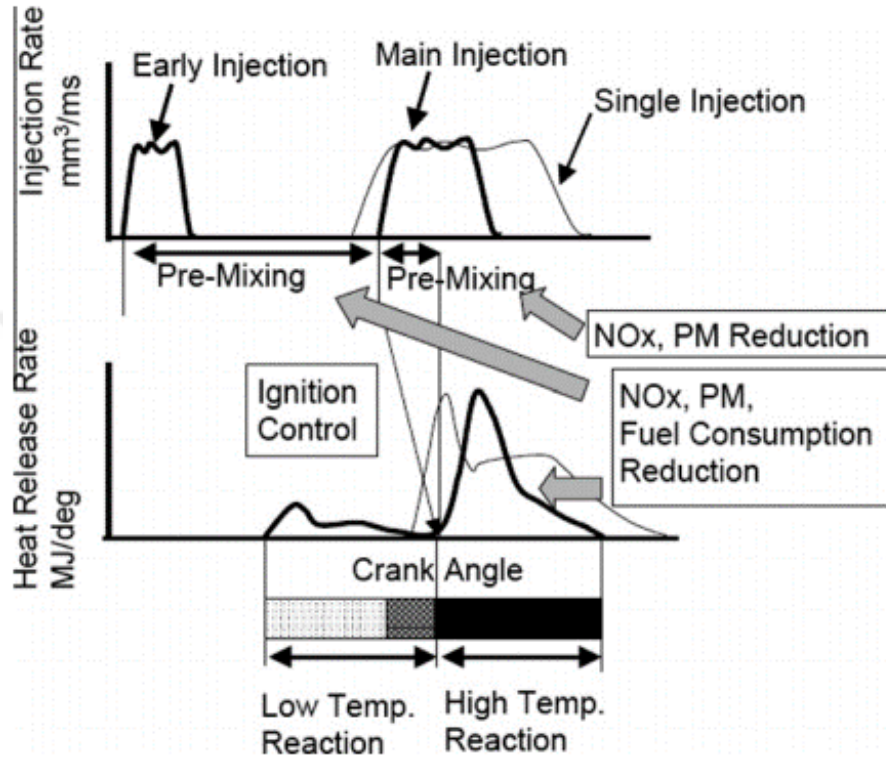


Figure 1.8 : Combustion strategy of UNIBUS [62].

Modulated Kinematics (MK) is low-temperature, premixed combustion system that was aimed to lower NO_x and soot via reducing combustion temperature below 1700K. MK was the best alternative among the listed methods in the Figure 1.7. Kimura et al. prepared the schematics given in Figure 1.9 to reveal the basic concept of the developed low-temperature combustion system.

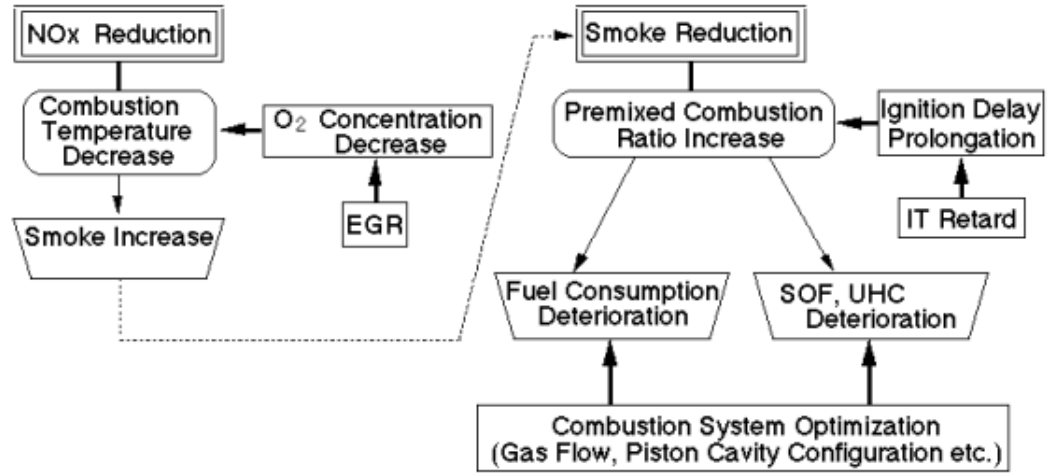


Figure 1.9 : Combustion concept of modulated kinematics [6,32].

MK concept started with increasing EGR ratio to reduce combustion temperature as formation of NO_x mainly dependent on the temperature. However, reducing oxygen concentration led higher smoke with diffusion combustion. In order to reduce smoke, premixed combustion method was tried that aim to mix the air and fuel thoroughly before the SOC. Adequate mixing duration was secured via prolonging the ignition delay duration, in this way injected fuel was diffused in the chamber to achieve premixed combustion. In addition to adapting EGR and injection timing, hardware modification was performed to improve the fuel consumption and HC deterioration. Combustion chamber shape was adopted and higher swirl ratio was generated in the scope of hardware modification. They accomplished to reduce NO_x by 90% and smoke emission below one (Bosch smoke number) via applying mentioned methods [6].

Kimura et al. studied more on MK combustion system to apply this method over a wide operating range. They focused to prolong the ignition delay duration and to shorten the injection duration. In order to achieve longer ignition delay duration, EGR was cooled and compression ratio was reduced, in this way the gas temperature in the combustion was decreased at TDC. Shorter injection duration was accomplished via increasing fuel pressure and nozzle diameter. This allowed expanding MK combustion region 1500 rpm 5 bmep to 2500 rpm 10 bmep [32].

1.2.3 Effects of biodiesel on combustion and emission

Alternative fuels are convenient substitutes to conventional fuels in the automotive industry, which reduces emission based air pollution and dependency on use of conventional fuels. Owing to the stringent emission regulations introduced by the European Union (EU), the use of biodiesels, as an alternative fuel, has become quite popular over the last decade [63,64]. Similarly, the directive of 2009/28/EC further encouraged the use of biodiesels by allowing the addition of alternative fuels to conventional diesels up to 10% in volume. Biodiesels have considerable benefits compared to their conventional counterparts as they are renewable energy sources, biodegradable, and non-toxic [38]. Moreover, their higher cetane number results in a reduction in the ignition delay duration (IDD) compared to conventional diesels [65]. In addition, facts of the higher O₂ in their chemical chain and the absence of both aromatics and sulfur in the chemical composition reduce the particle emission and lead to a more complete and cleaner combustion [66]. Despite the mentioned benefits of biodiesels, they have higher viscosity (than conventional diesels) that causes less atomization, and lower energy content because of high oxygen by weight [67].

Due to the benefits of biodiesels, they have been extensively studied to assess their improvements on compression ignition engines and significant results regarding their effect on emission and performance were obtained [68]. Earlier studies [69,70] showed that utilization of biodiesel causes a rise in the brake specific fuel consumption and a reduction in the thermal efficiency of the combustion due to their lower gross calorific value (GCV) than diesels (i.e., biodiesels reduce the power of engine and worsen the fuel consumption). On the one hand, it has been shown that utilization of biodiesel reduces Hydrocarbon (HC), Carbon monoxide (CO), and Particulate mass (PM) emissions due to their higher oxygen content and less aromatic compounds compared to diesels [71,72]. Furthermore, it has been presented that Nitrogen Oxides (NO_x) and Carbon Dioxide (CO₂) emissions of compression ignition engines increase with biodiesel addition to diesel [65,73].

In addition to effects of biodiesel on emissions and performance of diesel engine, the physical properties of the pure diesel and biodiesel were investigated to recognize their differences. Table 1.2 shows the physical and chemical properties of the pure diesel and biodiesel, which were also utilized in the experiments of this thesis. Although the biodiesel has similar properties with the diesel, there are some differences, which are

summarized in Table 1.2. The density, viscosity, and flash point temperature of pure biodiesel (B100) are higher than pure diesel (B00); on the other hand, the lubricity and GCV of B00 are higher than B100.

Table 1.2 : Physico-chemical properties of the diesel (B00) and biodiesel (B100).

Property	unit	EN590 Limit	B00	B100
Density (15 °C)	kg/m ³	820-845	831.8	883.7
Viscosity (40 °C)	mm ² /s	2.00-4.50	2.88	4.32
Fatty Acid Methyl Ester	% (v/v)	<7	<0.05	NA
Carbon Content	% (m/m)	0.30	<0.1	NA
Sulphur	mg/kg	<10	7.2	1.54
Flash Point	°C	>55	58.5	184
Lubricity	µm	<460	406	-
Cetane Number	-	>51	55.0	51.2
Gross Calorific Value (GCV)	cal/g	-	11074	9537
Oxygen content	(%weight)		0	10-12
Air–fuel equivalence ratio	-		~15	~12.5

Comparison of the properties listed in Table 1.2 are presented from the studies in the literature as below [74,75];

1. Biodiesel contain Oxygen in its chemical chain so ~11% of its weight is oxygen, which cause biodiesel to have lower energy density compared to diesel. In order to keep the same power of a diesel-fueled engine, higher amount of biodiesel needed to be injected.
2. In general, biodiesel has higher cetane number compared to diesel as it has lower number of twin bonds than diesel. Since, cetane number is directly related with the ignition delay duration, the higher the cetane number shortens the ignition delay duration.
3. Biodiesel has higher viscosity than diesel as fatty acid chain length and level of unsaturation of biodiesel is higher than diesel. As fuel injection parameters of the engine fuel system generated according to diesel's physical properties, precision of the fuel injection reduces and poorer atomization is observed.

4. Biodiesel has higher density compared to diesel. Conventionally the injectors determines the quantity of the fuel that sprayed in to the cylinder according to volume, therefore injected mass of biodiesel is more than diesel with conventional fuel systems
5. Higher flash point, make biodiesel safer compared to diesel.

Many researchers also used biodiesel under advanced combustion methods such as low temperature combustion (LTC), homogenous charge compression ignition (HCCI) and premixed compression ignition (PCI) to investigate effects of biodiesel on NO_x and particulate emissions under homogenous combustion. Tormos et al. [76] reported better emission under LTC with biodiesel compared to conventional diesel. Higher oxygen content of the biodiesel may improve the operation range of advanced combustion methods, as it is possible to use earlier start of injection (SOI) timing and higher EGR rate than conventional diesel combustion.

In order to present the results of more recent studies, which focused on the effects biodiesel, review of Xue at al. are given in Table 1.3 that summarized the studies on biodiesel by making increase, decrease and similar result groups. Most of the studies indicate that power performance, aromatic compounds, PM, CO, and HC emissions decrease, whereas economy performance, carbonyl compounds, NO_x and CO_2 emissions reduce with utilization of biodiesel as fuel [71]. Therefore optimizing fuel injection parameters according to biodiesel blended fuel may enable to improve the emissions and performance compared to use of biodiesel with injection parameters that were optimized for pure diesel.

Table 1.3 : Statistical analysis of different studies of biodiesel [71].

Property	Total # of reference	Increase		Similar		Decrease	
		Number	%	Number	%	Number	%
Power performance	27	2	7.4	6	22.2	19	70.4
Economy performance	62	54	87.1	2	3.2	6	9.7
PM emissions	73	7	9.6	2	2.7	64	87.7
NO _x emissions	69	45	65.2	4	5.8	20	29
CO emissions	66	7	10.6	2	3	57	84.4
HC emissions	57	3	5.3	3	5.3	51	89.5
CO ₂ emissions	13	6	46.2	2	15.4	5	38.5
Aromatic compounds	13	–	–	2	15.4	11	84.6
Carbonyl compounds	10	8	80	–	–	2	20

1.2.4 New generation vehicles and regulations

Electrical and hybrid vehicles are becoming more attractive due to stringent emission regulations and current trends in decreasing fossil fuel dependency [77]. Many European countries, such as Germany and France, declared new regulations to limit the direct sales of diesel vehicles in order to encourage the use of low-emission vehicles [78]. Most Original Equipment Manufacturers (OEMs) are planning to meet emerging regulations by incorporating fully electric, hybrid and/or range-extended electric vehicles in their fleets.

The European Union (EU) proposed 30% reduction for the average emissions of the fleet of first-hand vehicles in 2030 compared to 2021. Moreover, an incentive mechanism for the hybrid and pure electric vehicles were also proposed. EU will give incentive to the manufacturers, which achieve to increase the share of their hybrid and electric vehicles more than the projected benchmark level; 15% in 2025 and 30% in 2030 [2].

EU incentives due to Paris Agreement support the electric and hybrid vehicles sales. Sales of new electric car passed one million over the world in 2017, which is 54% higher than 2016. Moreover, the global electric car stock exceeded 3 million vehicles in 2017. Figure 1.10 shows the rise in electrical vehicle stock from 2013 to 2017, share of the countries and distribution of the battery electric vehicle (BEV) and Plug in Hybrid Electrical Vehicle (PHEV)[2].

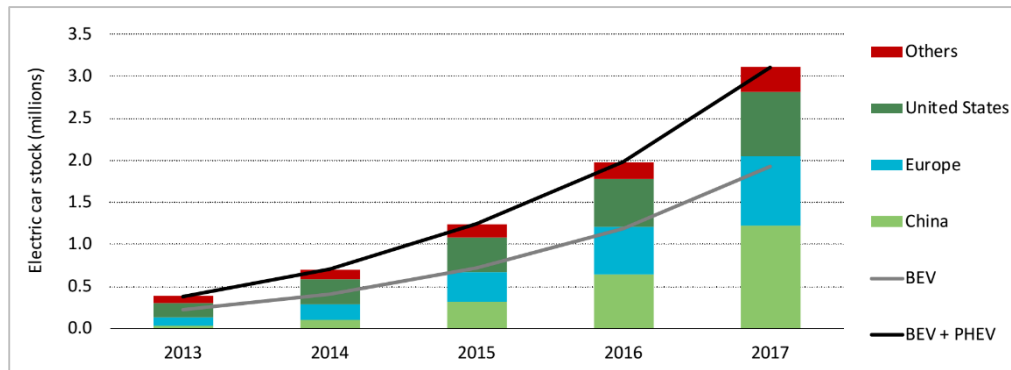


Figure 1.10 : Progress of the global electric car stock [2].

International Energy Agency released Global EV Outlook 2018 to reveal the current and future position of the electric and hybrid vehicles. There are two scenarios to predict the number of electric vehicles to 2030 namely; New Policies Scenario (the current and announced policies) and EV30@30 (portion of electric vehicles in market will be 30% of the total automobiles by 2030). Figure 1.11 shows both scenarios, where; PLDVs, LCVs, BEVs and PHEV are the abbreviations of passenger light duty vehicles, light commercial vehicles, battery electric vehicles, and plug-in hybrid electric vehicles, respectively. The number of electric light-duty vehicles extent 125 million by 2030 according to the New Policies Scenario, considering the current and announced policies. Whereas EV30@30 estimate 130 million pure electric vehicles and 90 million plug-in hybrid vehicles (total 220 million of electric light-duty vehicles) in 2030 as seen in Figure 1.11 [2].

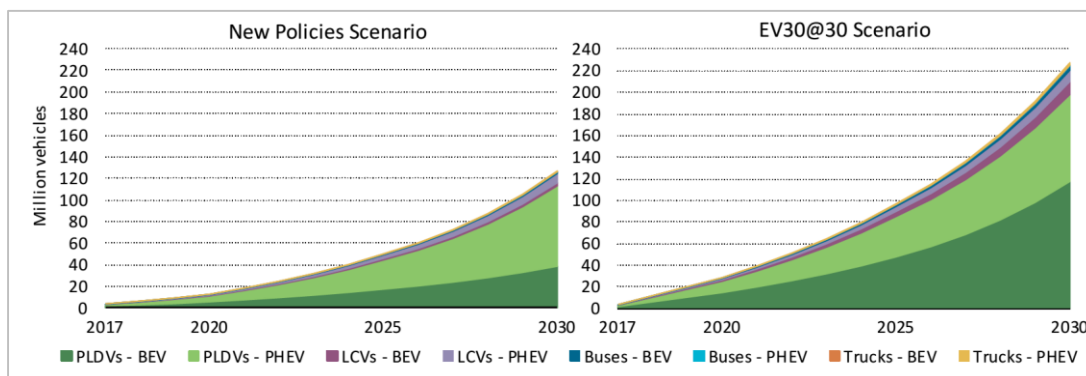
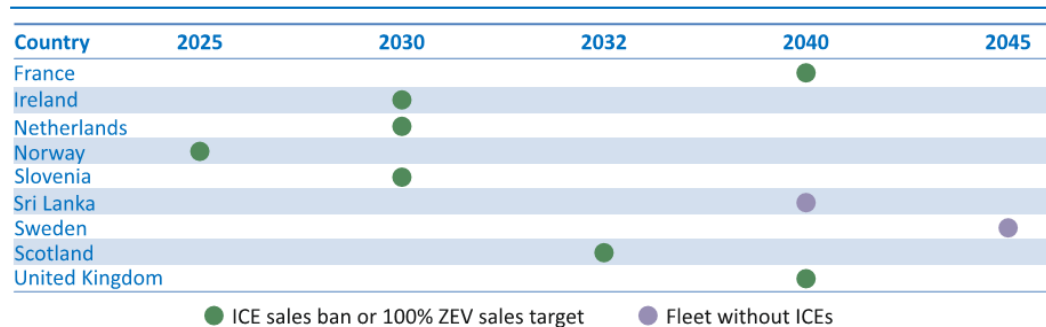


Figure 1.11 : Projection of electrical vehicle stock in the new policies and EV30@30 scenarios [2].

Moreover, many EU countries announced to end the sale of first-hand internal combustion engine vehicles to achieve electrical vehicle development commitment of the EU from the Paris agreement. Table 1.4 shows the countries that ban the sales of vehicles that only have internal combustion engine with application years. All these

national internal combustion engine prohibitions refer to announcements guaranteeing to finalize sales or registration of new gasoline and diesel vehicles (excluding PHEVs). In the case of Sri Lanka, the government is targeting to replace all vehicles with zero emission vehicles (ZEV) that has full electric or hybrid powertrain configuration. Therefore, hybrid vehicles will be more popular in the future and it is anticipated that special internal combustion engines will be developed for hybrid electric vehicles.

Table 1.4 : Announced sales prohibitions for internal combustion engine vehicles.



Although battery manufacturers are increasingly developing higher-capacity and cheaper products, the selling prices of long-range pure electric vehicles are still not comparable to conventional vehicles [79]. Due to high costs of batteries as well as charging infrastructure limitations, many OEMs began to develop range-extended electric vehicles that can travel long distances. As the fuel consumption and emissions of range-extended electric vehicles are calculated by the combination of pure electric driving distance and 25-km fuel-dependent driving, emission of them will be within the acceptable range of the EU legislation.

Most range-extended hybrid electrical vehicles have generators with gasoline engines to charge their battery and to supply energy for traction motors. Moreover, engineering companies (such as Lotus, Getrag, FEV and Mahle) developed prototype range-extended electrical vehicles with small gasoline engines [80]. Due to necessity of complex and expensive exhaust system to deal with NO_x and PM emissions, there is a limited number of studies regarding range-extended electric vehicles with diesel engine. Because of the additional after treatment catalysts, diesel range-extendors will be more expensive than gasoline range-extendors; thus, manufacturers do not prefer to use diesel engines as range-extendors despite their lower fuel consumption (compared to gasoline engines).

1.3 Scope of Thesis

This thesis proposes new fuel injection configuration with high EGR rate. The proposed engine operation condition relies on early fuel injection. Comparison of the proposed method and the available methods in the literature can be seen in Figure 1.12. The proposed three different injection configurations, which can be grouped under PPCI and PCI, are shown in the red dotted rectangular. Start of combustion for these configurations are before TDC, which is the difference from other PPCI and PCI methods. Proposed configurations are the outcome of the optimization study of early main injection and high EGR rate. The purpose of these injection configurations is to reduce NO_x and soot without fuel penalty compared to the original configuration of the light-duty vehicle.

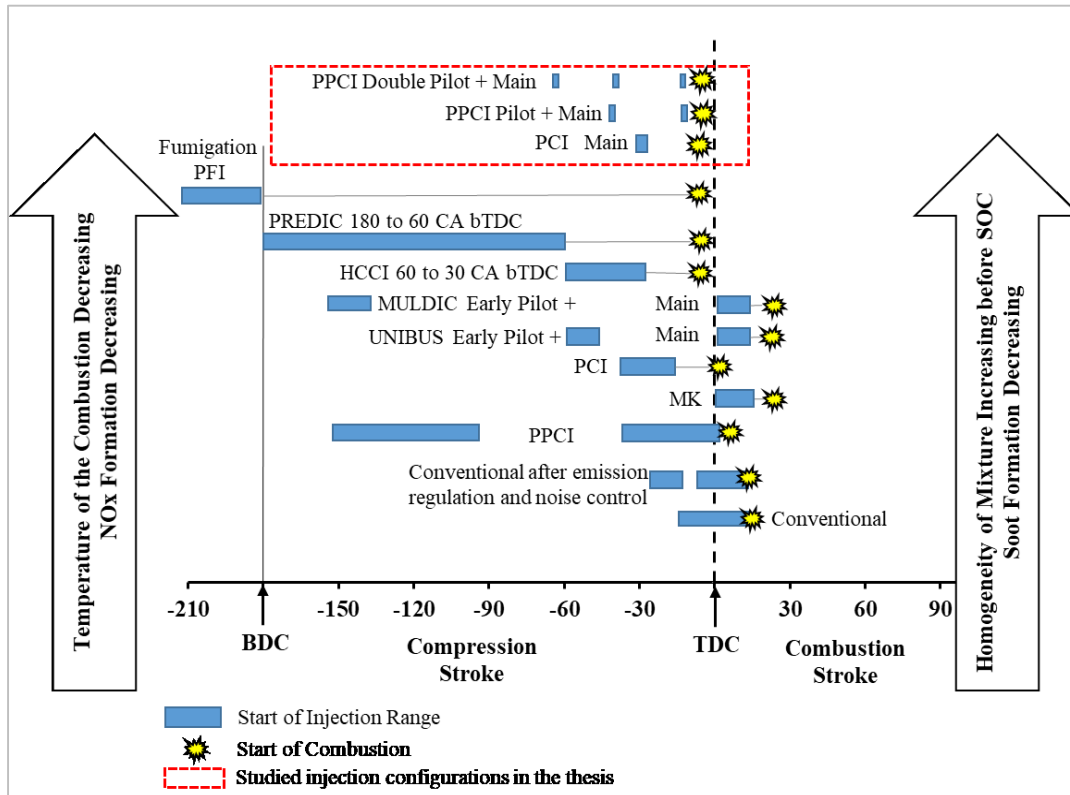


Figure 1.12 : Diesel combustion methods and proposed injection configuration.

Since, the study focuses on different diesel combustion methods, in the first section of the thesis an introduction is presented with a literature review on the history and progress of the diesel combustion. Besides, literature about the effects of biodiesel on combustion and future trends in the powertrains of the vehicles are discussed in the last part of the literature review.

After introduction section, experimental set-up section is prepared, as the experiments constitute a large part of the study. Measurement devices, engine dynamometer, chassis dynamometer, and test fuels are described in the experimental set-up section.

In the third section of the thesis, sequence of the experiments are listed in a table. Purpose of the each group of experiment are described. Besides, experimental procedure are given in this section after presenting the order of experiments to describe them clearly.

After describing the experimental set-up and procedure, results and discussion of each experiments are presented in the fourth section. Results of the experiments section has five sub-sections to show the results of each group of experiment.

In the fifth section, range extended diesel hybrid vehicle model is presented. Energy consumption and emission calculation of the range extended electric vehicle are shown in this section. Benefits of the proposed advanced combustion method are revealed by comparing the fuel consumption, NO_x and soot emissions of the range extended vehicle with conventional vehicle.

The last section of the thesis includes conclusions. Highlighted results of the experiments are presented briefly. Afterward, recommendations are discussed to achieve further improvement with proposed combustion and range extender application.



2. EXPERIMENTAL SET-UP

Experiments are the black bone of the thesis as all developed strategy to suggest a new advanced combustion method depend on the results of the experiments. Different type of testing were executed in both chassis and engine dynamometer of TÜBİTAK Marmara Research Center (MRC) during the study. TÜBİTAK MRC laboratories were built via the fund given by Ministry of Development. Chassis and Engine dynamometers have been working since 2011 and maintenance of the devices were performed annually. Certification tests can be performed in both dynamometers.

Table 2.1 : Utilized devices in the engine and chassis dynamometer.

Device – Sensor	Engine Dynamometer	Chassis Dynamometer
Engine Torque	HBM Torque Flange	Calculation
Wheel Power	X	Zöllner Load Cell
Vehicle Velocity	X	Zöllner PWM Sensor
Engine Speed	AVL Angle Encoder	Bosch PWM Sensor
In-Cylinder Pressure	AVL Indismart-612	AVL Indismart-612
Raw Emissions	AVL AMA-i60	AVL AMA-i60
Diluted Emissions	X	AVL AMA-i60
Particulate Number	X	AVL-489
Opacity	AVL-439	AVL-439
Air Flow	ABB SensyFlow	Bosch Maf Sensor
Fuel Consumption	AVL-735S	OBD2 data
Engine sensor data	Bosch ECU	OBD2
Control of engine	Bosch ECU	Bosch ECU
Temperature Sensor (extra)	AVL Puma	X
Pressure Sensor (extra)	AVL Puma	X
Fuel Conditioning	AVL-753	AVL-753
Engine Coolant Conditioning	AVL-553	Vehicle Fan and Cooler
Engine Oil Conditioning	AVL-554	Engine Oil Cooler
Sucked Air Conditioning	ACS	IMTECH

Although, engine dynamometer and chassis dynamometer have different measurement devices, some of them are common. Table 2.1 prepared to show the devices that were utilized during the experiments in both dynamometers. In addition, Table 2.2 shows the accuracy of the measurements that acquired from devices listed in Table 2.1.

Table 2.2 : Accuracy of the measurements.

Measurement	Device	Accuracy
Torque	HBM Torque Flange	$\pm 0.1\%$
Power	Zöllner Load Cell	$\pm 0.11\%$
Vehicle Velocity	Zöllner PWM sensor	$\leq \pm 0.2\text{CA}$
Engine Speed	AVL Encoder	$\leq \pm 0.1\text{CA}$
Engine Speed	Bosch PWM Sensor	$\leq \pm 0.5\text{CA}$
In-cylinder Pressure	AVL Indismart612	$< 0.2\%$
Injection timing	Angle Encoder	$\pm 0.1\text{CA}$
Fuel Injection Pressure	Delphi Pressure Sensor	$\leq \pm 1\%$
NO _x	Chemiluminescence Detector	$\leq \pm 1\%$
THC	Flame Ionization Detector	$\leq \pm 0.5\%$
CO ₂	Non-Dispersive Infrared Rays	$\leq \pm 0.5\%$
CO	Non-Dispersive Infrared Rays	$\leq \pm 0.5\%$
O ₂	Paramagnetic	$\leq \pm 0.5\%$
Particulate Number	AVL-489	$\pm 10\%$ for 1cm^{-3}
Opacity	AVL-439	$\pm 0.0025\text{m}^{-1}$
Air Flow	ABB SensyFlow	$< \pm 1.5\%$
Air Flow	Bosch Maf sensor	$< \pm 2\%$
Fuel Consumption	AVL-735S	$< 0.15\%$
Temperature Sensors	PT100 (K Type)	$\leq \pm 1\%$
Pressure Sensors	Can Open Transducer	$\leq 0.2\%$

2.1 Engine Dynamometer Experimental Setup and Control PC

Experiments were performed at Engine and Chassis dynamometer of the TÜBİTAK MRC Energy Institute. Engine dynamometer, which controlled by AVL Puma system, has an AC electrical motor brake with the power of 385 kW up to 8000 rpm and torque of 780 Nm up to 4500 rpm. Measurement and conditioning devices that are available in the engine dynamometer are shown in Figure 2.1. Torque was measured via HBM T10FS torque flange that was mounted on to the connection shaft of engine and electric motor. AVL-735S and AVL753C fuel devices were used to both measure fuel consumption and condition fuel's pressure and temperature. Sucked clean air by the engine was measured before entering compressor by ABB flow-meter. AVL AMA-i60 emission bench was used to measure NO_x, HC, O₂, CO and CO₂ emissions both before and after diesel oxidation catalyst (DOC). Instantaneous opacity after DOC was measured via AVL-439 Opacimeter. AVL Indismart 612 was used to measure the in-cylinder pressure and injection signal of the injectors. Glow plug of the second cylinder was replaced with a GH13P type piezoelectric sensor. Besides, Fluke-600 current clump was placed to the second injector's harness. Current flow over the harness for

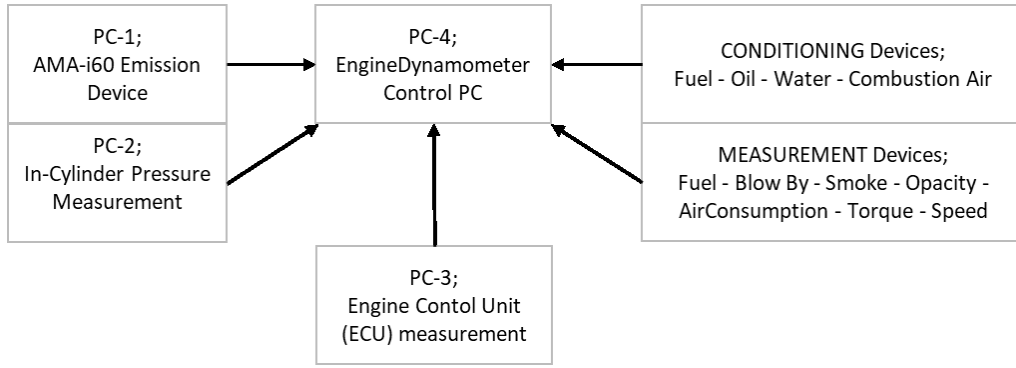


Figure 2.2 : Engine dynamometer control PCs and devices.

2.2 Chassis Dynamometer Experimental Setup and Control PC

Experiments on the test vehicle were performed in the twin axle chassis dynamometer of TUBITAK MRC Energy Institute. The chassis dynamometer can decelerate or accelerate the test vehicle by its 220-kW AC electric motors on its both axle. Exhaust emission measurement system of the chassis dynamometer has dilution tunnel, constant volume sampler, AVL AMA-i60 dilute and raw exhaust gas analyzers, AVL particulate number counter device, IMTECH test cell temperature and humidity controller, SADA wheel power and speed measurement computer, and driver aid computer. A schematic of the chassis dynamometer with measurement devices can be seen in Figure 2.3.

Driver aid computer is necessary for driver to follow up the transient cycle. It shows current and target vehicle velocity in time scale. Power, force and velocity of the vehicle were measured via chassis dynamometer computer. In addition, coast down parameters, which were selected from “simulated inertia and dyno loading requirements” table in UNECE R83 [81], were also applied from chassis dynamometer computer. Temperature and humidity of the chassis dynamometer were controlled with Imtech climatic system. All tests were performed at 24 ± 1 °C and 40 ± 5 percent relative humidity. Diluted emission measurement were performed according to UNECE R101 regulation by dilute AVL AMA-i60 analyzers. Besides, AVL AMA-i60 raw gas analyzers measured raw emissions of CO, HC, NO_x, and CO₂ before and after Diesel Oxidation Catalyst (DOC) to be able to realize instantaneous emission behavior during the NEDC.

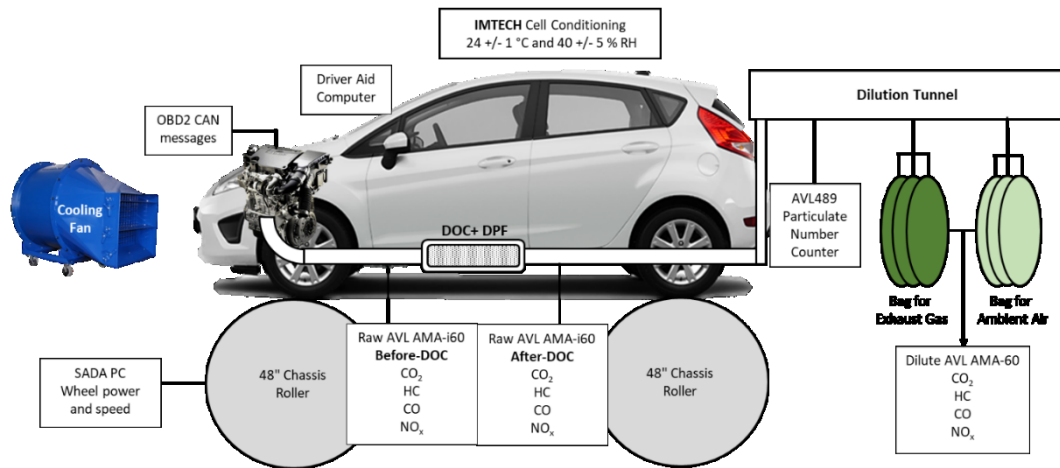


Figure 2.3 : Emission measurement configuration of the chassis dynamometer.

AVL diesel particulate number measurement device was utilized to record Particulate Number (PN) emission over the NEDC and WLTC as PN emission limit has been applied after September 2014 with Euro-5b emission regulation.

In addition to emission data, the On Board Diagnostic (OBD) port of the vehicle was utilized to acquire available engine sensor data. Condition of the vehicle was monitored and evaluated via OBD data during experiments. Original Equipment Manufacturers (OEMs) release CAN messages with the same ID via the OBD port to meet European OBD requirements for 2000 and later model year vehicles. Data of the engine sensors that were acquired from the OBD port are as follows; acceleration pedal position (APP), engine speed, manifold air pressure, air flow sucked by engine, exhaust gas recirculation (EGR) valve position, exhaust gas temperature before and after the diesel particulate filter, battery voltage, injection fuel pressure, total fuel injected to the cylinder and engine coolant temperature.

Figure 2.4 shows the real picture of the chassis dynamometer with test vehicle Ford Fiesta. Cables that lie on the floor are the OBD tool, Indismart-612, NI Data acquisition card, Fluke-600 and in-cylinder pressure cable. All equipment was placed into the vehicle during the experiments to acquire the data. The test vehicle has 4-cylinder 1.5 liter diesel engine. The engine has common rail injection system, cooled EGR and turbocharger without intercooler. Besides, engine has diesel oxidation catalyst (DOC) and diesel particulate filter (DPF) in the exhaust layout and can comply EURO-5 emission standards. The maximum torque and power of the engine is 75hP at 4000 rpm and 185 Nm at 1700 rpm.



Figure 2.4 : Photo of the chassis dynamometer.

Chassis dynamometer has less measurement devices and sensors compared to engine dynamometer. Figure 2.5 shows the PCs in the control room of the chassis dynamometer. As vehicle has its own conditioning systems for oil, fuel and water, additional conditioning devices are not necessary. The PC-3 in the left of the Figure 2.5 is for the air and humidity control of the test cell. The PC-2 in the middle of the Figure 2.5 measures the vehicle speed and force at the wheels of the vehicle. The PC-1 on the right side of the Figure 2.5 is the central PC and it has connection to all PCs in the system. There are 6 different PCs in the chassis dynamometer as shown in two separate figures in Figure 2.5 and Figure 2.6. PC-4 is for the AVL AMA-i60 emission measurement device. In-cylinder pressure measurement is performed by PC-5 in the Figure 2.6. PC-6 is available to control the engine parameters as it has connection to the ECU of the engine of the vehicle. PC-5 and PC-6 are not standard available measurements in the chassis dynamometer, because usual transient cycle tests are conducted without any modification on the engine. However, these measurements were performed during the experiments to be able to investigate the combustion under different injection parameters.

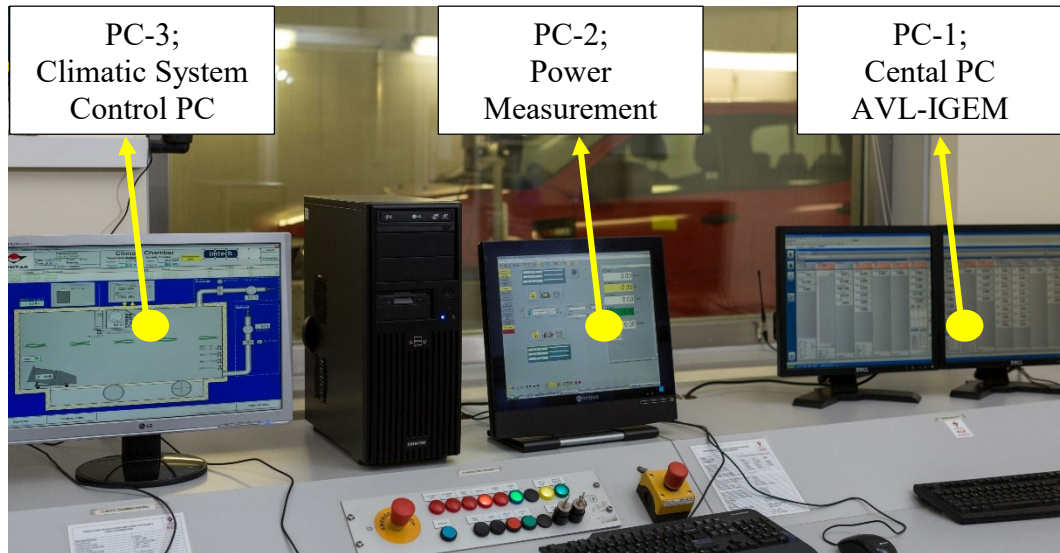


Figure 2.5 : Chassis dynamometer control PCs.

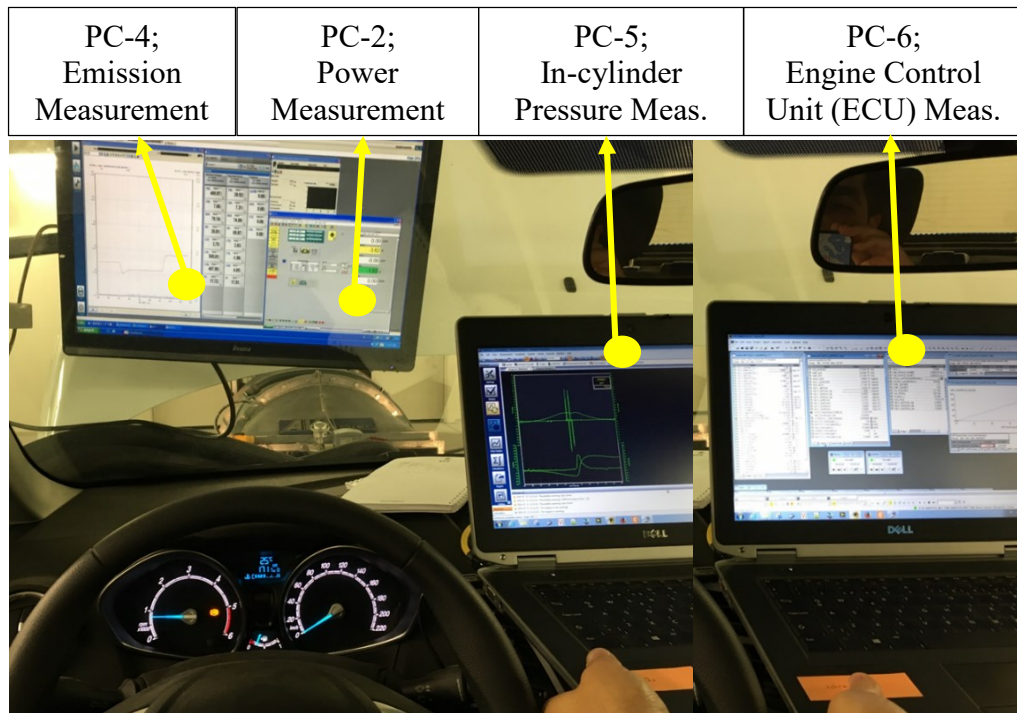


Figure 2.6 : Chassis dynamometer measurement PCs.

2.3 Biodiesel and Diesel Blending and Fuel Analyses

Turkish Government permits the use of diesel and biodiesel complying with EN590 and EN14214 standards, respectively. Diesel fuel that was necessary for experiments was bought from one of the gas stations of Shell Oil Company, whereas pure biodiesel, which was produced from canola oil, was supplied from DB Tarımsal Energy Company in İzmir.

Private companies cannot sell pure biodiesel in market, as the laws of Turkey forbid it. 15 liters of biodiesel fuel was obtained from DB Energy Company with official request of TÜBİTAK MRC for the purpose of to utilize in research and development studies.

Five blends of diesel and biodiesel with different volumetric proportions were prepared. The following definition is used for the blends: “B-XX”, where “XX” denotes the biodiesel percentage of the blend in volume. B00, B05, B10, B20, B30 and B50 were used in this study. Mixtures of these fuels were prepared according the volumes given in Table 2.3

Table 2.3 : Biodiesel additive percentage and necessary biodiesel quantity.

% of Biodiesel	0% (B00)	5% (B05)	10% (B10)	20% (B20)	30% (B30)	50% (B50)
Diesel (liters)	4.00	3.80	3.60	3.20	4.20	3.00
Pure Biodiesel (liters)	0.00	0.20	0.40	0.80	1.80	3.00
Total Fuel (liters)	4	4	4	4	6	6

The physical properties of the diesel blends were analyzed at the fuel analysis laboratory of TÜBİTAK Marmara Research Center (MRC) Energy Institute. The analyses results of pure biodiesel were acquired from the DB Tarımsal Energy Company. The physical properties of all test fuel are provided in Table 2.4. Although biodiesel had similar properties with the diesel, Table 2.4 summarizes the differences. The density, viscosity, and flash point temperature of pure biodiesel (B100) were higher than pure diesel (B00); on the other hand, the GCV of B00 were higher than B100.

In addition to the physical properties of biodiesel and diesel, the molecular formulas of diesel and biodiesel are different. Heywood provided the chemical formula of diesel as $C_nH_{1.8n}$ [11]. Furthermore, V. Bermudez et al., Salamanca et al., H.K Suh and C.S. Lee ,and B. Ozener et al. used similar molecular formula for diesel in their study, whereas, they provided the molecular formula of biodiesel as $C_{18.3}H_{34}O_2$, $C_{18.05}H_{34.9}O_2$, $C_{18.74}H_{34.43}O_2$ and $C_{18.74}H_{34.51}O_2$ respectively [66,73,82,83]. The percentage of biodiesel’s oxygen weight to total weight of the whole molecule is more than 11%.

Theoretical stoichiometric Air Fuel Ratio (AFR) of pure diesel and biodiesel are different as their molecular formulas are different. Molecular formula of diesel $C_{14.1}H_{25.8}$ were selected from the experimental studies. Molecular formula of biodiesel

$C_{18.4}H_{34.2}O_2$ was provided by DB Energy, which is calculated from fatty acid distribution. Then necessary oxygen amount for theoretical combustion of diesel and biodiesel were calculated according to equation (2.1) and equation (2.2). AFR for diesel and biodiesel calculated by dividing the molecular weight of each fuel to the weight of necessary air as provided in Table 2.4. 14.54 kg air is necessary to burn 1 kg pure diesel, while 12.47 kg air is required to burn 1 kg pure biodiesel. AFR values of the prepared fuel blends are utilized to find the lambda in the results section of the thesis.

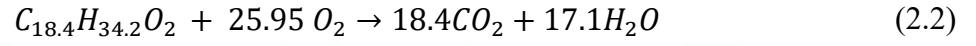
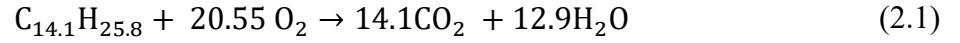


Table 2.4 : Physical properties of test fuels.

Property	unit	EN590 limit	B00	B05	B10	B20	B30	B50	B100	Analyze Method
Density (15 °C)	kg/m³	820-845	831.8	834.4	837.0	842.2	847.4	859.4	883.7	TS EN ISO 12185
Viscosity (40 °C)	mm²/s	2.00-4.50	2.88	2.96	3.03	3.17	3.21	3.61	4.32	TS 1451
Gross Calorific Value (GCV)	kg/m³	-	11074	10990	10922	10765	10613	10303	9537	ASTM D240
Canola Oil Methyl Ester	% (v/v)	<7	<0.1	5.0	10.2	20.3	30.2	49.9	NA	TS EN 14078
Cold Filter Plugging Point	°C	-15 (winter)	-14.2	-13.4	-12.9	-11.3	-12	-10.4	-3	TS EN 116
Carbon Content	% (m/m)	0.30	<0.1	<0.1	<0.1	<0.1	<0.1	<0.1	NA	TS 6148
Sulphur	mg/kg	<10	7.2	6.7	6.6	6.0	5.8	5.5	1.54	TS EN ISO 20846
Flash Point	°C	>55	58.5	59.5	61	62.5	66.5	69.5	184	TS EN ISO 2719
Cetane Number	-	>51	55.0	-	-	-	55.1	-	51.2	TS 10317
Lubricity	µm	<460	406	-	-	-	187	-	-	TS EN ISO 12156-1
AFR	-	-	14.54	14.43	14.33	14.12	13.92	13.50	12.47	-

3. SEQUENCE AND PROCEDURES OF THE EXPERIMENTS

Purpose of the thesis is to investigate the combustion of diesel engine under different conditions. In order to achieve this target, different set of experiments were performed in both chassis dynamometer and engine dynamometer. This section briefly describe the aim of the performed experiments and their sequence. Table 2.5 shows the experiments that were performed during the thesis.

Table 2.5 : Performed experiments in chassis and engine dynamometer.

Set of Experiment	Test Name	Fuel	Dynamometer
1	Full load	B00, B05, B10, B20, B30, B50	Chassis
2	NEDC	B00, B05, B10, B20, B30	Chassis
3	Cruise Speeds of NEDC	B00, B30	Chassis
4	WLTC	B00, B30	Chassis
5.1	Single Injection of 6.6 mg @2000 rpm	B00, B30	Chassis and Engine
5.2	Double Injection of 6.6 mg (1.1+5.5) @2000 rpm	B00	Engine
5.3	Triple Injection of 6.6 mg (0.9+1.1+5.5) @2000 rpm	B00	Engine
5.4	Single Injection of 12 mg @2000 rpm	B00	Engine

First set of experiment were performed in the chassis dynamometer at full open acceleration pedal position of the vehicle at third gear. The purpose was to investigate the effects of biodiesel on emission, power and combustion parameters and to verify the experimental setup via comparing the measured data with the declared specifications of the vehicle. Full load operation points of the diesel engine were examined with various blends of biodiesel as given in Table 6.1, while measuring the in-cylinder pressure, emissions and performance of the engine.

Second and third set of experiment were performed with the purpose that was similar to full load experiments, however in these experiments it was aimed to understand the changes in the combustion according to biodiesel utilization at transient regulation cycles and cruise speeds of the NEDC. In order to realize this objective, the EURO-5 compliant diesel passenger vehicle was tested under NEDC in the second set of experiment. In the third set of experiment, part load conditions of the NEDC, which are the cruise speeds of the Urban Driving Cycle (UDC), were tested with biodiesel blended diesel. In this way, the relationship between the change in emissions and combustion parameters were evaluated.

Fourth set of experiments were also performed on chassis dynamometer to compare the effects of biodiesel-blended diesel on the obsolete (NEDC) and the new (WLTC) regulation cycles. Thus, the vehicle was tested under the NEDC and then WLTC. The emission and performance results of the regulation cycles were compared to assess the differences in the effects of biodiesel.

Fifth set of experiments were performed with different fuel injection configurations under heavy EGR to propose a new injection configuration that can reduce the NO_x and soot emission without increasing fuel consumption. Each injection configuration was grouped separately to present the results of the experiments clearly. Single injection of the 6.6mg fuel was investigated with B00 and B30 fuel. After, double and triple injection configurations of the 6.6 mg fuel at 2000 rpm were tested via modifying injection configuration and EGR rate. Later, single injection configuration of the 12 mg fuel was tested to investigate the applicability of heavy EGR under high load condition.

3.1 Test Procedure of the Full Load Experiments on the Chassis Dynamometer

Full Load (FL) experiments were carried out in chassis dynamometer by using constant speed operation mode of Vecon computer. Constant speed operation mode allows to keep the vehicle's velocity constant independent from the acceleration pedal position (APP) by the assist of coupled AC electric motors at the rollers. Roller speed was set to velocities of 102.8, 90.0, 77.1, 64.3 and 51.4 km/h at third gear of the vehicle, which respectively correspond to the engine speed of 4000, 3500, 3000, 2500 and 2000 rpm. APP of the vehicle was set to fully open position to get the highest power of each engine speed. OBD data was observed during all experiments to have the same vehicle condition.

In order to get the same vehicle and engine condition before each experiment several controls were performed. First, vehicle was run until all fuel in the tank burned by engine. Once engine of the vehicle cannot be cranked via battery, prepared 3 liters of first test fuel (B00) was filled to the tank. Second, 12 Volt battery of the vehicle was controlled and if it was not fully charged, it was charged at least for 6 hours. Third, pressure of the tires were set to recommended pressure of 30PSI. After controls had completed, full load experiment was carried out from both low speed to high speed (2000 rpm to 4000 rpm) and then high speed to low speed (4000 rpm to 2000 rpm) by running vehicle at least 90 seconds for each specified engine speed. At the end of each fuel blend's FL experiment, remaining fuel in the tank was finished by running vehicle. Same procedure was applied respectively to remaining fuel blends B05, B10, B20, B30, and B50.

In-cylinder pressure measurements were performed for 100 cycles and the combustion parameters were calculated as average of these 100 cycles. Figure 2.7 shows the graphical interface of the AVL Concerto program that enables to calculate the combustion parameters. Rate of heat release (ROHR), cumulative heat release, SOC, BD, mass burned fuel (MBF) 5%, 10%, 50%, 90% can be calculated via the pre-defined block "Thermodynamics2" as can be seen in Figure 2.7. Besides, maximum in-cylinder pressure can be calculated via "PeakValue" block.

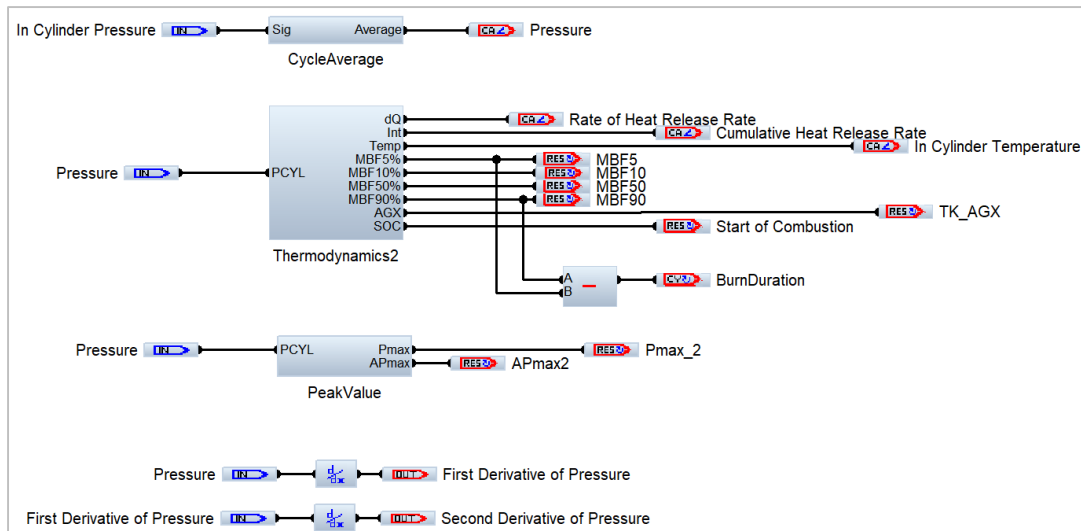


Figure 2.7 : Graphical interface of AVL concerto calculation program.

In this way, effects of different blends of biodiesel on power, emissions and combustion parameters were investigated under full load condition of the engine. Most of experimental studies in the literature focused on the chemical and physical properties of the alternative fuels to point out the effects of biodiesel on emission and performance. However, a few of them clarify the variations in emissions according to the combustion parameters. Thus, change in combustion parameters were analyzed via evaluating the in-cylinder pressure measurement and correlated with the change in the emissions.

3.2 Test Procedure of the NEDC, WLTC and cruise points of the NEDC

The NEDC and WLTC tests were conducted with diesel and biodiesel-diesel blend to investigate emissions of each cycles, to explore the biodiesel impact to new coming type approval (TA) procedure and to compare the effects biodiesel on each cycles.

Experiments started with the NEDC test. First, to drain fuel tank, vehicle was driven in the chassis dynamometer until the all fuel was depleted. Afterwards, the vehicle was re-fuelled with three liters of fuel and then parked to the soaking area. The vehicle stayed at least 8 hours in the soaking area to condition the oil and water temperature of the vehicle to 24 °C before the NEDC testing as defined in the regulation. During soaking, 12-volt battery of the vehicle was charged and tire pressure was set to 30 psi as they have impact on the results. After soaking, vehicle was placed on the chassis dynamometer without cranking the engine as described in UNECE R83. All NEDC experiments were conducted with the same driver and under the same conditions to

minimize the external effects on results. Three NEDC experiments were performed with each prepared fuel; B00, B05, B10, B20 and then B30.

In addition to the NEDC tests, experiments were carried out to understand the effects of biodiesel on the combustion characteristics at constant travel speeds of 15, 32, 35, and 50 km/h in the first phase of NEDC. In addition to emission and power measurements, in-cylinder pressure measurement values were also examined at constant speed operating points where the engine speed was constant. Tests were made with B00 and B30 fuels. Test points are shown in Table 2.6. B00 fuel was first tested for each point. In order to achieve the same test conditions, the vehicle was run at 50 km/h for 15 minutes in second gear. In this way, temperature of the engine coolant and exhaust gases reached to 90 °C and 200 °C, respectively, before the measurements were taken. After the engine started to run stable under the operation point, 3 different measurements were taken for 5 minutes. Experiments were repeated in the same way with B30 fuel for all cruise speeds that are listed in Table 2.6.

Table 2.6 : Cruise speed, gear position, engine speed and acceleration pedal position.

Vehicle Velocity	Gear	Engine Speed	Acceleration Pedal Position (APP)
km/h	-	rpm	%
15	1	1800	20
32	2	2000	17
35	3	1380	13
50	3	1950	18

After completing the NEDC and cruise speed experiments, the WLTC tests were performed with B00 and B30 in order to examine the effects of biodiesel on regulated emission of new TA procedure. Similar to the NEDC procedure, first B00 and then B30 were added to the drained fuel tank. Three cold WLTC experiments were also done for each fuel after 8 hours of soaking.

The conditions of both test vehicle and laboratory have impacts on the results of regulation cycles. Brace et al. studied the effect of testing factors and found out that the battery state of charge, engine oil level, pedal busyness, speed error, road fan speed, vehicle alignment, tire type, tire pressure, and simulated vehicle mass have significant impacts on the results [84]. As the same test vehicle was utilized for all experiments, tire type and vehicle mass were same for all cases. Influence of engine oil level, tire pressure, and battery state of charge on results were eliminated by controlling and

recording values before each experiment by the test operator. Vehicle alignment, speed error and road fan speed were controlled via sensors on the chassis dynamometer control system. In order to minimize the effect of pedal busyness same driver performed all experiments. In addition to these parameters, engine sensors were monitored via OBD during each experiment to have the same conditions. Figure 2.8 shows the engine speed, APP, battery voltage and engine coolant temperature of the vehicle during the NEDC experiments performed with B00, B05, B10, B20 and B30. There were minor differences at engine speed and driver could drive the vehicle without any violation in all cases, as stated in TA procedure.

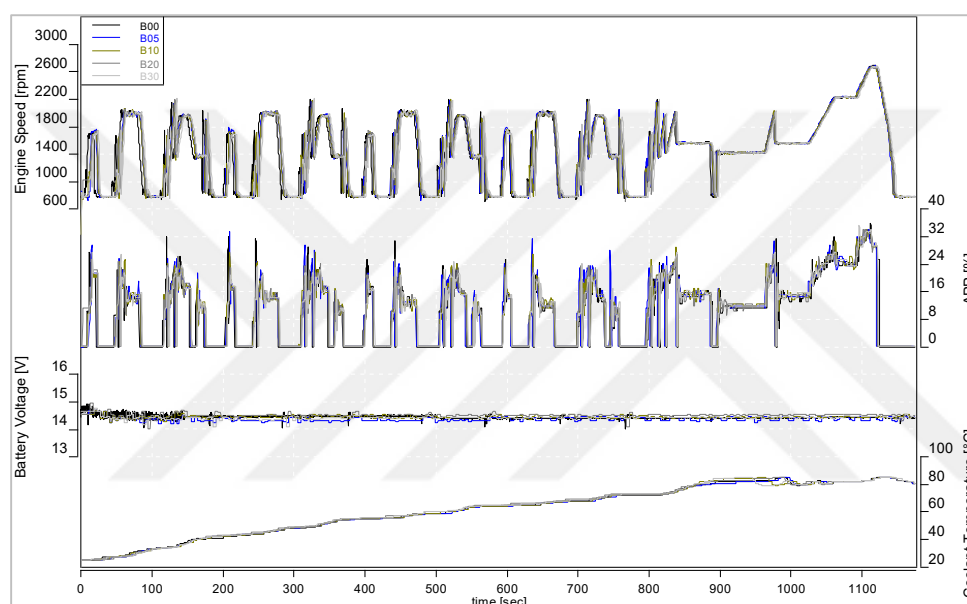


Figure 2.8 : Engine speed, acceleration pedal position, battery voltage and coolant temperature of NEDC experiments.

3.3 Experiments With Constant Fuel Quantity While Modifying Injection Parameters

Low and medium load conditions of the diesel engine were investigated in this section by modifying injection parameters and EGR flow rate at 2000 rpm. The purpose of the experiments was to investigate the change in combustion as well as performance and emissions. In order to eliminate effect of fuel quantity, the same amount of fuel was injected during all experiments instead of keeping the engine power by adjusting fuel quantity.

Table 2.7 shows modified parameters during the experiments that were performed under the steady state operation. Definition of the injections can be seen in Figure 2.9.

As the vehicle has split injection configuration under the regulation cycles, experiments were performed at triple, double and single injection configurations.

Table 2.7 : Modified engine parameters.

Description	Abbreviation	Unit	Modification Range
Rate of Exhaust Gas Recirculation	EGR	%	0 to 60
Quantity of Main Injection	Main-Q	mg/stroke	1.5 to 6.6
Start of Injection of Main Injection	Main SOI	Crank Angle	-10 to 10
Quantity of Pilot1 Injection	Pilot1-Q	mg/stroke	1.1 to 5.1
Start of Injection of Pilot1 Injection	Pilot1 SOI	Crank Angle	-50 to -10
Quantity of Pilot2 Injection	Pilot2-Q	mg/stroke	0.9
Start of Injection of Pilot2 Injection	Pilot2 SOI	Crank Angle	-90 to -30

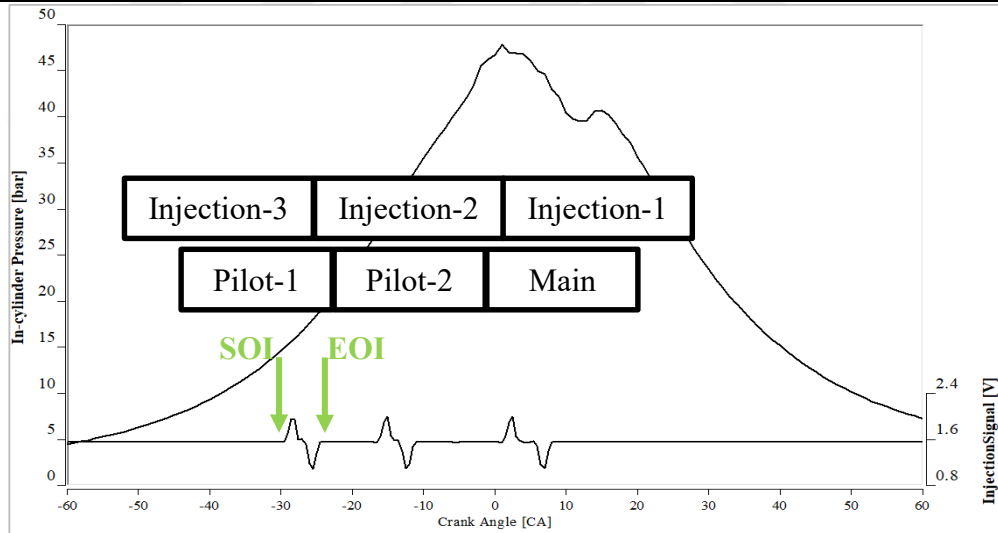


Figure 2.9 : Definitions of the fuel injections.

Solenoid type injectors are available on the engine and valve of the injector stays open during the injection process via applied current from ECU. However, AVL Indicom system gives the data in the format that was shown in Figure 2.9. Beginning of the first triangle is the start of injection (SOI) and end of the second reversed triangle is the end of injection (EOI). Current is applied to the injector between SOI and EOI to inject fuel into the combustion chamber.

EGR rate and fuel injection parameters, which is presented in Table 2.7, were modified in sequence during the experiments. There are many options as there are seven different variable that can be adjusted in a wide range. As the purpose of the study is not to investigate all possibilities, parameters were adjusted to achieve alternative advanced combustion methods. Value of the first parameter was modified while keeping the other parameters constant. Parameter was modified until the trend at power or NO_x was changed. Optimum value of the first modified parameter was selected from the adjustment range of the parameter. Then the value of the first parameter was kept at the selected optimum value during the adjustment experiments of the other parameters.

Measurements of the each engine condition was performed after stable operation of the engine. A duration of 30 seconds was waited for stabilization of the emissions and power, after adjusting the each engine parameter. Once engine was operated under stable condition, a measurement record of 60 seconds was performed under stable steady state condition. Two set of experiments were performed for each engine operation point.

4. RESULTS OF THE EXPERIMENTS

Five set of experiments were performed through the thesis study to investigate the regulation cycles, the effects of biodiesel, and the advanced combustion methods with different injection configurations.

In the fourth section of the thesis, results of the experiments are presented in five subsections by discussing the effects of the modified parameters on the results.

4.1 Results of the Full Load Experiments with Diesel and Biodiesel Blends

EU, Turkey, and other countries encourage the use of biodiesel as it is produced from organic compounds and has lower environmental influence compared to petroleum diesel. In order to evaluate the effects of these organic compounds and the potential of biodiesel for advanced combustion methods, full load experiments were performed with B00, B05, B10, B20, B30 and B50. Combustion characteristics, power, fuel consumption, CO₂, CO, HC and exhaust gas temperature were investigated to realize the effects of biodiesel blended diesel on these parameters.

4.1.1 Combustion characteristics

Vast majority of the experiments about utilization of biodiesel for diesel engines show similar trends on exhaust emission and performance. Root cause of the variations at emissions and performance is generally clarified by analyzing combustion parameters such as; heat release rate (HRR), start of combustion (SOC), ignition delay duration (IDD) and burn duration (BD). Moreover, understanding the effects of the biodiesel on IDD, SOC and BD will offer an opportunity to further improve the performance and emissions of diesel engines. Therefore measured in-cylinder pressure data were evaluated to discover the effects of biodiesel on HRR, IDD, SOC and BD.

Figure 4.1 shows in-cylinder pressure and injection signal data of the FL experiment at 3500 rpm for B00, B05, B10, B20, B30, and B50. The darkest blue line shows data of the B50 and lines are getting lighter blue with decreasing biodiesel blend ratio where B00 is shown with the lightest blue line.

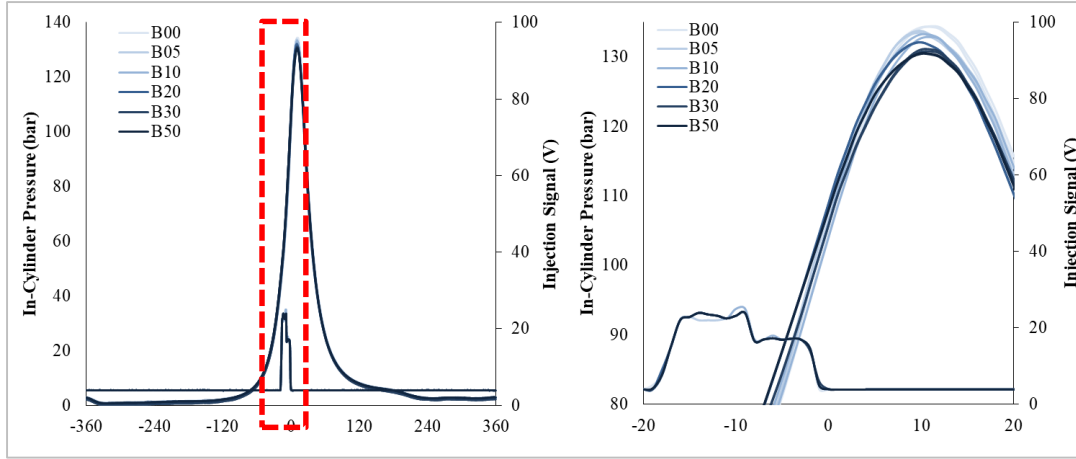


Figure 4.1 : In-Cylinder Pressure and Injection Signal of fuel blends at 3500 rpm.

Start of Injection (SOI) signal began at Crank Angle (CA) -18, which is 18 CA before Top Dead Center (bTDC), and injection signal continues up to TDC as can be seen in Figure 4.1. Although fuel injection signal was identical for all fuels, in-cylinder pressure was affected from added biodiesel blend. Increased biodiesel blend ratio caused earlier rise of in-cylinder pressure as darker blue lines are higher than the lighter blue lines up to TDC. Biodiesel's excess oxygen content potentially caused higher in-cylinder pressure rise compared to pure diesel by enabling burn of more fuel just after SOC. B50 reached peak pressure earlier than B00 while B00 had higher peak in-cylinder pressure compared to B50 as can be seen in zoomed section of Figure 4.1.

Rate of heat release (ROHR) was calculated by using equation (4.1) [11]; where i is crank angle measurement interval, κ is polytrophic coefficient found by dividing specific heat at constant pressure (c_p) to specific heat at constant pressure volume (c_v), P is the in-cylinder pressure and V is the swept volume of the cylinder.

$$Q_i = \frac{1}{\kappa - 1} \times (\kappa \times P_i \times (V_{i+1} - V_{i-1}) + V_i \times (P_{i+1} - P_{i-1})) \quad (4.1)$$

In addition to ROHR, calculation method of the combustion parameters SOC, IDD, and BD was shown in Figure 4.2.

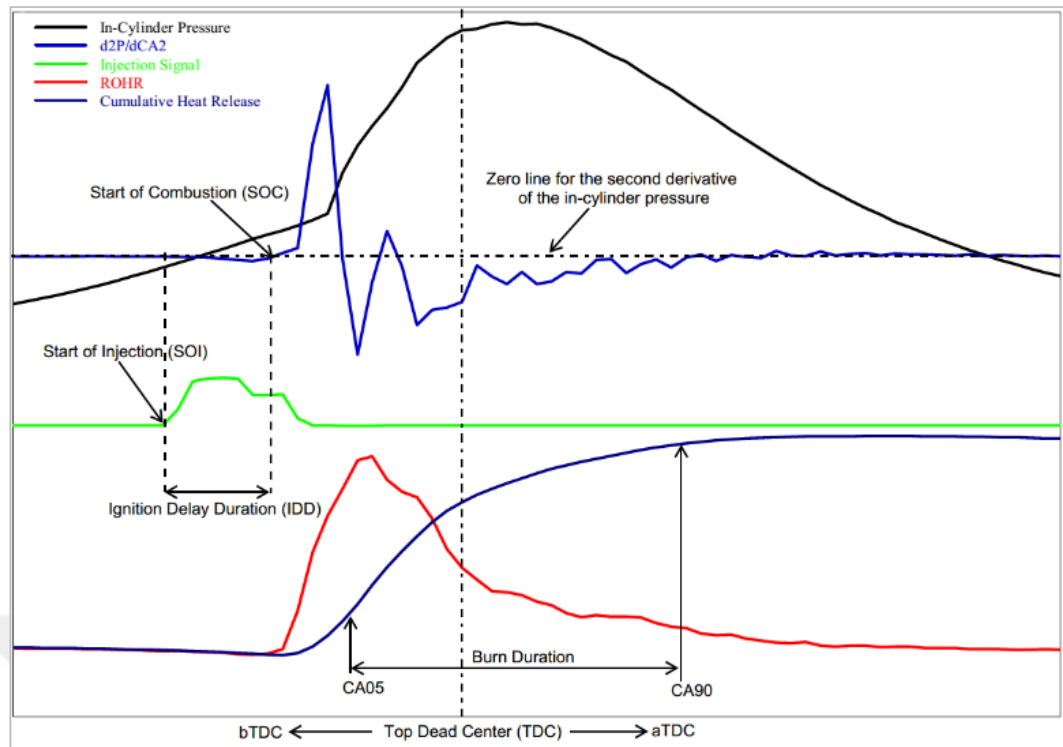


Figure 4.2 : Definition of Combustion Parameters.

There are different methods to determine the SOC. One of the method determines SOC as the crank angle where the second derivative of the in-cylinder pressure intersects the zero line, which is shown in Figure 4.2 [85–87]. The other method determines SOC as the crank angle of the engine where the HRR becomes positive [55,88]. The second method was utilized to determine the SOC in the thesis study. Figure 4.3 shows the determined cranks angle of the SOC for all fuels at 3500 rpm where HRR become positive.

Beside, Figure 4.3 shows the ROHR for all fuel blends at 3500 rpm operation point. Similar to in-cylinder pressure data, rise of heat release rate of biodiesel added fuel started earlier compared to pure diesel.

Besides, the earliest SOC was observed for B50, whereas B00 had the latest SOC. There is approximately 2.5 CA difference between the SOC of B00 and B50 at 3500 rpm as can be seen in zoomed section of Figure 4.3, which is close to the reviewed literature [85–87]. Cetane number and oxygen content of the biodiesel blended fuel enable the mixture to ignite easier as the fuel droplets that reached to ignition condition can meet with oxygen more simply compared to the mixture of pure diesel and air.

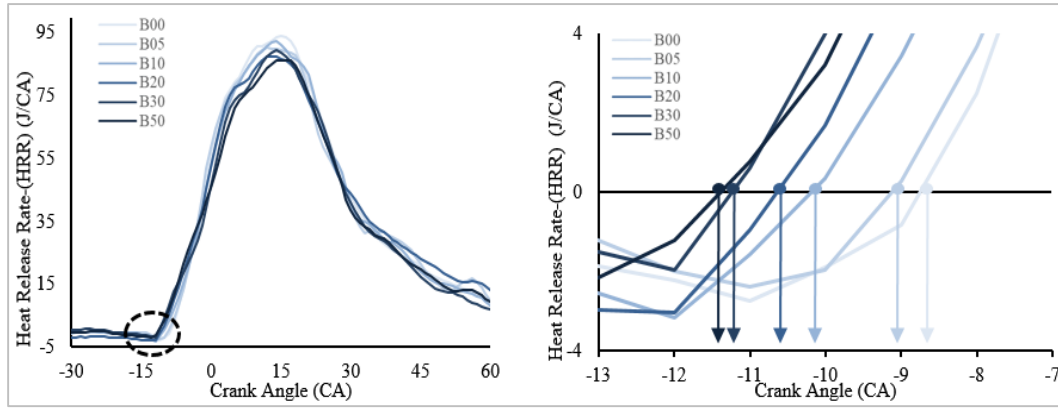


Figure 4.3 : Heat Release Rate of all fuel blends at 3500 rpm.

After computing SOC, ignition delay duration (IDD) was found by calculating the CA difference between the SOC and the SOI as shown in Figure 4.2. IDD was longer for pure diesel than biodiesel added blends for all FL operation conditions as shown in Figure 4.4. Higher oxygen content of the biodiesel led shorter ignition delay duration (IDD) compared to pure diesel. Lower CO and HC emissions may be correlated with earlier SOC and shorter IDD as these combustion parameters revealed that auto ignition quality was improved and more complete combustion was achieved.

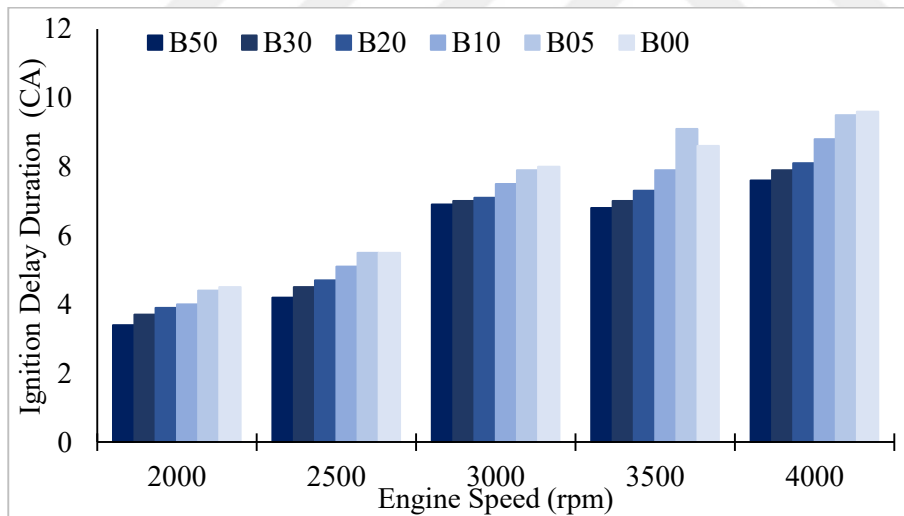


Figure 4.4 : Ignition Delay Duration of all fuel blends.

In addition to SOC and IDD calculation, burn duration (BD) was also calculated by HRR. BD is defined as CA difference between 90% and 5% of the cumulative heat release as shown in Figure 4.2 [89]. Increase of the biodiesel blend ratio reduced the BD compared to pure diesel as shown in Figure 4.5. Effect of biodiesel on BD was more obvious when the volume of the blended biodiesel is more than 10%.

Analyzed in-cylinder pressure data indicated that biodiesel addition shorten the IDD and BD up to 3 CA, which result in higher rise of in-cylinder pressure and temperature compared to pure diesel. Rapid temperature rise under higher oxygen condition is possibly further increase the formation of NO_x and is another root cause of higher NO_x emission of biodiesel blends compared to pure diesel.

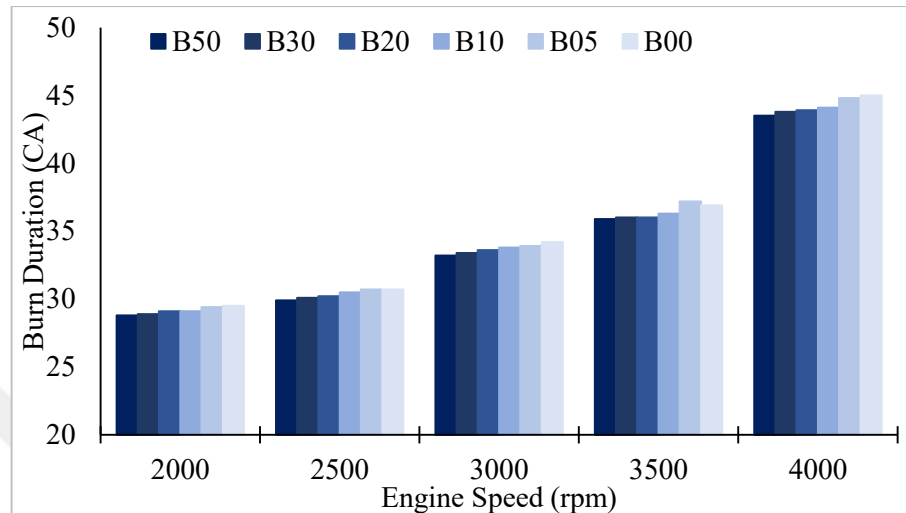


Figure 4.5 : Burn Duration of all fuel blends.

4.1.2 Power, fuel consumption and CO_2 emission

Power was reduced with increasing biodiesel blend ratio for all of the FL operation points as shown in Figure 4.6. As the injected amount of fuel was same for each engine speed, fuel consumption was increased with addition of biodiesel. Decrease of power or in other terms increase of fuel consumption is related to lower GCV of the biodiesel compared to the diesel.

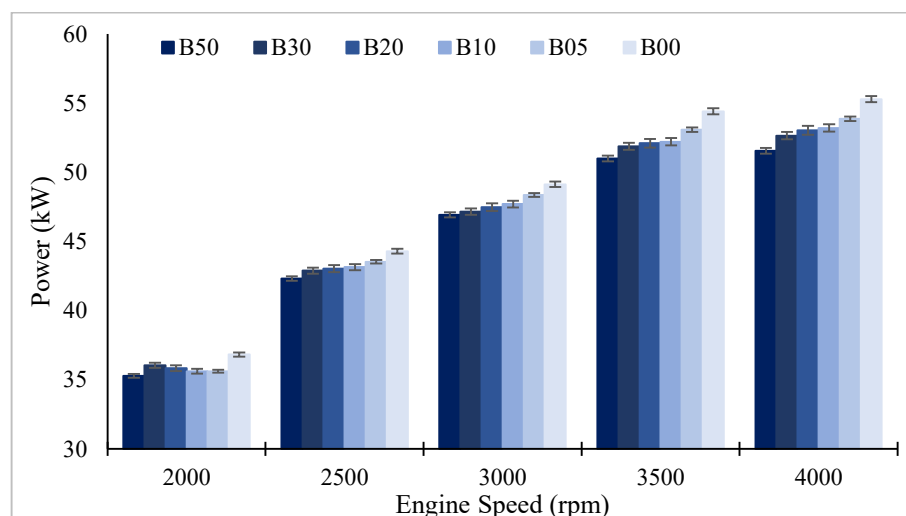


Figure 4.6 : Wheel power over the FL experiments of all fuel blends.

However, CO₂ emission did not have same trend with wheel power as shown in Figure 4.8. Experiments revealed that the utilization of biodiesel did not give a coherent global trend for CO₂ emission whereas wheel power decreased continuously. However, B50 had the highest CO₂ emission at all engine speeds. When B50 was excluded, a slight reduction trend of CO₂ emission was observed at 2000 and 2500 rpm with increasing biodiesel ratio. There was not consistent trend for CO₂ emission at the engine speeds 3000, 3500, and 4000 rpm and CO₂ emission is almost at the same level for each fuel.

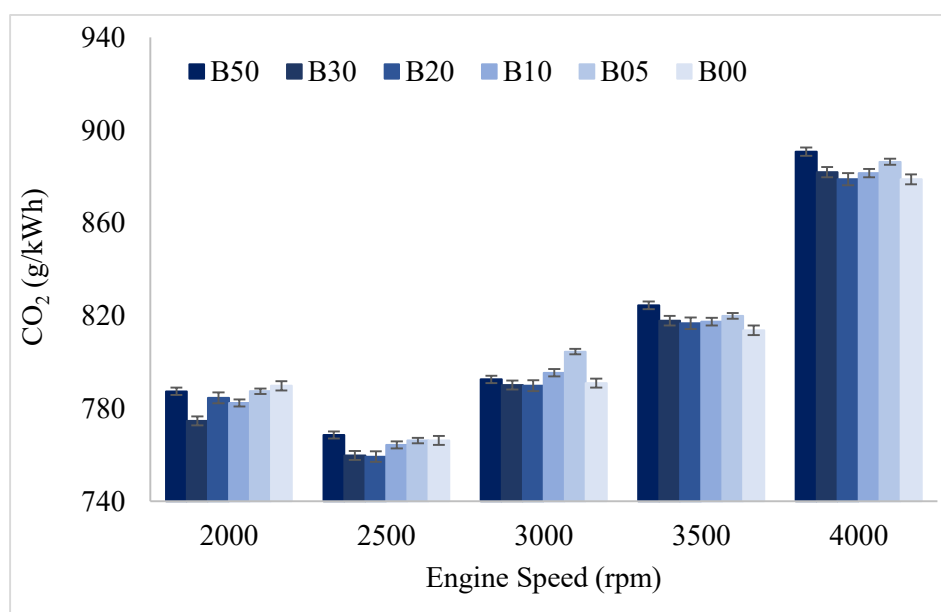


Figure 4.7 : CO₂ emission over the FL experiments of all fuel blends.

4.1.3 NO_x emission

NO_x emission increased with increasing biodiesel blend ratio consistently for all engine speeds as can be seen in Figure 4.8. Higher oxygen content enabled higher in-cylinder pressure rise as mentioned in the combustion characteristics section. As in-cylinder temperature has same trend with in-cylinder pressure, rapid rise of in-cylinder temperature of biodiesel blended fuel compared to diesel may be the reason of higher NO_x emission with increasing biodiesel blend ratio.

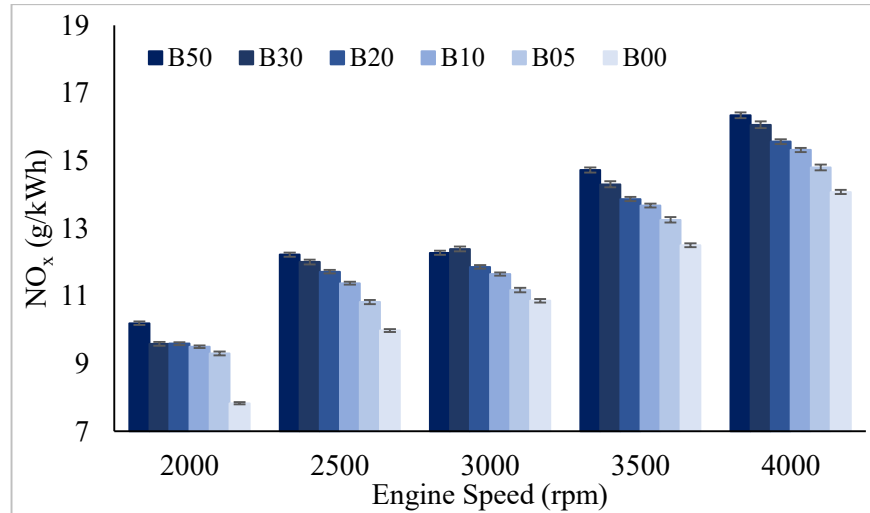


Figure 4.8 : NO_x emission over the FL experiments of all fuel blends.

4.1.4 CO and HC emission

Engines emit CO and HC emissions due to incomplete combustion. Although diesel engines work under lean operation conditions, available oxygen in the combustion chamber cannot burn all carbon and hydrogen atoms in the fuel chain [11]. Addition of biodiesel blend increases the lambda due to its higher oxygen content compared to diesel as shown in Table 4.1. Lambda of each fuel was calculated by equation (4.2). AFR of each fuel was utilized from Table 2.4 (14.54 for B00, 13.50 for B50), which was calculated by equations (2.1) and (2.2). Therefore CO emission reduction under all engine speeds, which is given in Figure 4.9, can be associated to the oxygen content of the biodiesel.

$$\lambda = \frac{\text{MAF}/F_{\text{mass}}}{\text{AFR}} \quad (4.2)$$

Table 4.1 : Injected fuel mass and lambda change according to biodiesel addition.

Engine Speed	Fuel Blend	MAF	Fvol	Fmass	Lambda (λ)
rpm	Bxx	kg/h	l/h	kg/h	-
2000	B00, B05, B10, B20, B30, B50	162.0	11.3	9.40, 9.43, 9.46, 9.52, 9.58, 9.69	1.18, 1.19, 1.19, 1.20, 1.21, 1.23
2500	B00, B05, B10, B20, B30, B50	203.8	13.4	11.15, 11.18, 11.22, 11.29, 11.35, 11.49	1.25, 1.26, 1.26, 1.27, 1.28, 1.30
3000	B00, B05, B10, B20, B30, B50	238.7	15.4	12.81, 12.85, 12.89, 12.97, 13.05, 13.21	1.28, 1.28, 1.29, 1.30, 1.31, 1.33
3500	B00, B05, B10, B20, B30, B50	274.3	17.2	14.31, 14.35, 14.40, 14.49, 14.57, 14.75	1.32, 1.32, 1.32, 1.33, 1.34, 1.36
4000	B00, B05, B10, B20, B30, B50	291.2	18.6	15.47, 15.52, 15.57, 15.66, 15.76, 15.95	1.29, 1.30, 1.30, 1.31, 1.32, 1.34

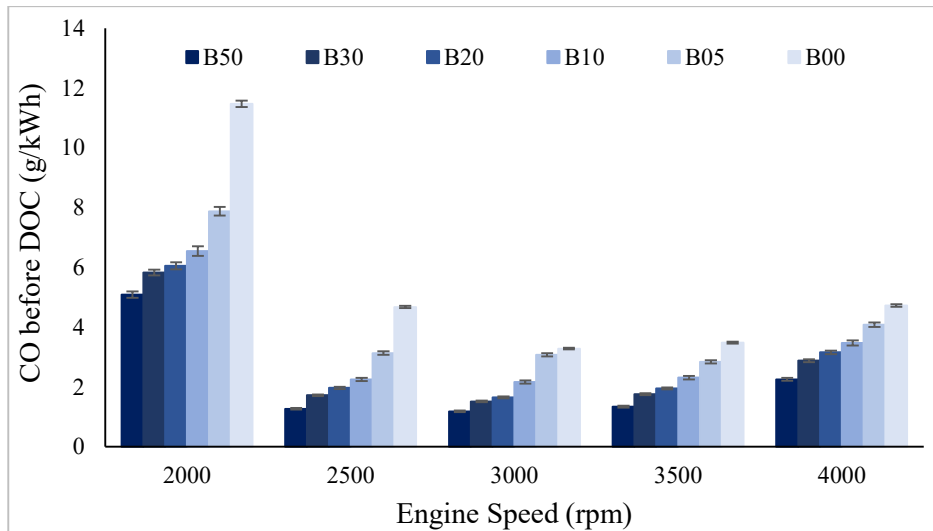


Figure 4.9 : CO emission before DOC over the FL experiments of all fuel blends.

Regulation cycles mandated by authorities for light duty passenger vehicles do not cover the all power operation region of the vehicle. Most of the OEMs do not open EGR valve under the FL operation conditions, as there is no obligation to reduce emissions for this load region. Therefore engines emit lower HC emission under FL condition compared to low load operation region where EGR is active. Although there was very low HC emission in the FL test as can be seen in Figure 4.10, biodiesel addition decreased the HC emission compared to pure diesel. In addition to oxygen content, shorter ignition delay duration of biodiesel blended diesel enabled cleaner combustion and reduced the CO and HC emissions.

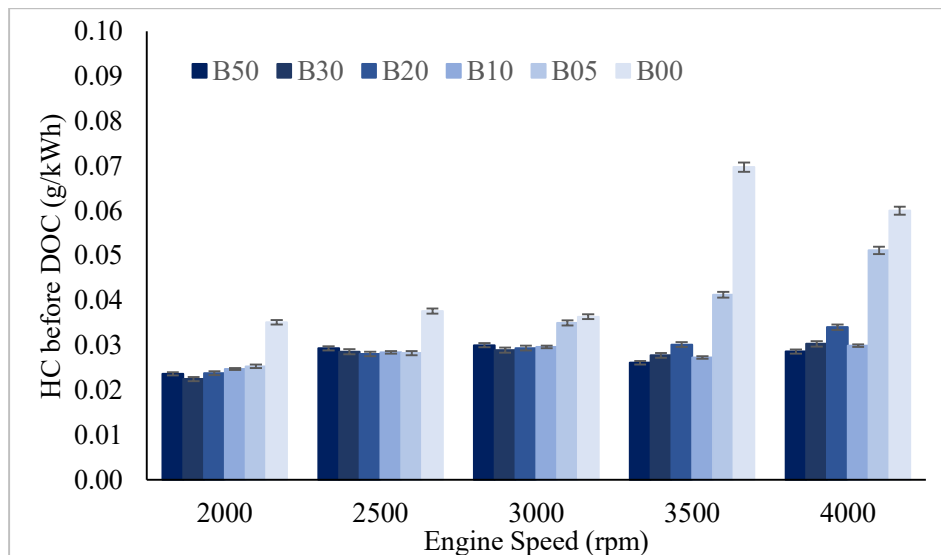


Figure 4.10 : HC emission before DOC over the FL experiments of all fuel blends.

4.1.5 Exhaust temperature

Exhaust temperature is an indirect indication of combustion efficiency as unburned fuel caused temperature rise in the exhaust gas [11]. Figure 4.11 shows the exhaust gas temperature (EGT-1) sensor data that is located in front of the DOC. Exhaust temperature decreased with increasing biodiesel blend ratio, which was likely due to cleaner combustion and shorter combustion duration. Because, injected biodiesel blended fuel was burned earlier than pure diesel and more time was available for biodiesel-blended fuel in the expansion cycle for cooling. Besides, increasing biodiesel volume result in more lean mixture in the chamber and adiabatic flame temperature of the diesel fuel is higher than biofuels [90].

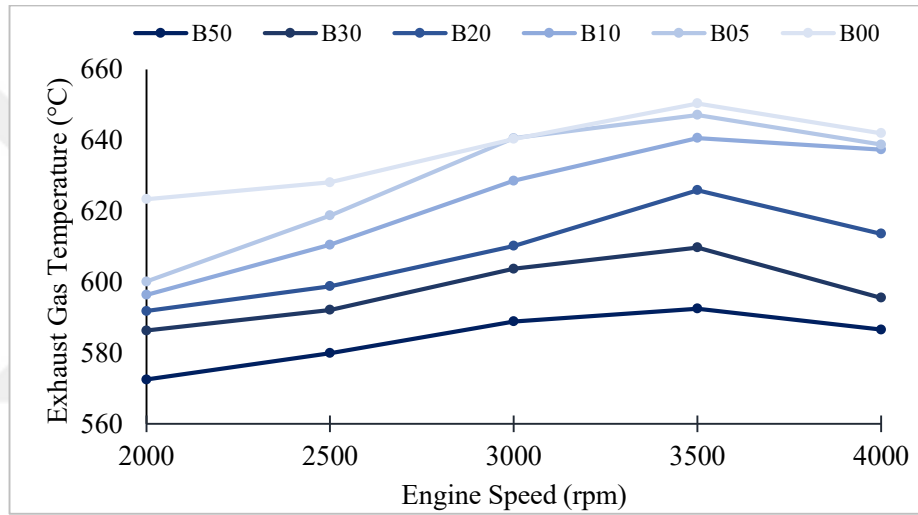


Figure 4.11 : Exhaust gas temperature before DOC over the FL experiments of all fuel blends.

4.1.6 Evaluation of the results for advanced combustion method

Utilization of biodiesel lowers environmental impact of exhaust emissions and fossil fuel dependency; however, it is not possible to meet EURO-6 emission with EURO-5 equipped vehicle by only fuelling with biodiesel due to stringent NO_x emission limit. Biodiesel were widely utilized under advanced combustion methods (LTC, HCCI and PCI) to accomplish very low NO_x and particulate emissions. Although, performed experiments revealed shorter IDD and BD which may worsen the homogenous charge preparation, Fattah et al. [91] stated that shorter IDD of biodiesel decreased CO and HC emission by reducing fuel-rich regions. Moreover, Petersen and Northrop stated that shorter premixed combustion duration and higher ignition quality of the biodiesel caused shorter BD compared to conventional diesel, which enabled to achieve lower

emissions with biodiesel under advanced combustion than pure diesel [92,93]. Therefore, optimizing fuel injection parameters according to effects of biodiesel on combustion characteristics may provide the opportunity to achieve further improvements under both conventional and advanced combustion operation conditions.

4.2 Results of the NEDC Testing with Diesel and Biodiesel Blends

4.2.1 CO₂ emission and fuel consumption

A considerable amount of research has been conducted regarding biodiesel's impact on CO₂ emission, as it has favorable influence on the environment. There are a wide range of experimental results in the literature showing that addition of biodiesel to diesel both increase and decrease the CO₂ emission [68,70,71]. According to some researchers it is proposed that increase of the CO₂ is related to 11% oxygen content of biodiesel in the molecular formula [68,94] and more complete combustion [70]. Whereas many other researchers claimed that the lower carbon to hydrogen ratio of biodiesel causes less CO₂ emission than pure diesel [71,95]. However, the biodiesel utilized in the experiments of this study has higher carbon to hydrogen ratio than diesel.

Similar to reviewed literature [68,70,71], CO₂ emission variation of the urban driving cycle (UDC), and extra urban driving cycle (EUDC) experiments had different trends according to biodiesel addition. Biodiesel added fuel emitted more CO₂ emission than diesel fuel under the NEDC as shown Figure 4.12.

Since identical engine parameters were utilized during biodiesel and diesel tests, it is found that combustion of biodiesel yielded more CO₂ emission increase under UDC than EUDC. Engine speed is more changing under UDC compared to EUDC, so UDC is more dynamic. This may lead more rise in CO₂ emission with use of biodiesel at UDC compared to EUDC. Besides; higher density of biodiesel led higher amount of fuel injection, higher bulk modulus of biodiesel led to injection of biodiesel earlier, and higher viscosity of biodiesel caused smaller and more distributed fuel droplets, which improve spray characteristics, fuel evaporation, and flame propagation of biodiesel [68]. These parameters enabled biodiesel blended fuel to prepare air-fuel mixture that can ignite earlier and more complete compared to diesel. The higher CO₂

emission during the UDC can be explained with different mixing characteristics of biodiesel and diesel. However, increasing cylinder temperature fasten the chemical reaction and mixing effect of biodiesel reduces at hot engine. This may be the reason of the reduced CO₂ emission increase of biodiesel under EUDC.

Results of the experiments showed that biodiesel added blends have higher fuel consumption than pure diesel under the NEDC due to biodiesel's lower GCV than petroleum diesel. Besides having lower GCV, due to combined effect of other parameters (AFR, combustion temperature, injected fuel pressure, SOI, piston bowl geometry, and biodiesel blend ratio) on combustion, CO₂ emission varied both according to load and engine conditions. However, even though there is an increase at measured CO₂ of exhaust gas with biodiesel usage, it should cause less environmental concern as the study at Argonne National Laboratory on B100 reveals that life-cycle greenhouse gas emission of biodiesel is %74 lower than petroleum diesel [96].

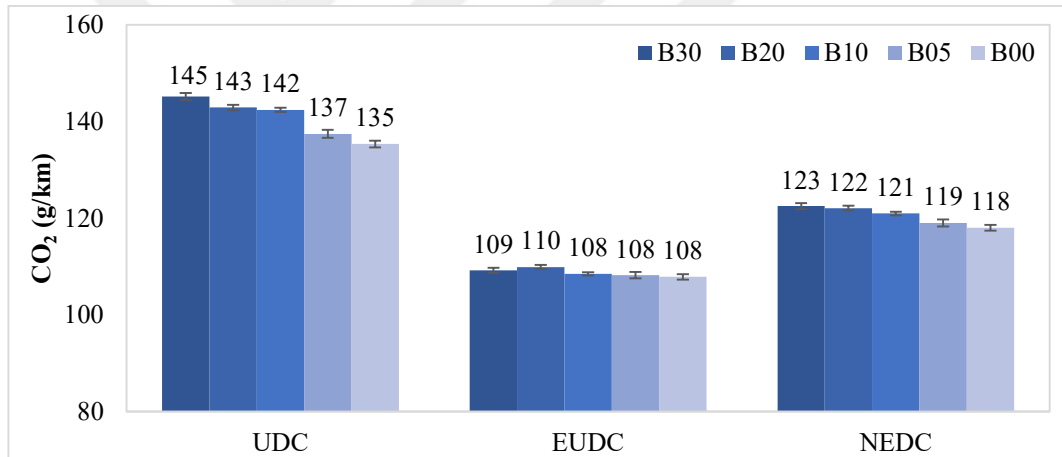


Figure 4.12 : CO₂ emissions of fuel blends under the UDC, EUDC and NEDC.

4.2.2 NO_x emission

NO_x emission, emitted from the engine, is related with thermal, prompt and fuel-bounded NO_x formation sources [97]. Among these three sources, thermal NO_x is the most relevant to combustion and explained by the Zeldovich Mechanism. Equation (4.3) that is derived by Zeldovich Mechanism, shows that NO_x formation is highly depend on combustion temperature and oxygen concentration where $d[NO]/dt$ is NO formation, T is the in-cylinder temperature, $[O_2]$ and $[N_2]$ are their equilibrium concentrations [11]. It is noted by many researchers that biodiesel has higher density and oxygen content compared to diesel that cause higher in-cylinder temperature, which increased formation of NO_x [68,70].

$$\frac{d[NO]}{dt} = \frac{6 \times 10^{16}}{T^{1/2}} \times \exp\left(\frac{-69,090}{T}\right) \times [O_2]^{1/2} \times [N_2] \quad (4.3)$$

Compatible with theoretical explanation, NO_x emission increased under the NEDC experiments with increasing biodiesel ratio as can be seen in Figure 4.13. The basic parameter affecting NO_x emission in biodiesel usage was oxygen concentration in fuel as it increased combustion (flame) temperature, which in turn, increased formation of NO_x. Besides, measured NO_x emission of the same fuel was higher under the UDC than the EUDC, due to longer period of high load conditions per km under UDC compared to EUDC. Therefore, it can be concluded that higher NO_x was observed as the engine worked under higher power and transient condition.

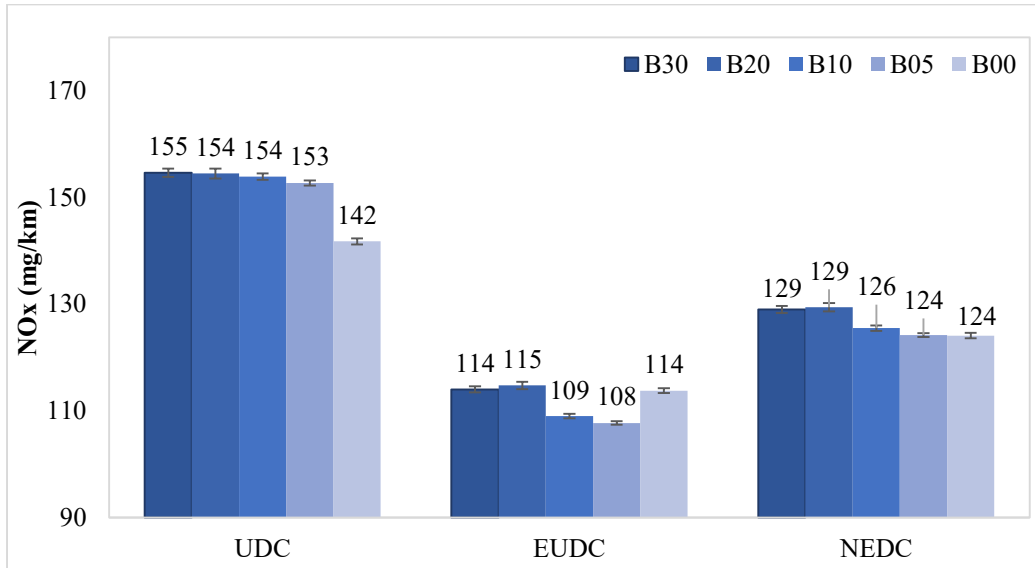


Figure 4.13 : NO_x emissions of fuel blends under the UDC, EUDC and NEDC.

4.2.3 CO emission

Since CO emissions are proportional to the air-fuel ratio, lower CO emissions are observed in combustion that occurred under lean mixture because they can react easily with oxygen atoms to convert carbon atoms in the combustion chamber to CO₂ [11,68]. As seen in Figure 4.14, the increase in biodiesel ratio decreased the CO emission. Oxygen in the chemical chain of biodiesel ensures that the fuel-air mixture is leaner than that of the diesel and thus reduces CO emissions. The CO emission decreased with increasing biodiesel ratio from 405 mg/km to 241 mg/km, however the reducing effect of biodiesel reduced above 10% biodiesel blend as shown in Figure 4.14.

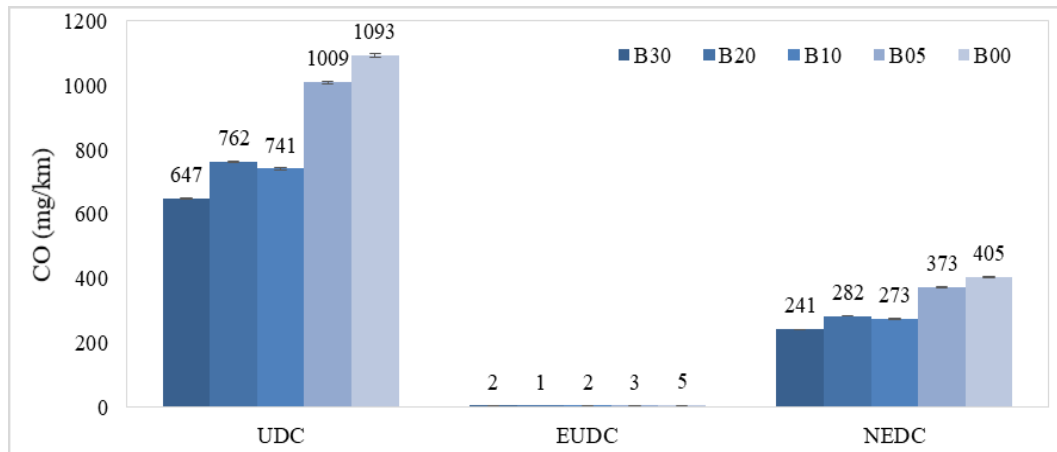


Figure 4.14 : CO emissions of fuel blends under the UDC, EUDC and NEDC.

4.2.4 HC emission

HC emissions are resulted due to incomplete combustion related the reactions of hydrogen and carbon atoms in the fuel chain with oxygen and therefore it can also be used as a measure of combustion efficiency [11]. Same amount of biodiesel blended diesel (in grams) can be injected in shorter duration than pure diesel as the bulk modulus and density of biodiesel are higher than diesel, which caused lower ignition delay duration for biodiesel compared to diesel. This resulted lower HC emission and more complete combustion for the NEDC test. As can be seen in Figure 4.15, HC emissions decreased with increasing amount of biodiesel. The HC reduction was not proportional with the added biodiesel ratio. The effect of biodiesel on reducing HC emission decreased after 10%, therefore it was evaluated that using more than 10% biodiesel is not effective in reducing the HC emission.

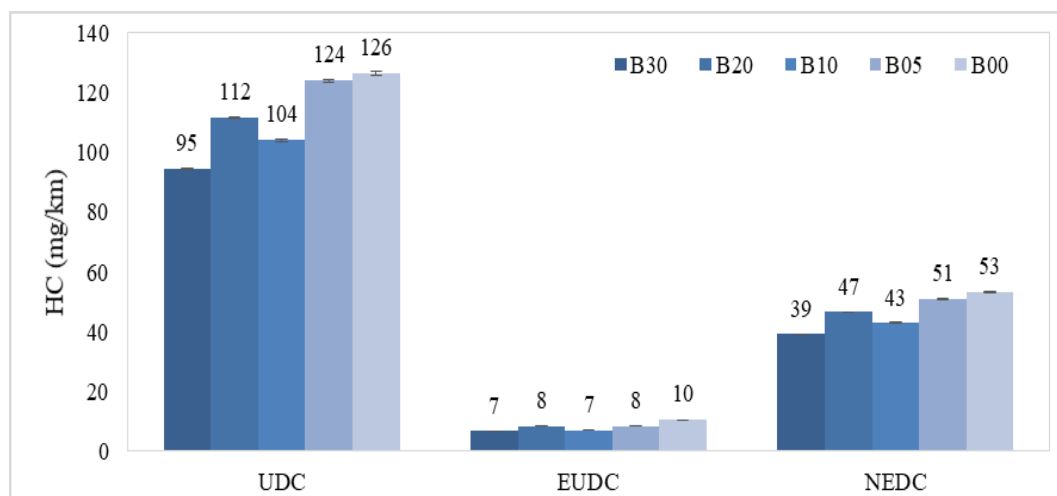


Figure 4.15 : HC emissions of fuel blends under the UDC, EUDC and NEDC.

4.2.5 PN emission

Since the emission of the measured particles was in the order of 10^{10} , PN emission is shown in Figure 4.16 by drawing on the logarithmic scale. Addition of biodiesel reduced the PN emission due to higher oxygen content of the biodiesel than diesel. Besides, higher in-cylinder temperature was observed with the addition of biodiesel, which also caused reduction in PN emission. The use of 10% biodiesel reduced the PN emission from the level of 10^{11} to 10^9 . The reducing effect of biodiesel on PN emission decreased after 10% in volume like HC emission.

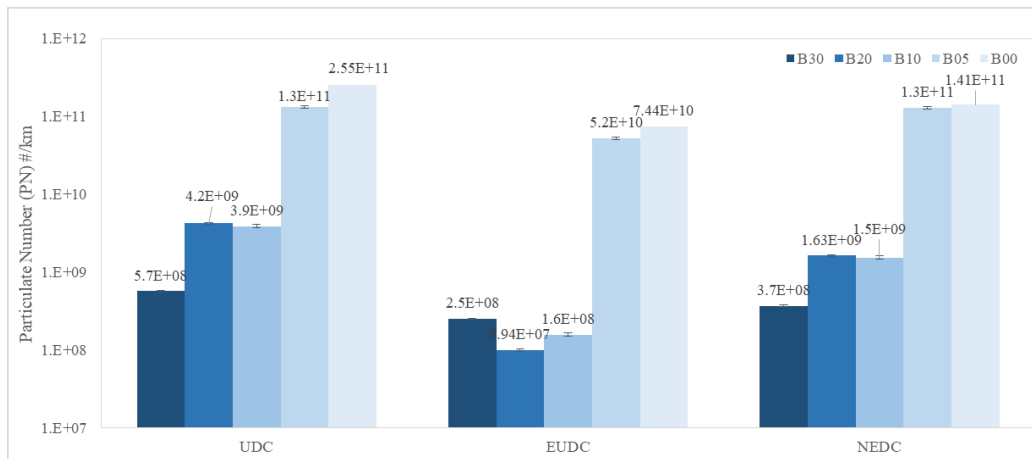


Figure 4.16 : PN emissions of fuel blends under the UDC, EUDC and NEDC.

4.2.6 Evaluation of the results

Utilization of biodiesel reduced the CO and HC emissions while increased the NO_x and CO_2 emissions. EU apply limit for CO and HC emissions. The DOC is utilized in the exhaust systems to satisfy the limits of CO and HC emissions. The amount of valuable metals (platinum, palladium, and rhodium), which are used in the catalyst of the DOC, can be reduced by utilization of biodiesel. Besides, size of the DPF can be reduced as PN emission can be decreased with biodiesel.

Although CO, HC and PN emissions can be reduced by utilizing higher blend ratio of biodiesel, power and NO_x emission worsen with increasing biodiesel ratio. Therefore, an optimum ratio have to be utilized to achieve balanced improvement. Evaluated results indicated that making the B10 standard fuel in the gas stations will reduce the CO, HC and PN emissions with minor impact on fuel consumption and NO_x emission. In this way, OEMs can use cheaper exhaust system in the diesel vehicles for complying emission regulations. Moreover, greenhouse effect of petroleum fuel can be reduced as biodiesel is generated from organic compounds.

4.3 Results of the Cruse Points with Diesel and Biodiesel Blend

In addition to the cycle tests, experiments were carried out at cruising velocities of UDC test in order to obtain effect of biodiesel additive on emissions by putting special emphasis on combustion parameters. The vehicle and engine speeds of the urban part of the NEDC cycle, where the experiments were performed, are shown in Figure 4.17. In the NEDC, the UDC loop is executed four times, besides in the EUDC phase, there are 69 seconds of 50 km/h cruising.

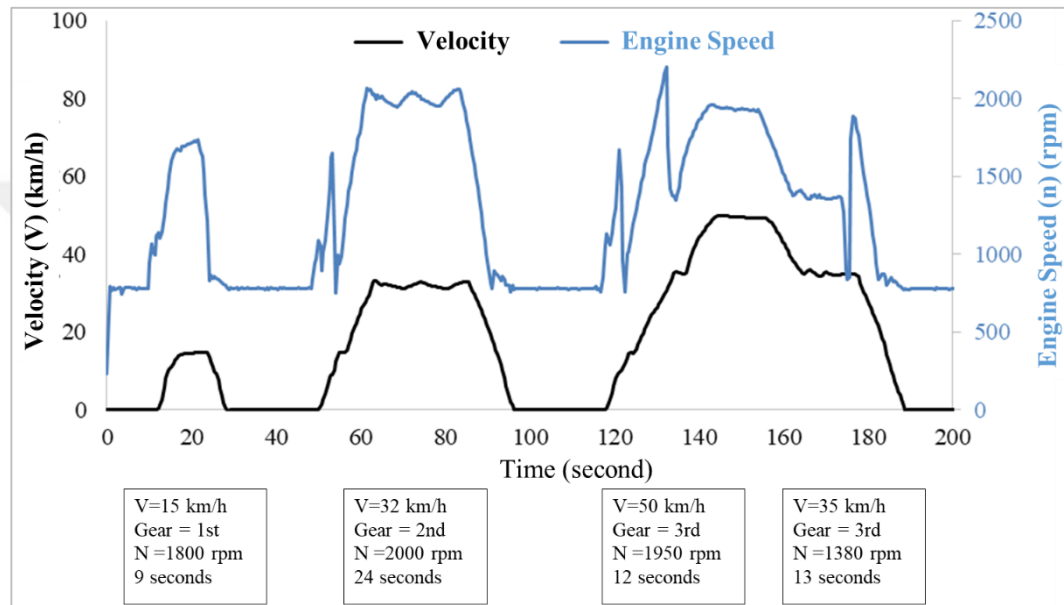


Figure 4.17 : Vehicle and engine speed at urban driving cycle.

Acceleration Pedal Position (APP) and fuel amount were kept constant during the constant engine speed experiments of the B00 and B30 fuels. Engine parameters at constant velocities of NEDC are given in Table 4.2. All listed parameter; the APP, fuel quantity, EGR, and fuel injection pressure were kept constant during the experiments, to be able to investigate the effect of the fuels on emissions and combustion parameters.

Table 4.2 : Engine parameters at cruse speeds of UDC.

Operation Point	APP	Fuel Quantity	EGR	SOI-1	SOI-2	SOI-3	Fuel Pressure
-	%	mm ³ /stroke	%	CA	CA	CA	bar
OP-1 1800 rpm	20	4.9	27	-30.5	-18.4	1.1	570
OP-2 2000 rpm	17	6.6	25	-30.3	-17.1	-0.2	620
OP-3 1950 rpm	13	7.5	23	-30.2	-17.3	0.1	630
OP-4 1400 rpm	18	6.8	26	-24.7	-14.7	0.4	520

Table 4.3 shows the comparison of engine power, NO_x, Opacity, CO and HC emissions with the same amount of B00 and B30 fuel under the same engine parameters. The effects of biodiesel on the emission and power at cruising speed was similar to that of the cycle tests. CO, HC and Opacity emissions were reduced at all operating points. Reduced power was observed with B30. NO_x emission was observed to remain stable at the 3rd operation point, however it was increased with the addition of biodiesel under the other three operating points.

Table 4.3 : Emission and power at part loads of the NEDC with B00 and B30.

Operating Point -	Engine Power (kW)		NO _x (g/kWh)		CO (g/kWh)		HC (g/kWh)		Opacity (%)	
	B00	B30	B00	B30	B00	B30	B00	B30	B00	B30
OP-1 1800rpm	1.24	1.15	4.2	4.8	111.3	96.5	15.7	14.3	20	11
OP-2 2000rpm	2.53	2.38	1.6	1.8	44.3	40.3	10.1	9.4	25	16
OP-3 1950rpm	3.62	3.47	0.4	0.5	28.0	22.2	9.7	6.1	51	26
OP-4 1400rpm	2.67	2.56	0.8	0.8	37.5	37.1	7.5	5.9	12	11

The in-cylinder pressure and injection signal measurements of the B00 and B30 fuels are shown in Figure 4.18. In addition to the measurements, the AVL Indicom program calculates the in-cylinder temperature and rate of heat release (ROHR) according to the equation (4.4) that was also described in FL section. The average temperature in the cylinder was calculated from measured in-cylinder pressure via equation (4.5). The P_i and V_i are the in-cylinder pressure and volume values according to CA. Volumetric efficiency is defined with L . V_h shows the total engine volume. P_s and T_s are pressure, temperature values in the intake manifold, and R is the universal gas constant. In-cylinder temperature was calculated to examine the temperature at the start of the combustion and the average highest temperature during the combustion.

$$Q_i = \frac{1}{\kappa - 1} \times (\kappa \times P_i \times (V_{i+1} - V_{i-1}) + V_i \times (P_{i+1} - P_{i-1})) \quad (4.4)$$

$$T_i = \frac{P_i \times V_i}{m \times R} = \frac{P_i \times V_i}{L \times V_h \times \frac{P_s}{R \times T_s} \left(\frac{R}{(1 + \frac{\sum_{i=1}^{i=n} Q_i \times 28}{P_s})} \right)} \quad (4.5)$$

The values for B00 fuel are shown with continuous lines and B30 with dotted lines. Figure 4.18 also shows the start and end timing of the three separate injections of both cases that were performed with identical injection parameters.

In order to investigate effects of biodiesel on the combustion parameters, Start of Combustion (SOC), temperature (T_{soc}) and pressure (P_{soc}) values at the crank angle where SOC observed were calculated by evaluating in-cylinder pressure measurement. In addition, the highest in-cylinder pressure (P_{max}) and temperature (T_{max}) observed during combustion were calculated. Moreover, mass of the burned fuel (MBF) for 5% (MBF5), 10 (MBF10), 50 (MBF50) and 90 (MBF90) total time for Burn Duration (BD) was calculated in unit of crank angle. All parameters are tabulated in Table 4.4 in order to compare the differences.

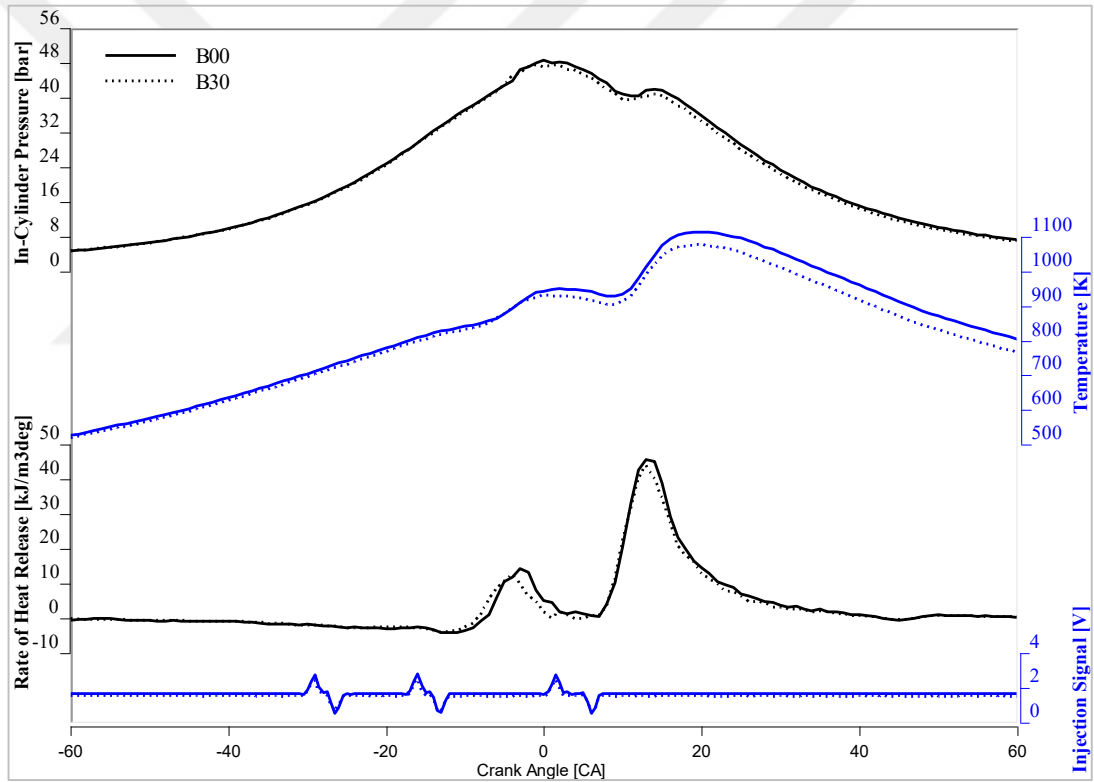


Figure 4.18 : Combustion parameters of the B00 and B30 fuels at 1950 rpm (OP-3).

The biodiesel enable to start the combustion 1 CA earlier compared to pure diesel injection case under all four different operating points. Ignition delay decreased with the addition of biodiesel. Indeed, ignition delay period is divided into a physical and chemical delay. Many parameters have influence on the physical delay (fuel type, air-fuel ratio, compression ratio, engine speed, cylinder gas pressure, the intake-air temperature, and injection pressure). Chemical delay depend on cylinder temperature,

cylinder pressure and fuel properties. Indeed, ignition delay duration mainly depend on the cetane number of the fuel; however, cetane number of the utilized biodiesel blend is slightly higher than pure diesel. Therefore, in this study earlier start of combustion of the biodiesel blended fuel depend on to the physical properties of the fuels. Higher density, viscosity and bulk modulus of biodiesel enable better atomization of it compared to diesel. Moreover, better atomized B30 can react with the oxygen easier than B00 due to the oxygen atoms in its chemical composition.

Consequently, slightly higher cetane number, lower compressibility (bulk modulus), and higher viscosity of biodiesel enable easier ignition compared to diesel. The shorter ignition delay of biodiesel also has a considerable influence on the exhaust emissions as the amount of fuel burned in premixed combustion phase directly depend on to ignition delay duration. Increasing the burned fuel fraction in the premixed phase increases NO_x emission however decreases unburned HC, CO and soot emissions [98].

Since combustion started earlier in biodiesel experiments, the cylinder had less compressed air pressure at the start of combustion and therefore the temperature (T_{soc}) and pressure (P_{soc}) at which the combustion started were lower than that of pure diesel. According to the results observed in the experiments, it is understood that biodiesel is more easily ignited than diesel.

The highest in-cylinder pressure (P_{max}) and temperature (T_{max}) of the B30 were lower than the B00 at four operating points, which can be explained with the lower energy density of the biodiesel. Burn duration (BD) of the injected same amount of fuel is longer for B00 compared to B30 as given in Table 4.4. There was at least 1 CA difference between the BD of B00 and B30 for all test points. B30 burned faster than B00 due to earlier SOC, higher amount of premixed combustion phase and lower AFR of biodiesel blended fuel compared to pure diesel. The faster burn of biodiesel led steeper rise of in-cylinder pressure and temperature and caused local high temperature zones, which increased NO_x emissions.

Table 4.4 : Comparison of the combustion characteristics of B00 and B30 fuels at cruise velocities of NEDC.

Operation Point	SOC		IDD		T _{soe}		P _{soe}		P _{max}		T _{max}		MBF5		MBF50		MBF90		BD	
	CA	ΔCA	ACA	K	K	bar	bar	bar	bar	K	K	ACA	ACA	ACA	ACA	ACA	ACA	ACA	ACA	ACA
OP-1 1800 rpm B00	-8.4	-6.9	-6.9	800	800	37.1	47.1	47.1	1026	6.2	14.2	25.4	19.2							
OP-1 1800 rpm B30	-9.5	-8.0	-8.0	793	793	35.0	47.0	47.0	1006	8.2	13.9	23.9	15.7							
OP-2 2000 rpm B00	-7.7	-6.2	-6.2	851	851	39.5	48.8	48.8	1114	10.2	15.2	27.9	17.7							
OP-2 2000 rpm B30	-8.4	-6.9	-6.9	836	836	37.8	47.7	47.7	1077	10.4	15.2	27.1	16.7							
OP-3 1950 rpm B00	-8.4	-6.9	-6.9	764	764	35.3	46.1	46.1	1095	0.2	12.0	23.2	23.0							
OP-3 1950 rpm B30	-9.5	-8.0	-8.0	766	766	33.9	46.4	46.4	1082	0.2	11.6	21.0	20.8							
OP-4 1400 rpm B00	-7.0	-5.5	-5.5	833	833	39.5	47.4	47.4	1123	9.9	15.4	28.5	18.6							
OP-4 1400 rpm B30	-7.9	-6.4	-6.4	827	827	38.3	47.5	47.5	1089	9.5	14.5	25.6	16.1							

4.4 Evaluation of the Effects of Biodiesel on Emissions and Performance by Comparing the Results of the New European Drive Cycle and Worldwide Harmonized Light Vehicles Test Cycle

NEDC has been used for twenty years for the Type Approval (TA) procedure, new TA procedure was introduced since 2018 with new regulation cycle namely Worldwide Harmonized Light Vehicles Test Cycle (WLTC). European Commission developed a longer cycle, in terms of both duration and distance than the NEDC for the TA procedure to simulate more realistic driving attitudes [99]. In order to evaluate the differences of each cycle comparison tests were performed with B00 and B30 fuel.

Impact of biodiesel on emission and performance increases when higher amount of biodiesel is blended to diesel. Nevertheless the effects of biodiesel blend is not proportional to blend ratio [100,101]. Moreover, higher blends of biodiesel to diesel more than 30% may exceed the legal limit of EN590 and are not common [102], therefore B30 was selected for experiments to compare the differences of WLTC and NEDC.

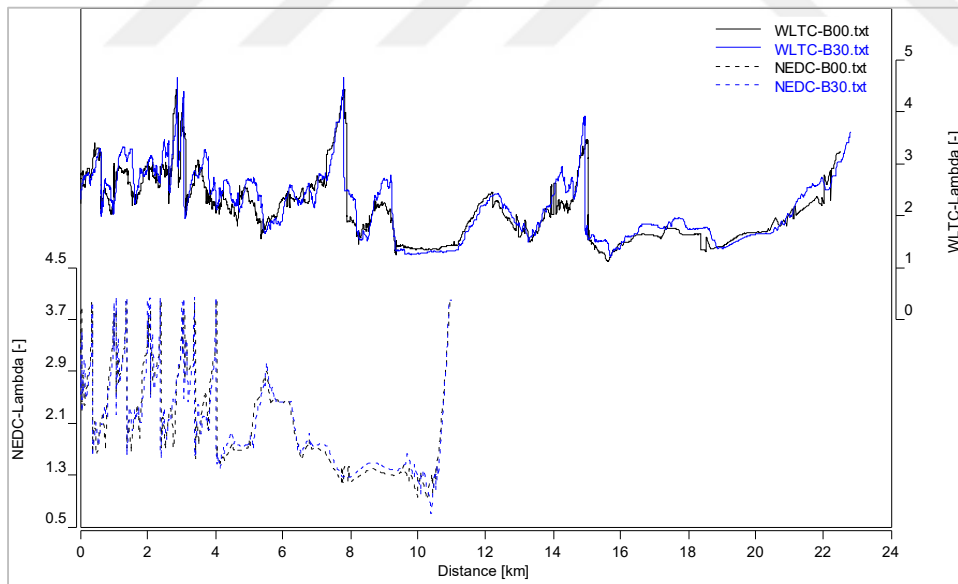


Figure 4.19 : Lambda (λ) change of B00 and B30 over the NEDC and WLTC.

Lambda (λ) of each fuel was calculated via equation (4.2). AFR of B00 and B30 was utilized from Table 2.4 (14.54 for B00, 13.92 for B30), which was calculated by equations (2.1) and (2.2). MAF and F_{mass} are the air and fuel flow to the engine, which were gathered from On Board Diagnostics (OBD) port of the vehicle. Figure 4.19

shows the λ change over both cycle for B00 and B30. Over the WLTC, λ altered from 1.2 to 4.5 according to engine operating point, whereas λ altered from 1.0 to 4.0 over the NEDC. In both cycle blue line (B30) is slightly higher than black line (B00), therefore B30 has higher lambda than B00 as can be seen in Figure 4.19.

4.4.1 Comparison of load profile and cycle dynamism of the WLTC and the NEDC

Travel duration and distance of the WLTC and NEDC are different. The WLTC are 23.27 km and it takes 1,800 seconds to complete whole cycle. On the other hand, the NEDC lasts 1,180 seconds with a total distance of 11.03km as can be seen in Figure 4.20.

Besides, engine of the test vehicle has split injection strategy as shown in Figure 4.20. SOI of the main injection was between the ranges of 3 to 6 crank angle (CA) before top dead center (bTDC). There is 12 CA duration between start of main and pilot injection. SOI of pilot injection was changing from 10 to 19 CA bTDC. More advanced SOI were observed for the main and pilot injections when engine run under higher loads, where the velocity of vehicle is over 50 km/h, compared to lower engine loads.

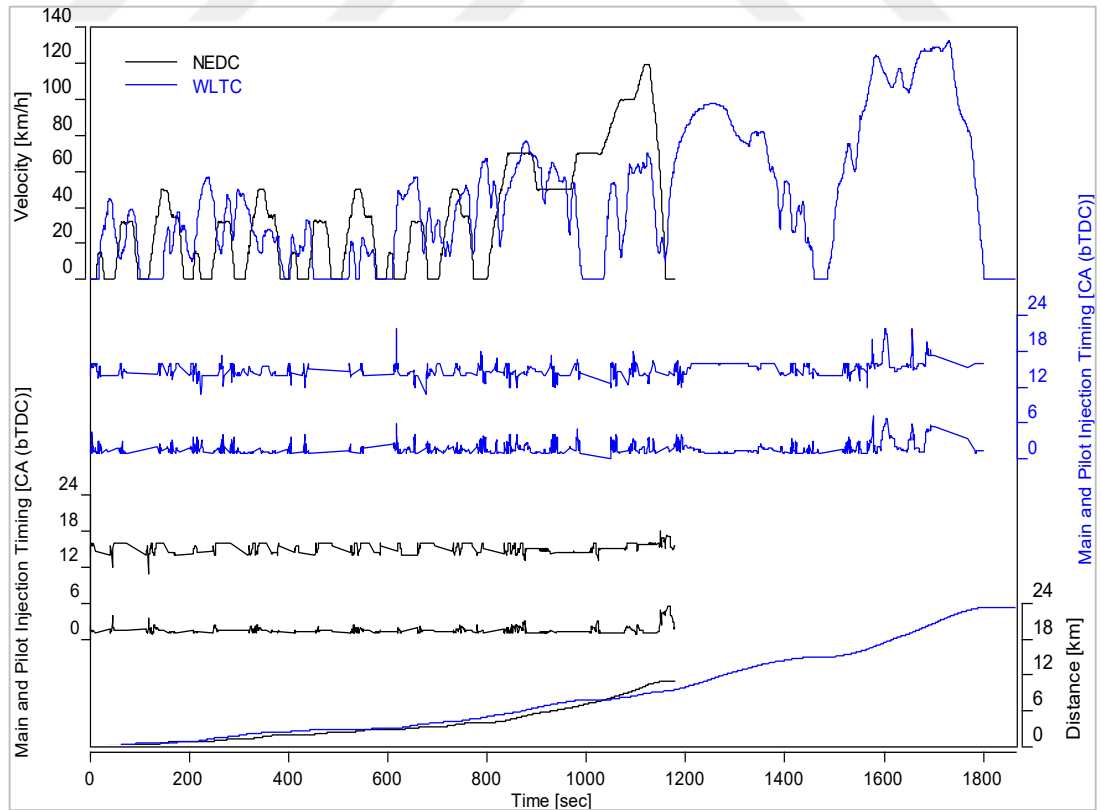


Figure 4.20 : Velocity, distance, main and pilot injection timing of the NEDC and WLTC.

In order to understand and compare the power distribution of the NEDC and WLTC, gathered wheel load data is grouped as shown in Figure 4.21. Injection timing also added to Figure 4.21, because both cycle has same injection configuration when the engine loads were same, there is one SOI data for pilot and main injection per load condition in Figure 4.21. Brake label is used for deceleration operation and total braking time is divided to the total cycle time to show frequency of braking duration in the overall cycle. The remaining zero (idle) and positive wheel power operation points are also grouped as braking operation and shown in Figure 4.21. Frequency of idling and power between 0 - 5 kW operation under the NEDC are 32% and 26%, which is more than half of the cycle, whereas under the WLTC are 16% and 18% respectively, which enables the WLTC to have a better distribution of the load over the cycle. Green and red bars show the main and pilot injection timings respectively. As there is no fuel injection under braking, injection time is 0. Engine has different injection configuration under idle; main injection timing was only after TDC at idle. Main and pilot injections were injected earlier under higher load conditions compared to lower load condition as can be seen in Figure 4.21. Pilot injection advanced from 15.8 to 19.1 CA bTDC from 5 to 40 kW. Similarly main injection timing altered from 2.4 to 6.2 CA bTDC.

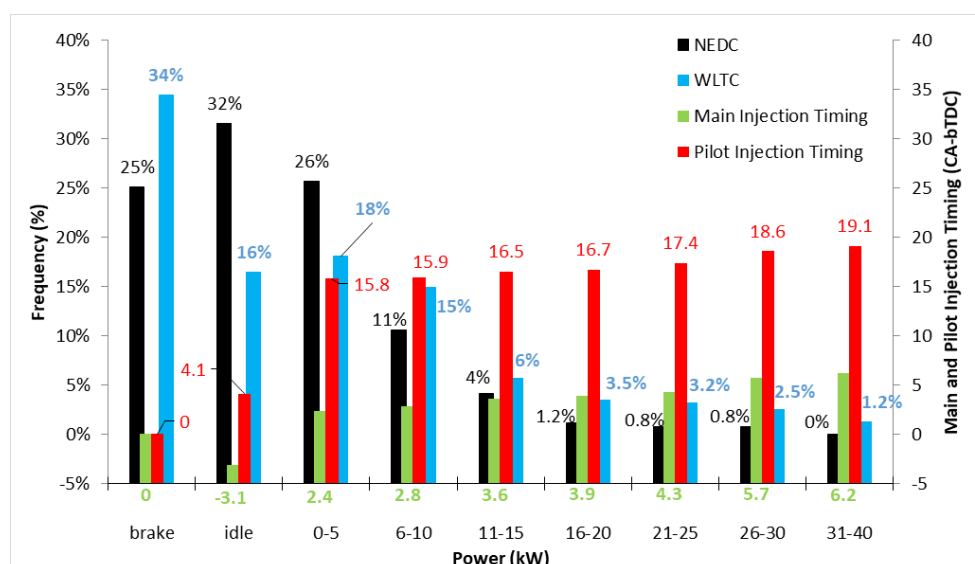


Figure 4.21 : Frequency and distribution of power and timings of main and pilot injections over the NEDC and WLTC.

Additionally, to realize comparative effects of the biodiesel derived from canola oil on emission and performance of the WLTC and the NEDC, results of each cycle are also compared for power and emission point of view.

4.4.2 CO₂ emission and fuel consumption

Biodiesel added fuel emits more CO₂ emission than diesel fuel under both the NEDC and the WLTC as shown in Figure 4.22. Although, identical engine parameters were used during biodiesel and diesel tests, the injected amount of biodiesel-blended fuel is higher than pure diesel due to higher density and lower GCV of biodiesel compared to diesel. Besides, carbon to hydrogen ratio of biodiesel is higher than diesel. Moreover, higher amount of oxygen content of biodiesel led more lean mixture and more complete combustion. Combination of these effects (higher amount of fuel, higher carbon ratio and lean mixture) increase the CO₂ emission when biodiesel is utilized. As CO₂ emission is directly related with fuel consumption, results of the experiments showed that biodiesel added blends have higher fuel consumption than pure diesel under the NEDC and WLTC, which is also due to biodiesel's lower GCV than pure diesel.

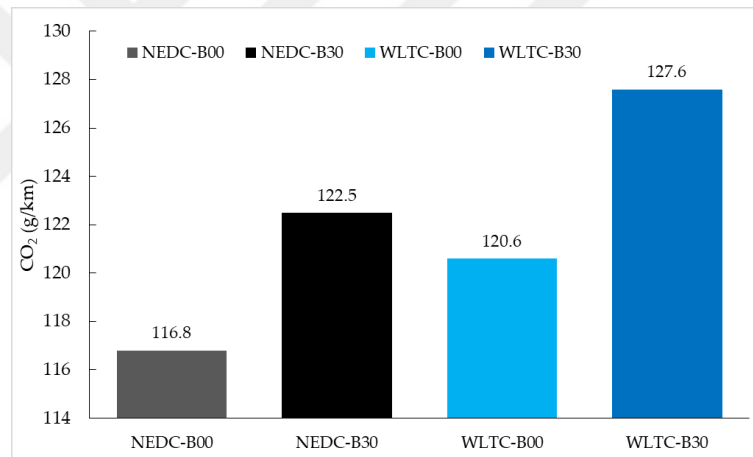


Figure 4.22 : CO₂ emissions of the WLTC experiments with B00 and B30.

Aside from the wheel power, engine load data according to engine speed under the WLTC and NEDC plotted to analyze the engine load profile as shown Figure 4.23. Besides, full load power of the engine, which was measured under fully open acceleration throttle position in chassis dynamometer, added to Figure 4.23 to indicate the upper power limit of the engine. Engine operates under higher speed and power conditions in the WLTC compared to the NEDC, which is consistent with the velocity profile of each cycle. Moreover, higher CO₂ emission in g/km released over the WLTC (120.6 and 127.6) than the NEDC (116.8 and 122.5) with B00 and B30 respectively. Therefore, the WLTC that is operated under more transient and higher load cause higher CO₂ emission compared to the NEDC.

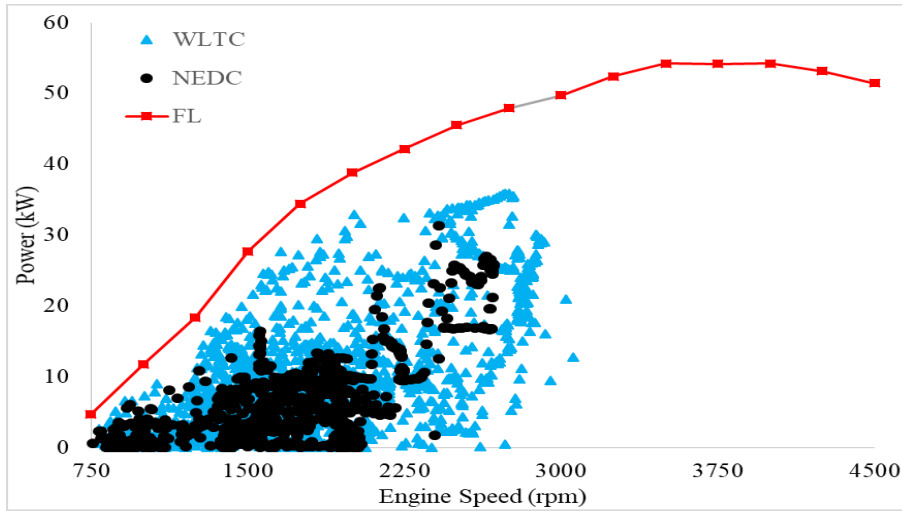


Figure 4.23 : Power distribution of the NEDC, WLTC and FL.

4.4.3 Comparison of CO, HC, and NO_x emissions

In this section, CO, HC, and NO_x emissions of the WLTC and the NEDC are compared for B00 and B30 in order to understand and compare emission sensitivity of each cycle. CO and HC conversion efficiency of the DOC is directly related to exhaust temperature due to working principle of the DOC. HC and CO in the exhaust can be converted to H₂O and CO₂ when the temperature at the entrance of the DOC reaches 200°C, which is the so-called light off temperature. Owing to higher velocity profile at the beginning of the WLTC compared to the NEDC, exhaust temperature escalation initiated earlier in the WLTC. Moreover, due to the relatively shorter idle duration in the WLTC (compared to the NEDC), resulted in better CO and HC emission conversion over the DOC as shown in Figure 4.24.

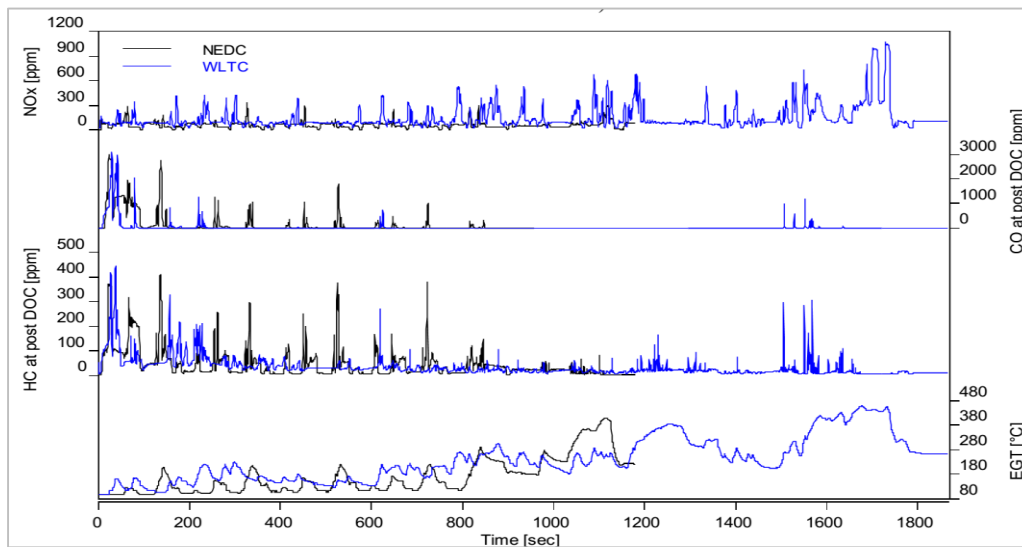


Figure 4.24 : Comparison of NO_x, CO, and HC emissions at post DOC and exhaust gas temperature.

In addition, the utilization of B30 reduces CO and HC emissions by 34% and 33% under the NEDC whereas the reduction becomes 55% and 40% under the WLTC respectively as can be seen in Figure 4.25. Therefore, it is concluded that vehicles can satisfy the CO and HC regulation limit easier under the WLTC compared to the NEDC.

In contrary to CO and HC emissions, in both cycles NO_x emission increases. Rise in NO_x emission of B30 fuel is higher over the WLTC than the NEDC as can be seen in Figure 4.25. The reason of the high NO_x emission can be explained with continuous measurement data that is shown in Figure 4.24. Due to transient and higher velocity operation profile under the WLTC, NO_x emission is higher during the WLTC. Especially in the last 200 seconds of the WLTC, where vehicle speed was over 130 km/h, the NO_x emission reached 900 ppm, while maximum NO_x emission during the NEDC was approximately 300 ppm as shown in Figure 4.24. The vehicle was optimized for NEDC, not for WLTC, this was the main reason for NO_x peaks during WTLC test.

In summary, experiments show that lower CO and HC emissions can be achieved under the WLTC compared to the NEDC while there is clear NO_x emission increase. This result is consistent with the Bielaczyc et al.'s experiments that compare emission of six different vehicles with pure diesel under the NEDC, WLTC, and FTP-75 [103]. In addition, variation of CO, HC, and NO_x emissions from B00 to B30 is higher in percentage under the WLTC than the NEDC, which may be attributed to higher in-cylinder temperature during the WLTC due to its higher engine speed and load profile compared to the NEDC.

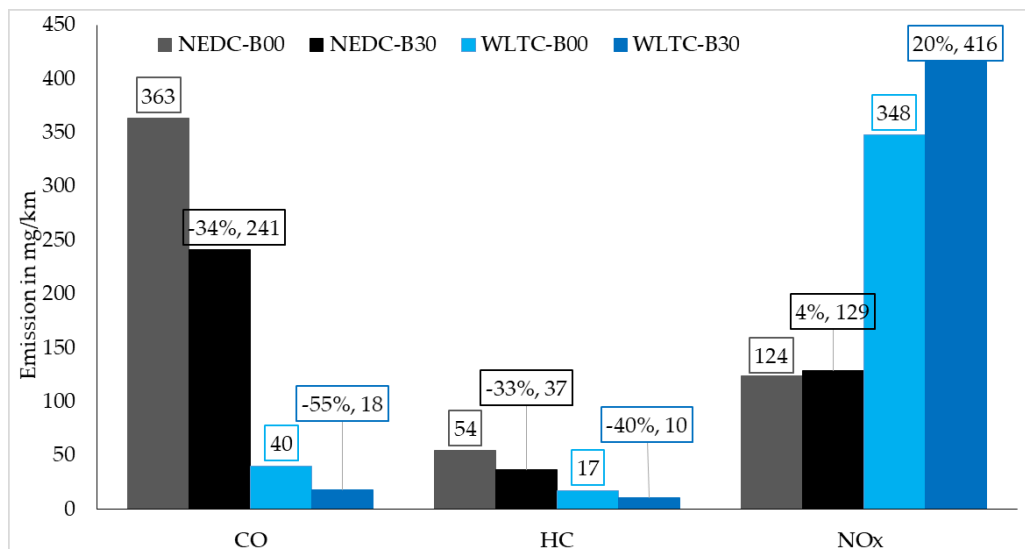


Figure 4.25 : CO, HC, NO_x emissions of the NEDC and WLTC with B00 and B30.

4.4.4 Comparison of particulate number emission

In addition to gaseous exhaust emissions (CO, HC, and NO_x), EU regulation have been restricting the particulate mass (PM) and the particulate number (PN) emissions due to their harmful effects on human health. Main source of the soot emission is the carbon atoms in the diesel fuel. Soot formation begins once the fuel has hydrogen to carbon ratio around 2 and contains 12 to 22 carbon atoms [11]. However, regardless of the carbon atom content, lower aromatics and higher oxygen content of biodiesel prevents the creation of particles and enhance the oxidation of particulate matter [68,72]. As PN and PM emissions have similar trend [104], only emitted PN emission under the NEDC and WLTC were compared.

The PN emission is plotted on a logarithmic scale in Figure 4.26 due to the large size of the measured value. Moreover, reduction rate of the PN emission was calculated over logarithmic value. Engine emit 25% and 31% lower PN emission under the NEDC and WLTC respectively with B30 compared to B00 as shown in Figure 4.26. Although higher PN emission was observed in the WLTC than the NEDC with B00, PN emission under the WLTC was reduced below to the PN emission under the NEDC with B30, which indicate that the WLTC is more sensitive than the NEDC to biodiesel use for also PN emission. Higher load profile (i.e. in-cylinder temperature) of the WLTC compared to the NEDC may be the reason of the higher PN emission decrease of the WLTC as Leung et al. was also found that biodiesel causes larger decreases in soot emission under higher load conditions of their single-cylinder engine [105].

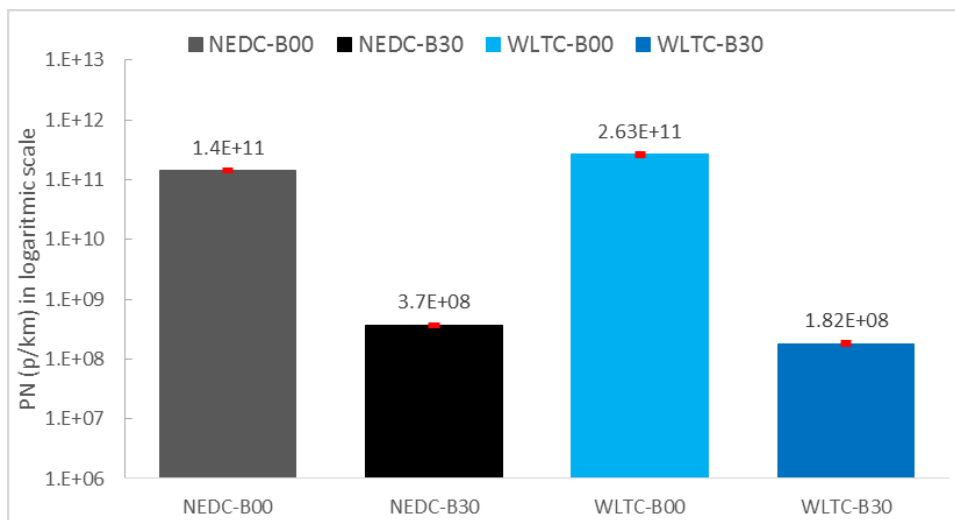


Figure 4.26 : PN emission of the NEDC and WLTC with B00 and B30.

4.5 Advanced Combustion Experiments at Selected Operation Points of the Regulation Cycles

Steady state and transient experiments were performed during the thesis study as mentioned in the previous sections. In order to investigate the possibility of to improve the performance and the emissions of the vehicle under regulation cycle, fuel injection parameters were modified. Operation points were selected from the cruise points of the NEDC, which were also repeated in the WLTC.

Experiments in the literature revealed that the LTC and HCCI methods can improve the NO_x and soot emissions however cause a fuel consumption increase. Once researchers understood the limits of these methods, they focused on premixed combustion method. As the aim of the thesis is to investigate the change in diesel combustion, engine fuel parameters were modified in a wide range. In this way, advanced combustion methods (PCI and PPCI) can be investigated and compared with each other.

Original injection configurations of the vehicle can be seen in Figure 4.27. During the NEDC tests, different injection configurations were observed for same engine speed. ECU of the engine determine the injection strategy according to engine coolant temperature and vehicle speed. It was observed that once the coolant temperature is above 60 °C injection was split into two parts. Single injection configuration was available when the coolant temperature was above 80 °C and speed was over 90 km/h. Injection configuration was optimized according to vehicle, therefore it may be different for other vehicles. Observed three different injection configurations, which is given in Figure 4.27, were investigated to clarify the differences of the single and split injections. Injection quantity was kept 6.6 mg/stroke for the all cases.

First set of experiments were performed with single injection configuration by adjusting the two parameters; EGR and SOI. These experiments were performed with B00 and B30. As B30 lead lower power and higher NO_x emission compared to B00, it was not used at the experiments of the split injection configuration.

Second set of experiments were performed with double injection configuration by adjusting five of the engine parameters. EGR, start of injection-1 and 2, quantity of injection 1 and 2 were modified by keeping injected fuel as 6.6 mg/stroke.

Third set of experiments were performed with triple injection configuration. EGR, SOI of injection 1, 2 and 3 were modified during these experiments by keeping total injection quantity constant (same as the previous experiments, 6.6 mg/stroke).

Last experiments were performed with single injection of 12 mg/stroke to investigate the results of advanced combustion at higher load conditions.

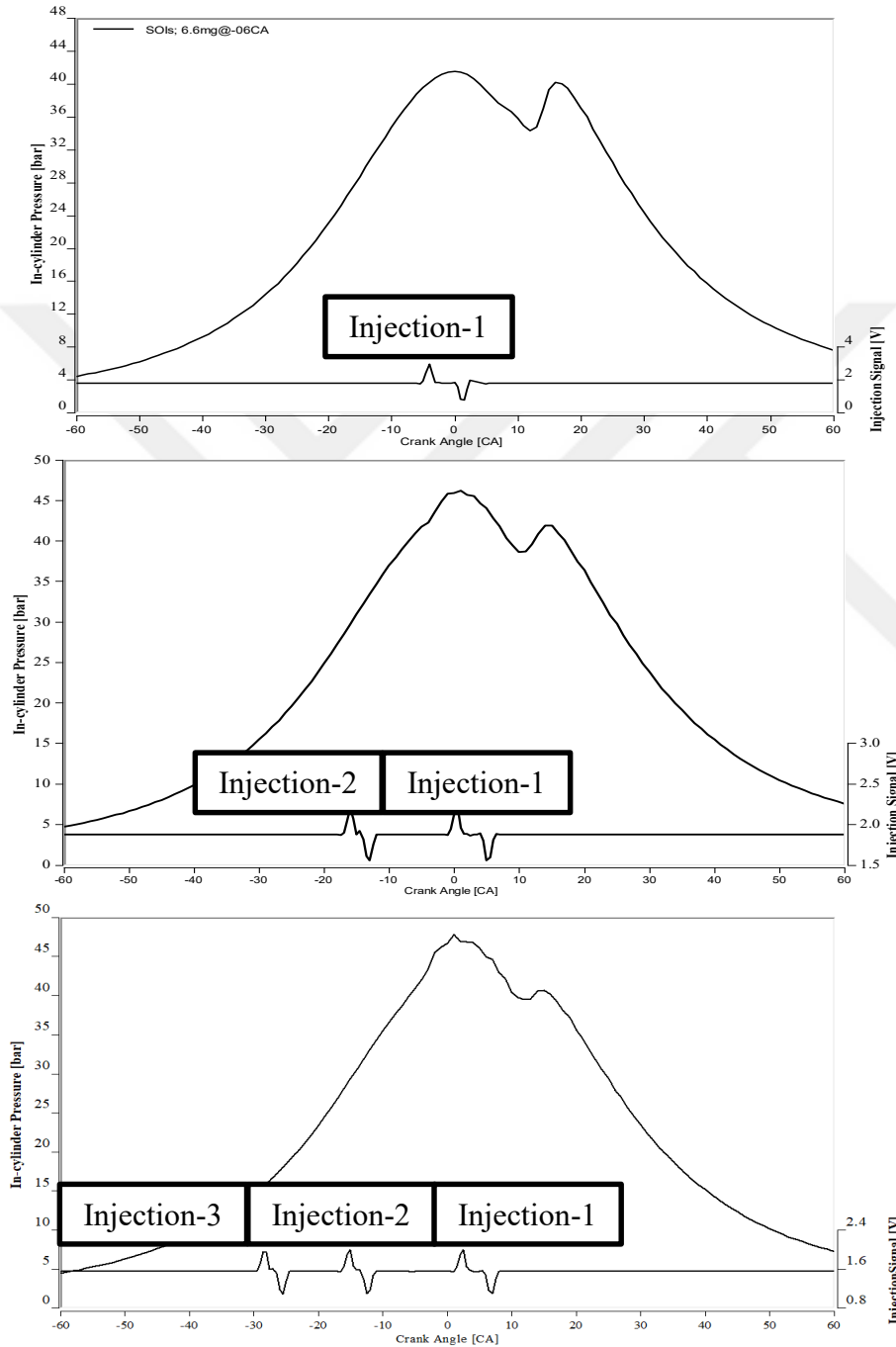


Figure 4.27: Injection definitions for single, double and triple injection configurations.

The EGR mass flow was calculated by extracting the airflow that was measured by the airflow sensor, from total mass flow that was calculated via ECU by utilizing the temperature and pressure data of the each intake port of the engine. Besides, temperature of total airflow, clean airflow and EGR flow were measured via instrumented thermocouples. Figure 4.28 shows the mass flow and temperature of the sucked clean air, calculated EGR and total flow for 6.6 mg injection at 2000 rpm.

Red lines in Figure 4.28 show the calculated EGR mass flow and temperature according to EGR valve position. Although the rise in EGR valve position was continuously increased, a higher EGR flow increase rate was observed up to 40 percent of the EGR valve position. Black lines in Figure 4.28 show mass flow and temperature of the sucked clean air and temperature. Temperature of the air at the entrance of the compressor was not change according to EGR valve position however mass flow reduced with increasing EGR valve position. Blue lines at Figure 4.28 show total mass flow and temperature of the mixture of EGR and clean air. Due to the increased temperature of air, total mass flow reduced slightly with increasing EGR.

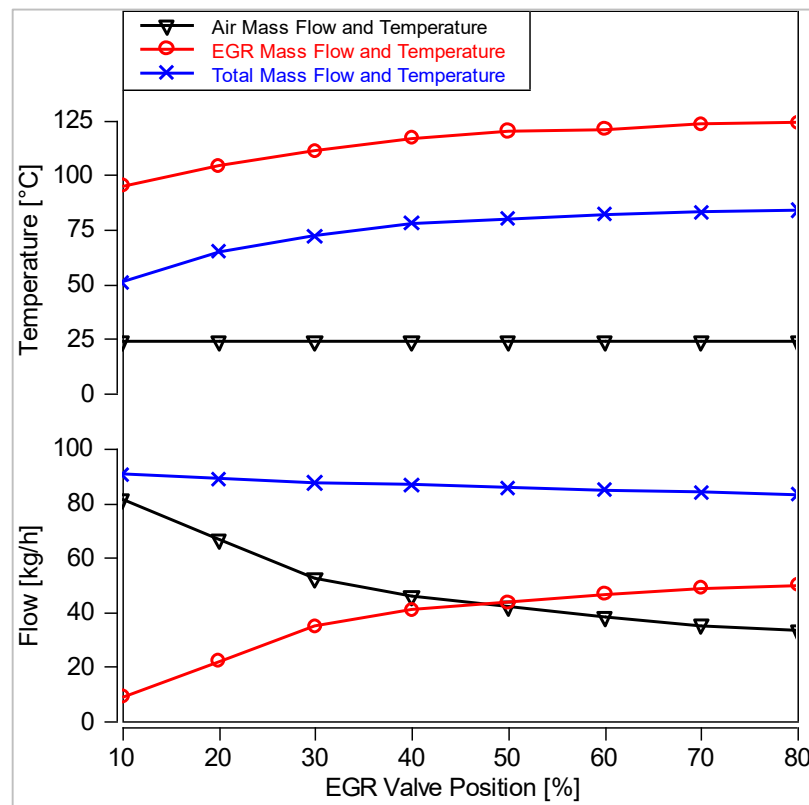


Figure 4.28 : Temperature and Flow Change According to EGR Valve Position.

EGR valve position was converted to the EGR rate from the mass flow of EGR and clean air. Results of the correlation were shown in Table 4.5. EGR Rate was used

during the study instead of EGR valve position, to indicate the flow rate of the EGR in the total mass flow.

Table 4.5 : Correlation of EGR valve position and EGR rate.

EGR Valve %	EGR Rate %	Air Flow kg/h	EGR Flow kg/h	Total Flow kg/h
10	10	81.5	9.1	90.5
20	25	66.7	22.2	88.9
30	40	52.4	34.9	87.3
40	47	45.8	40.7	86.5
50	51	41.9	43.6	85.5
60	55	38.2	46.7	84.9
70	58	35.3	48.7	84.0
80	60	33.3	49.9	83.2

4.5.1 Advanced combustion with single injection strategy

Emissions and performance of the vehicle with single injection configuration of 6.6 mg at 2000 rpm is shown in Figure 4.29 with in-cylinder pressure measurement. Original ECU of the engine started to inject the 6.6 mg fuel at -6 CA as shown in Figure 4.29. Fuel injection pressure and EGR rate was 620 bar and 30% respectively. Engine power was 3.87 kW under these parameters. 64 ppm NO_x emission and 3.9% opacity was observed at this single injection operation condition.

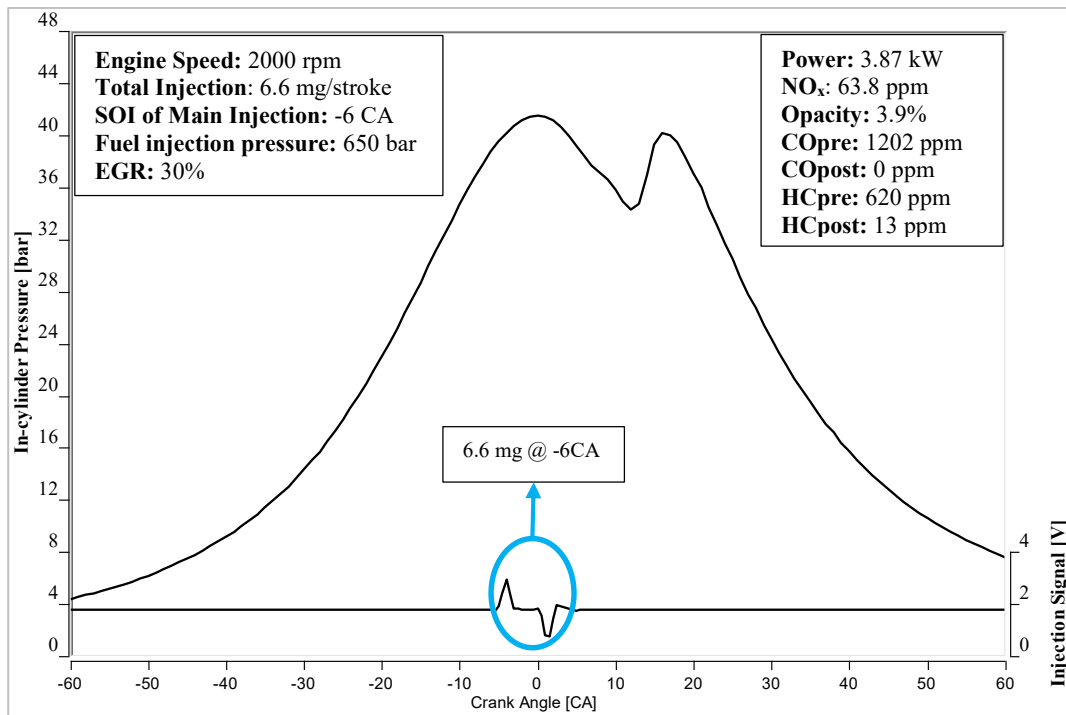


Figure 4.29 : Original engine parameters for single injection configuration.

4.5.1.1 Adjustment of the SOI under 30% EGR

First experiments were performed by adjusting the SOI from -6 to -45 CA as shown in Figure 4.30 while keeping EGR rate constant at 30%.

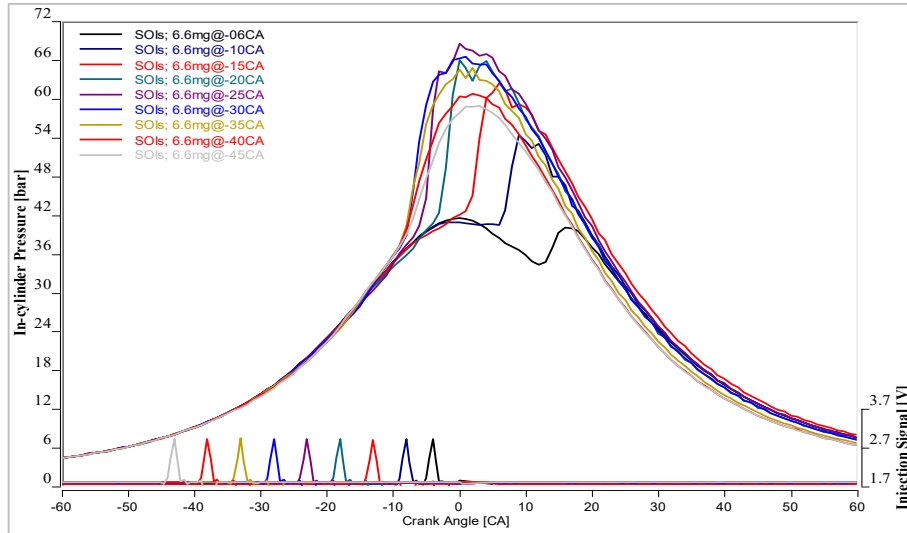


Figure 4.30 : Adjustment of SOI.

There are nine different lines for in-cylinder pressure according to SOI on Figure 4.30, it is hard to make a clear distinction between these lines. In order to reveal the differences at ROHR and in-cylinder pressure on the graphic, four of the SOIs, which indicate trend of whole range, were selected and plotted as shown on Figure 4.31.

ROHR of the SOI at -6 CA reached to its peak value later compared to other SOIs as can be seen in Figure 4.31, therefore fuel started to burn later when the SOI was at -6 CA. Rise in in-cylinder pressure was also observed after TDC for SOI -6CA. The highest heat release was observed when the SOI was at -15 CA, because limited fuel started to burn bTDC as can be understood from the rise of ROHR in Figure 4.31. Injected fuel created mixture up to TDC and started to burn at TDC, which led the peak in-cylinder pressure to be between 5-10 CA. Once the SOI was advanced to -30 CA, fuel was started to burn earlier compared to -15 CA, which led to higher in-cylinder pressure but lower power. The combustion was worsened as advancing the SOI more to -45 CA, due to ROHR during the compression stroke. This resulted in very low power output compared to the SOIs after -30 CA. As further advancing the SOI did not improve the emission and power output of the engine, experiments were not performed for the SOIs earlier than -45 CA.

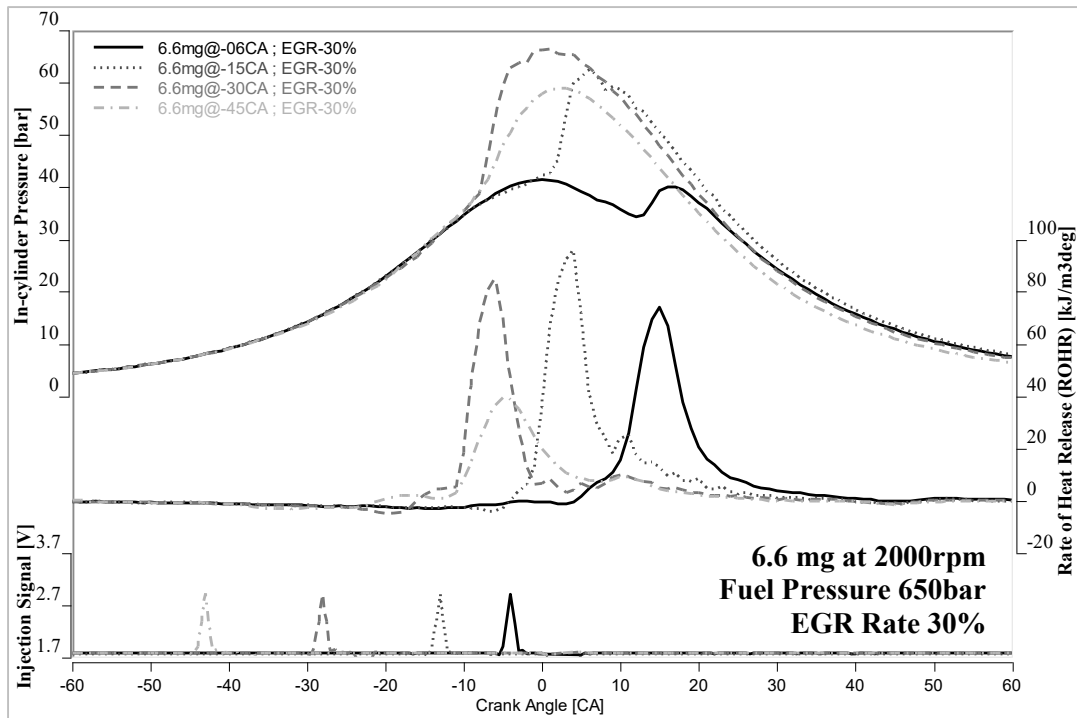


Figure 4.31 : Combustion data of SOI at -06, -15, -30 and -45CA.

SOIs of -6, -15, -30 and -45 has different ROHR and in-cylinder pressure characteristics as shown in Figure 4.31. Different SOIs in Figure 4.31 simply represents the idea of different advanced combustion methods in the literature;

- SOI at -6 CA represent the late injection advanced combustion that aim to burn the fuel in the expansion cycle by injecting the fuel around TDC.
- SOI at -15 CA represent the conventional combustion before emission regulations that try to achieve the highest in-cylinder pressure at 5CA aTDC.
- SOI at -30 CA represent the premixed combustion (PCI), however EGR is not high enough in this case. Therefore, ignition delay is not long enough and fuel started to burn in the compression stroke of the engine.
- SOI at -45 CA represent almost homogenous charge compression ignition (HCCI), where fuel and air has a mixture preparation duration of 30 CA up to SOC. Low NO_x and Opacity was observed under this condition but like HCCI studies in literature, power reduction was observed.

Effect of SOI on power, SOC, NO_x and opacity was plotted in Figure 4.32. Dotted red line in the plots shows the values of the original condition of the engine that was also shown in Figure 4.29. As these are the original output of the engine, reference label

was utilized in the plots.

There are 12 different SOI conditions in Figure 4.32 (-2,-3,-5,-8,-10,-15,-20,-25,-30,-35,-40,-45) to realize the trend in the change of power, SOC, NO_x and opacity.

The highest power was observed for the SOI of -15 CA as shown in Figure 4.32-a. This can be explained with the observed peak in-cylinder pressure at 5 CA bTDC, which was shown in Figure 4.31. Very early and late injection of fuel caused power decrease. When the SOI was earlier than -30 CA, steep power decrease was observed. Similarly, when the SOI was later than -5 CA, power was decreasing sharply.

Start of combustion were plotted at in Figure 4.32-b, SOC was continuously occurred earlier with advancing the SOI from -2 to -45 CA. However, the effect of SOI on SOC was reducing for the SOIs earlier than -30 CA.

NO_x and Opacity emissions have also similar trend with power as seen in Figure 4.32, respectively. NO_x emission is compared by using the ppm unit instead of gr/kWh to eliminate change in power. Although, the highest power was achieved at SOI -15 CA, the worst NO_x and opacity emissions were also observed under this condition. Injected fuel burned rapidly and caused local rich zones in the cylinder, this may be the reason of the highest opacity. Steep rise of the pressure caused also local high temperature zones in the cylinder, which caused rise in NO_x emission. Therefore -15 CA is not a suitable alternative for achieving stringent emission limits. NO_x of the reference condition is lower than most of the SOIs as seen in Figure 4.32. However, increasing the EGR rate led better NO_x emissions. Opacity was reduced with advanced SOI due to achieving more homogenous mixture in the chamber.

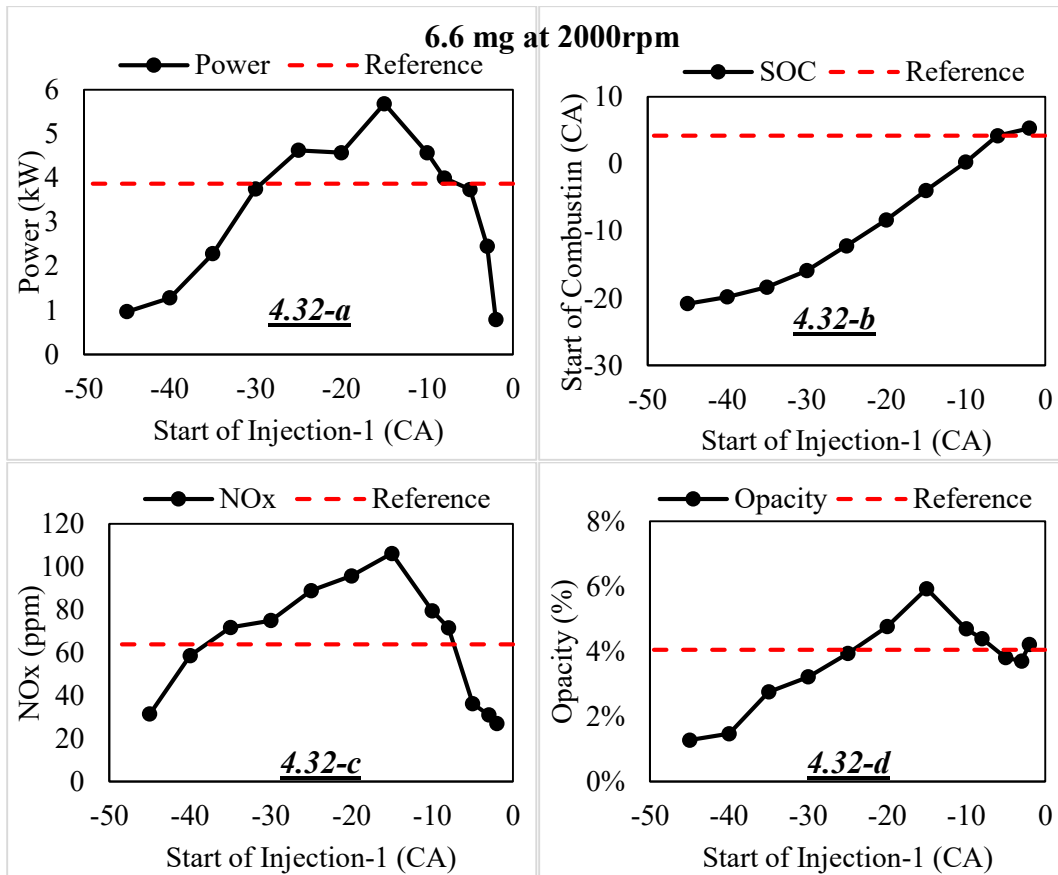


Figure 4.32 : Power, SOC, NO_x and Opacity change according to SOI.

4.5.1.2 Adjustment of the EGR rate under SOI -25 CA

EGR were modified while keeping other engine parameters constant as shown in Figure 4.33. 6.6 mg fuel was started to inject at -25 CA in all experiments and EGR rate was increased from 15% to 60%.

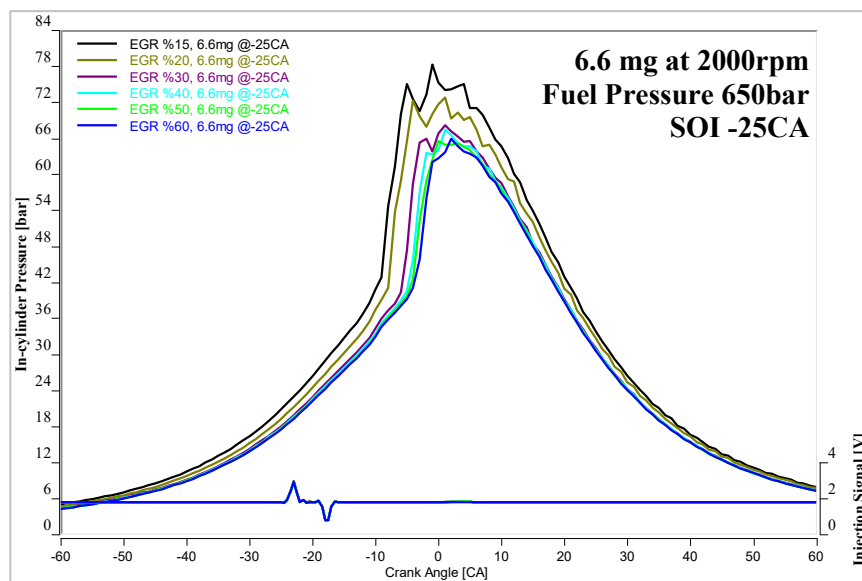


Figure 4.33 : In-cylinder pressure change according to EGR.

As it is hard to follow up the plots of all cases in one graph, Figure 4.34 was plotted with three WGR rates to investigate the combustion data trend according to EGR rate.

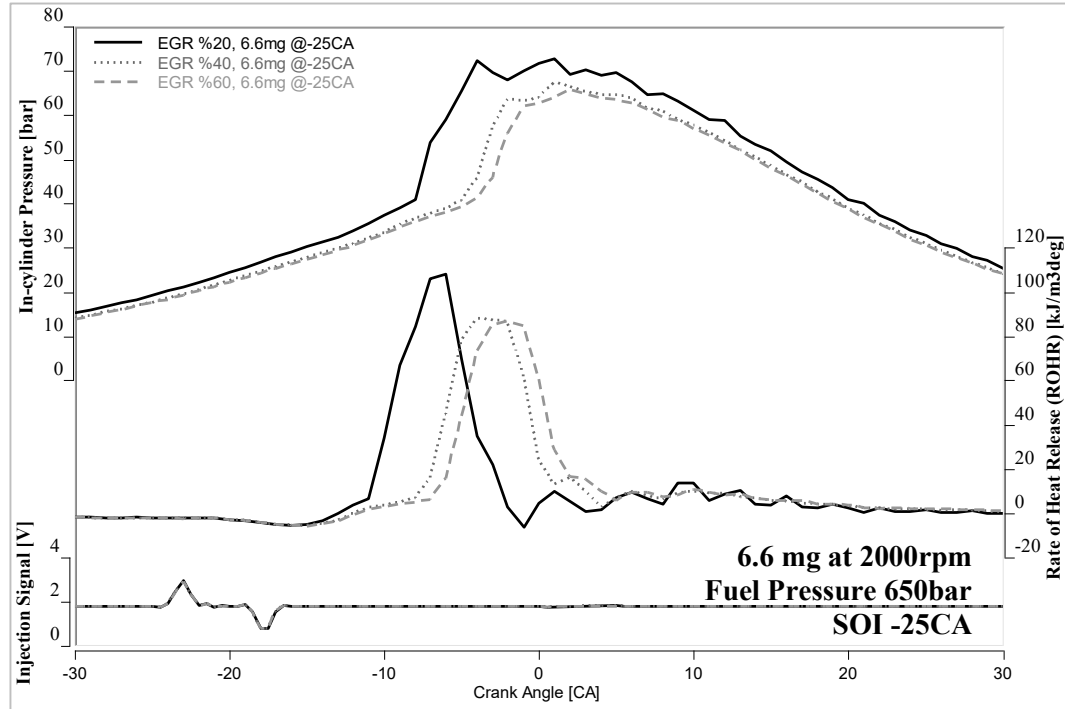


Figure 4.34 : Combustion data of EGR at 20%, 40% and 60%.

When the EGR rate was low, injected fuel could start to burn earlier as seen in Figure 4.34. Fuel ignited easier with lower EGR, because available oxygen in the combustion chamber is higher under 20% EGR compared to 40% EGR. Increasing EGR ratio reduced both temperature and available oxygen in the chamber, which led longer ignition delay period. In this way, ignition of the very early injected fuel can be postponed towards to the expansion cycle as can be seen in Figure 4.34.

Increasing the ignition delay period for very early injected fuel led more power output as can be seen in Figure 4.35-a. In the conventional combustion region rise in EGR rate causes a power loss. However, in this case power was improved with increasing EGR rate. There is limited data in the literature about the power benefit of the EGR under advanced combustion condition (SOI is earlier than -20 CA). In addition to improvement at power, high EGR rates also reduced the opacity as shown in Figure 4.35-d. Prolonged ignition delay duration with increasing EGR rate enabled the injected fuel to create more homogenous mixture in the combustion before SOC, which may be the reason of the reduced opacity. Therefore, this situation encourage the writer to extend experiments to higher power conditions with 12 mg injection.

Although SOI had big impact on SOC, change in EGR rate from 15 to 60 slightly affect the SOC as shown in Figure 4.35-b. SOC is approximately 20 CA earlier than the reference engine condition for the whole range of the EGR rate. In-cylinder pressure and temperature around -12 CA created suitable conditions for igniting the air-fuel mixture. Increased EGR reduced the available oxygen in the cylinder and prevented the injected fuel from burning, and postponed the SOC from -13 to -11 CA. Rise in EGR reduced the NO_x as shown in Figure 4.35-c. Although the EGR rate was 30% at reference engine condition, the NO_x emission can be reduced below reference condition by increasing the EGR rate above 45%. Higher NO_x compared to reference condition up to 45 CA can be explained with different SOIs. SOI of the reference condition was 6 CA bTDC, while it was 25 CA for the evaluated case. This was the expected trend for the NO_x emission as EGR reduced both temperature and available oxygen in the combustion. Nitrogen atoms in the air could not react with oxygen atom as the in-cylinder temperature reduced with increasing EGR.

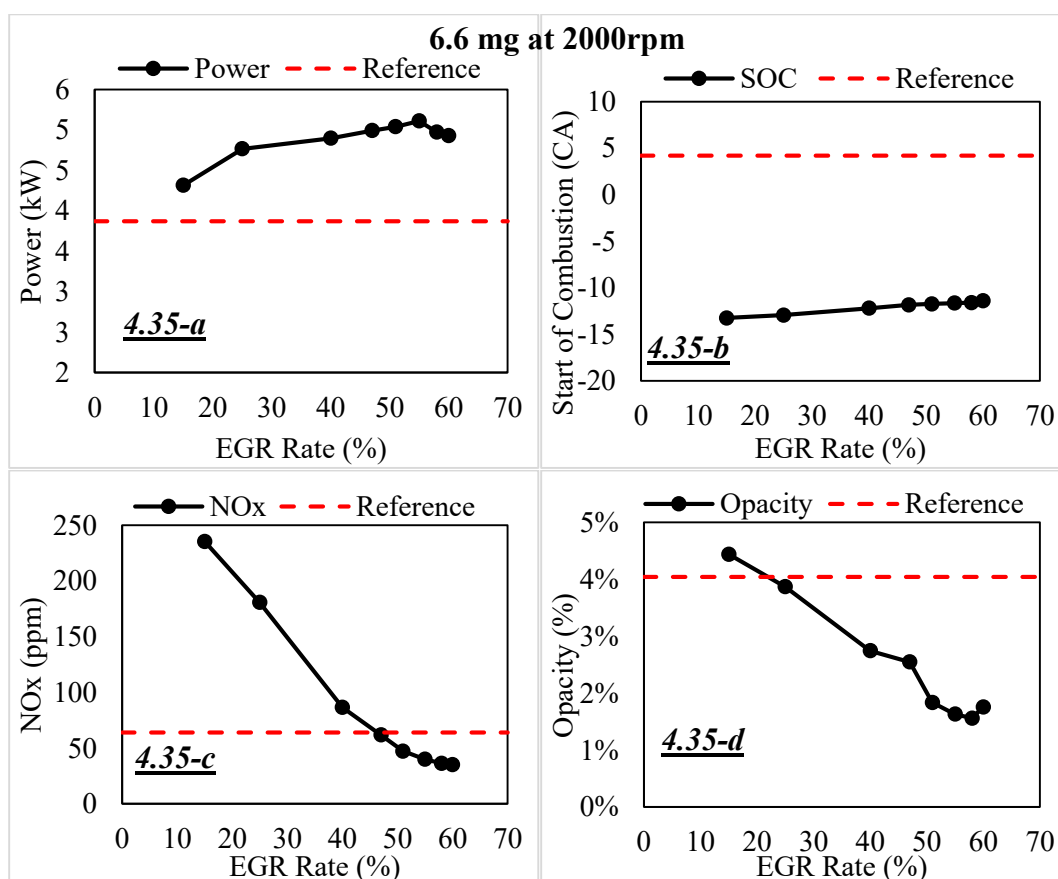


Figure 4.35 : Power change according to EGR at SOI -25 CA.

4.5.1.3 Comparison of the original case with selected optimum case

In order to apply advanced combustion methods and define optimum case, 75 different engine conditions were tested for the single injection of the 6.6 mg/stroke fuel at 2000 rpm as shown in Figure 4.36. Results of the SOI and EGR adjustments were analyzed in the previous sections. Similarly, results of all experiments were compared with each other and optimum engine configuration was selected among them to compare with original engine operation point.

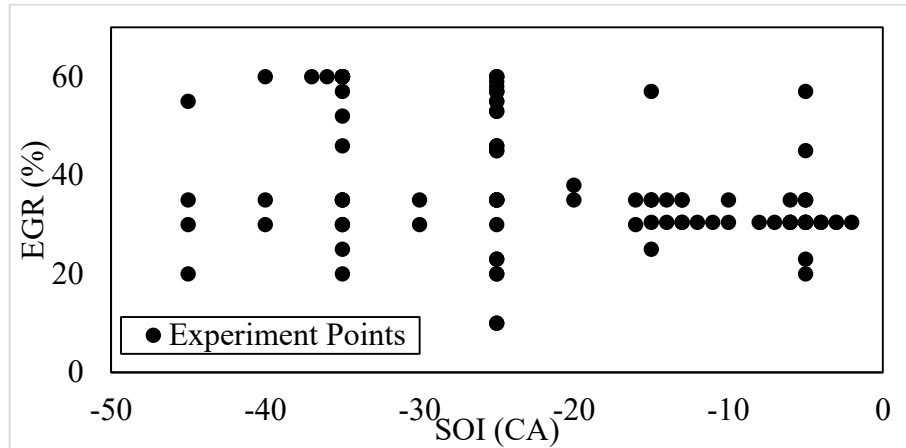


Figure 4.36 : Experiment points of single injection of 6.6 mg/stroke fuel.

Among performed 75 experiments, the optimum case was defined as the case where the highest reduction of the NO_x and opacity emissions were observed while there was an increase at power. Table 4.6 shows the parameters of the original (reference) and defined optimum engine operation point. EGR was increased from 30 to 55 percent and SOI was advanced from -6 to -25 CA to get more homogenous combustion to achieve low NO_x and soot, without increasing fuel consumption. The optimum case is an example of the PCI method that aim to achieve premixed combustion. This method has SOC just before the TDC where most of the study in the literature aim to achieve SOC just after TDC in PCI methods.

Table 4.6 : Engine parameters of the original case and optimum case.

Name of The Case	EGR rate	SOI CA
Original Case	30	-6.0
Optimum Case	55	-25.0

In-cylinder pressure, temperature and ROHR comparison of these points can be seen in Figure 4.37. Maximum in-cylinder pressure increased from 42 to 65 bar and peak

value of ROHR was higher than original case. Besides, the highest value of the average in-cylinder temperature increased from 1180 to 1280 K. The local high temperature zones in the combustion chamber may be the reason of the higher NO_x of the original case compared to optimized case. High EGR rate prevented local high temperature zones and low temperature combustion can be achieved at optimized case.

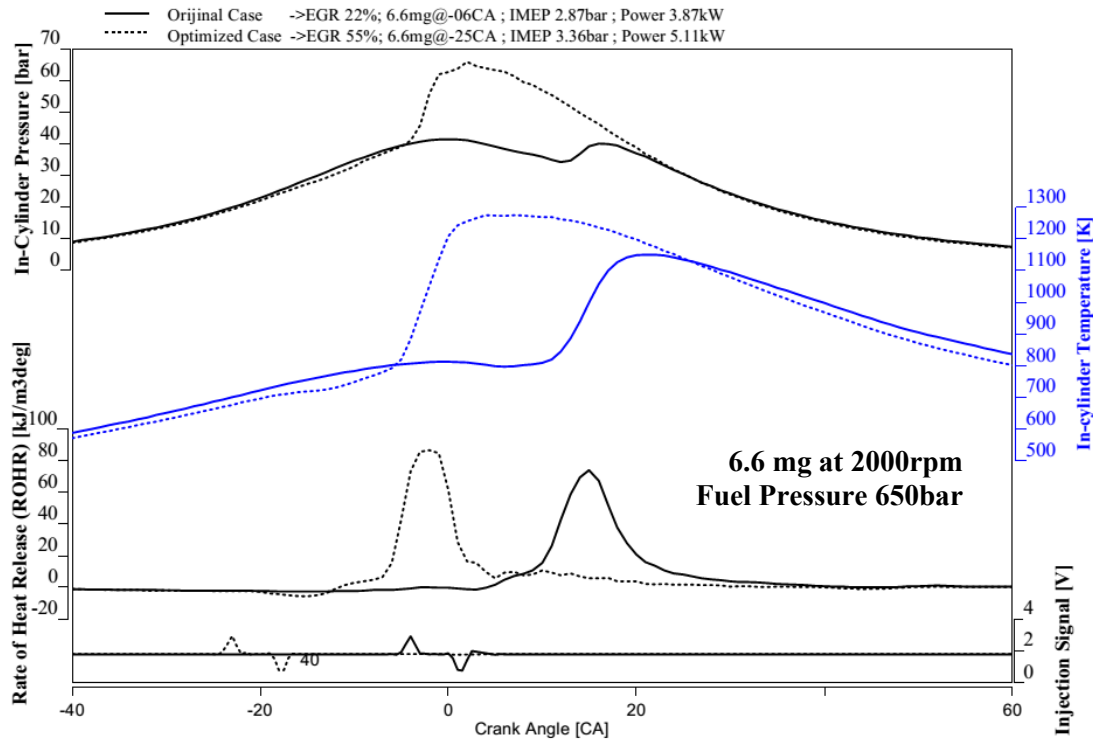


Figure 4.37 : Combustion data of original and optimized case.

Power, NO_x and Opacity measurements of the reference engine conditions were normalized to 100 to be able to plot increase/decrease percentage in one graph. As shown in Figure 4.38, power was improved 32% with new setup while reducing the NO_x and opacity emissions 38% and 60% respectively.

Therefore, it is concluded that combination of HCCI and LTC methods, which is named as premixed combustion (PCI), has potential to improve the performance and emission of the diesel engine. Moreover, it was found that utilization of EGR might have positive impact on power in contrary to its conventional use.

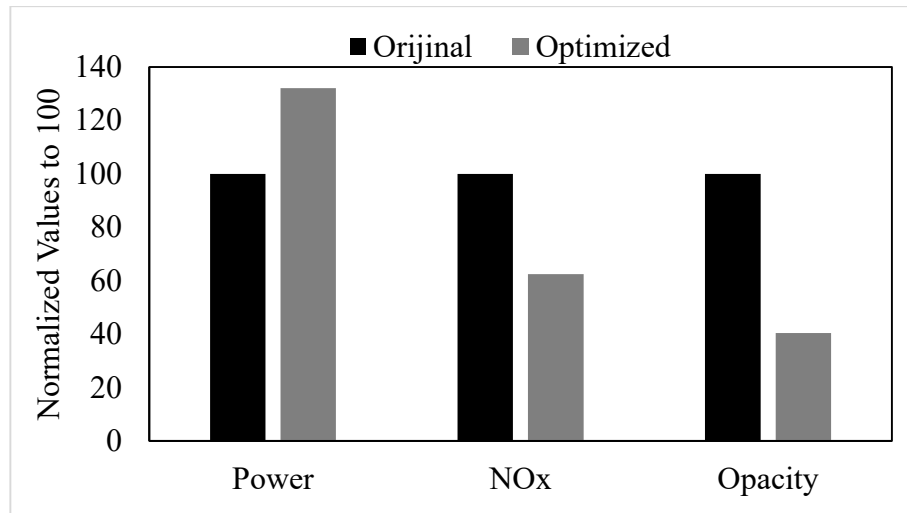


Figure 4.38 : Comparison of the original and optimized case.

Table 4.7 also shows the values of the power, NO_x and opacity that compared in Figure 4.38.

Table 4.7 : Results of the original and optimized case.

Name of The Case	Power	NO _x	Opacity
Unit	kW	ppm	%
Original Case	3.87	63.8	4.0
Optimum Case	5.11	39.9	1.6

4.5.1.4 Comparison of the B00 and B30 for single injection configuration

Advanced combustion method with single injection configuration was also studied with biodiesel-blended fuel to investigate the improvement potentials by utilizing biodiesel. B30 was utilized for the experiments. SOI of the main injection that had the same quantity (6.6 mg) at B00 experiments was adjusted from -6 to -40 CA as shown in Figure 4.39. Blue and black lines show the results of B30 and B00, respectively. Dotted red line show the results of the original working condition of the engine for single injection configuration as mentioned in previous section. EGR rate was 30% for all test conditions; reference, B00 and B30 cases. Effect of SOI on power, SOC, NO_x and opacity was plotted in Figure 4.39-a,b,c,d respectively.

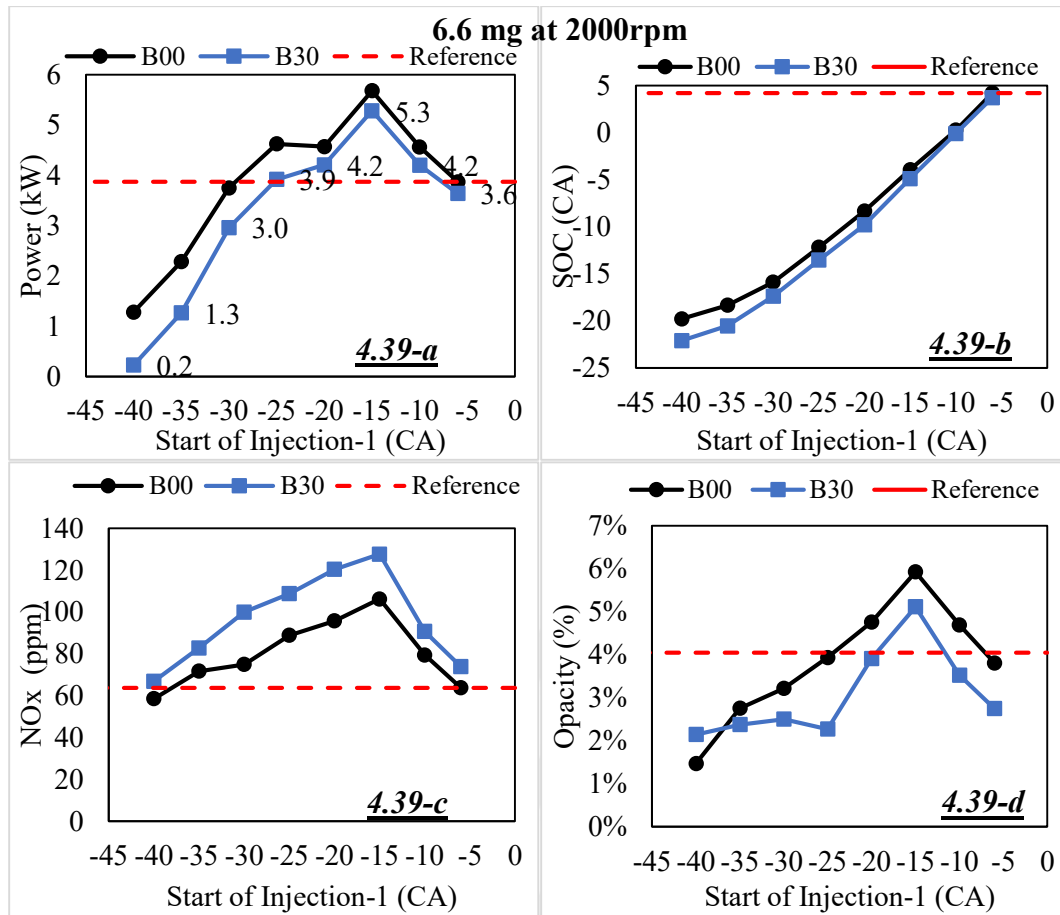


Figure 4.39 : Comparison of the B00 and B30.

Utilization of biodiesel reduced the power due to its lower GCV like FL and cruise speed experiments. The reduction was around 5% up to SOI 20 CA bTDC. However, injection the fuel at earlier SOIs (-25 to -40 CA) caused more power reduction. Higher oxygen content of the biodiesel enabled injected fuel to ignite earlier compared to diesel as can be seen in Figure 4.39-b. SOC occurred earlier with B30 compared to B00. The SOC difference increased with advancing the SOI. This led more power reduction for the cases where SOI was very early, as early combustion of the injected fuel caused more loss during compression stroke of the engine.

The NO_x emission increased and opacity reduced with utilization of biodiesel as shown in Figure 4.39-c and d respectively. The emission trend was same with the FL and cruise speed experiments. Although very early injection of B30 improved the opacity compared to B00, the NO_x emission and power were worse than the reference condition. Therefore, utilization of biodiesel did not improve the focused engine properties (power and NO_x) under advanced combustion methods due to its lower GCV, higher oxygen content and earlier SOC compared to diesel.

Further optimization study was also performed for B30 by modifying the EGR rate and the SOI. 60 different engine conditions were tested for the single injection of the 6.6 mg/stroke with B30 at 2000 rpm as shown in Figure 4.40. Results of the all experiments were investigated; the results of the optimum case was compared with the reference engine condition, to reveal the potential of the optimization.

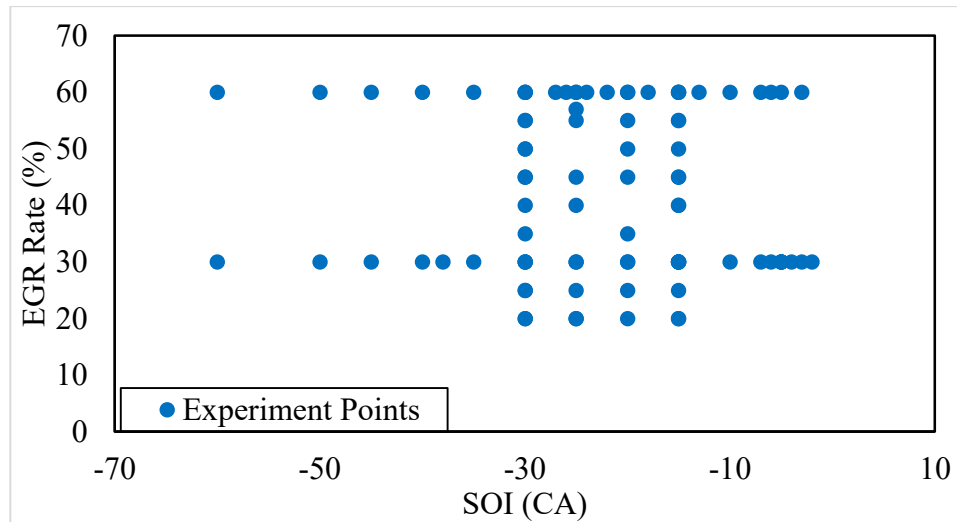


Figure 4.40 : Experiment points of single injection of 6.6 mg/stroke B30.

Similar to the experiments with pure diesel, the optimum case was selected by looking the lowest NO_x and Opacity among the cases where power was increased. Table 4.8 shows the EGR rate and SOI of original and optimized engine parameters.

EGR was increased from 30 to 60 percent and SOI was advanced from -6 to -23 CA to get more homogenous combustion to achieve low NO_x and soot, without increasing fuel consumption. As use of biodiesel increase the oxygen content in the cylinder, very early-injected fuel can start to burn earlier. In order to delay the SOC of B30 and to reduce the pumping loss, more EGR was used and SOC was retarded. Therefore, EGR was increased from 55 to 60 when fuel was changed from B00 to B30, to reduce the oxygen content in the cylinder. Besides, SOI was retarded from 25 to 23 CA bTDC as biodiesel has shorter IDD than diesel.

Table 4.8 : Engine parameters of the original case and optimum case.

Name of The Case	EGR rate	SOI CA
Original Case	30	-6.0
Optimum Case B00	55	-25.0
Optimum Case B30	60	-23.0

Figure 4.40 shows the in-cylinder pressure, temperature and ROHR of the three cases. Value of the maximum in-cylinder pressure of B30 and B00 were same, but it was observed earlier for B30 than B00 as can be seen in Figure 4.40. The peak value of ROHR of B30 was lower than B00 and observed earlier due to earlier SOC of B30 compared to B00.

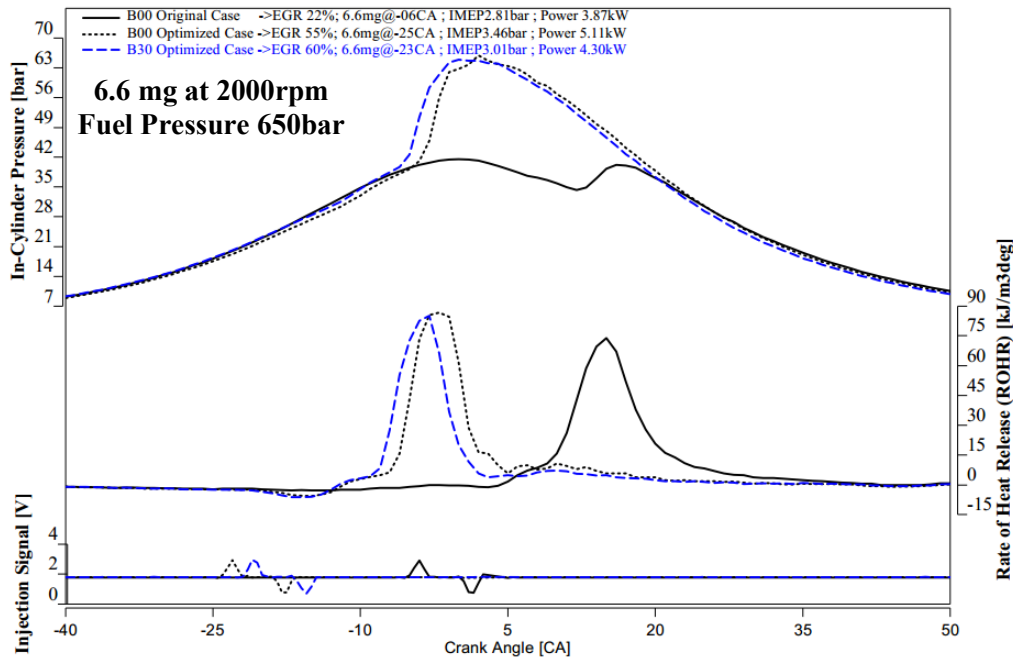


Figure 4.41 : Combustion data of original and optimized case.

Figure 4.41 shows the normalized values of Power, NO_x and Opacity. Power was improved 11% with new configuration of B30 while reducing the NO_x and PN emissions 5% and 90% respectively, as shown in Figure 4.41. Although power could be increased with reducing the NO_x and PN emissions with B30, the improvement at power and NO_x was 20 and 33% lower than B00, respectively.

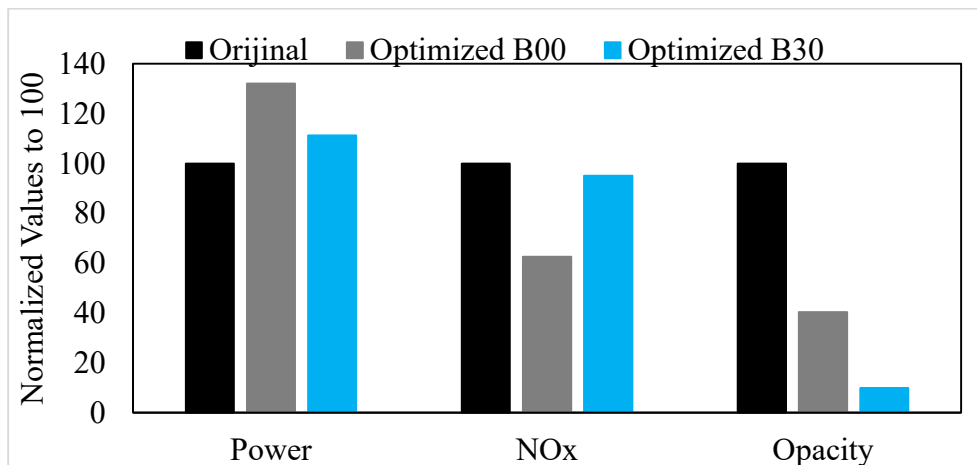


Figure 4.42 : Comparison of the original and optimized case.

4.5.2 Advanced combustion with double injection strategy

Emissions and performance of the vehicle with double injection configuration of 6.6 mg at 2000 rpm is shown in Figure 4.43 with in-cylinder pressure measurement. 6.6 mg fuel was injected at 2000 rpm engine speed. Injection was divided into two parts; as 1.1 mg at -17 CA and 5.5 mg at -1 CA as shown in Figure 4.43. Fuel injection pressure and EGR rate was 620 bar and 25% respectively. Engine power was 3.85 kW under these parameters. 69 ppm NO_x emission and 1.2% opacity was observed at this double injection operation condition.

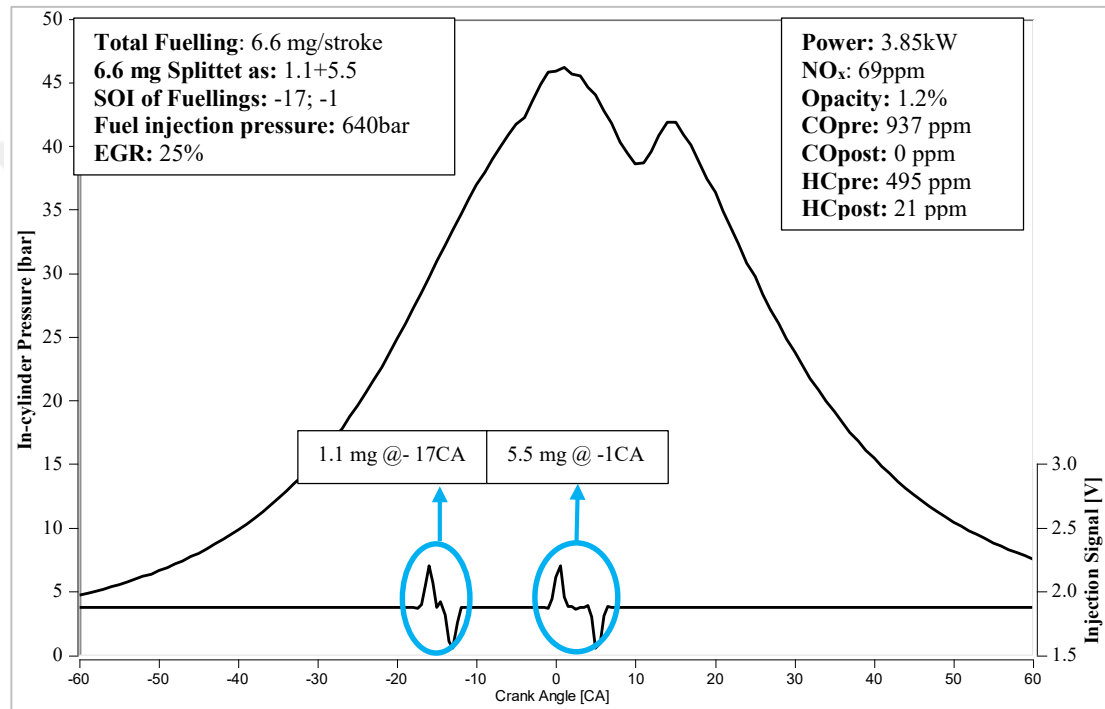


Figure 4.43 : Original engine parameters for double injection configuration.

In order to investigate the change in combustion, quantity, EGR rate and timing of the fuel injection parameters were modified. In total, four parameters were modified as shown in Figure 4.44;

1. SOI of the main injection was modified from 8 CA aTDC to 8CA bTDC. Duration between two injections were kept at least 10 CA to properly control solenoid valve of the injector.
2. SOI of the pilot injection was modified from -10 to -70 CA. Quantity of the pilot injection increased up to 3.8 mg/stroke (half of the injection), while keeping the total injection constant.

3. Quantity of the pilot injection (injection-2) and main injection (injection-1) were modified while keeping the total injection quantity constant as 6.6 mg/stroke.
4. EGR ratio was modified from 20 to 60 % to investigate the effect of EGR on performance and emissions. Figure 4.44 summarizes the modification range of the parameters.

There are more than 10,000 possibilities for engine operation condition for the above-mentioned adjustment range. As the aim of the study is to investigate the potential of the more homogenous combustion under low temperature, only one of the parameter was adjusted while keeping the rest same. In this way, instead of testing a huge number of engine condition, 113 different conditions were tested to investigate the double injection case of 6.6 mg at 2000 rpm. As plotting all of the condition cause repetition of the similar outcomes, results of the selected adjustments and conditions are presented in this section. The parameters that have a significant effect on the results will be explained in detail in following sub-sections.

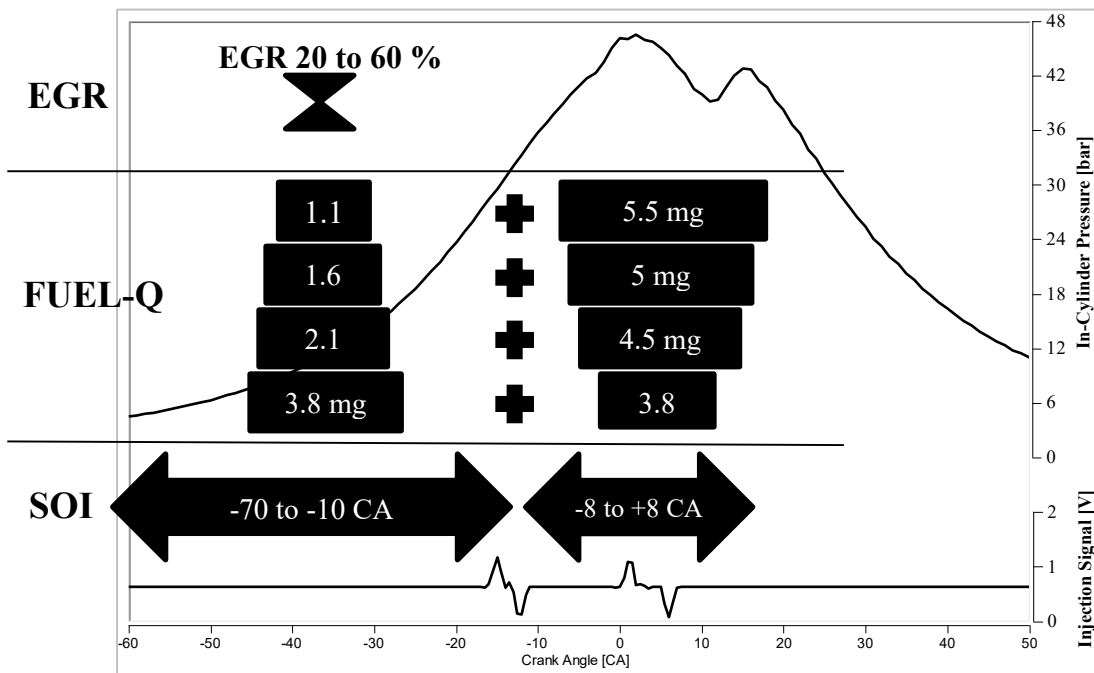


Figure 4.44 : Modified engine parameters and adjustment limits of the parameters.

4.5.2.1 Adjustment of the SOI of injection-2

First, adjustment of the start timing of pilot injection was performed. Figure 4.45 shows the changes at in-cylinder pressure and ROHR for the SOIs of pilot injection at -15, -30 and -60 CA. As seen from in-cylinder pressure, injected main fuel almost did

not burn when pilot injection was advanced from -15 to -60 CA. Moreover, heat release was postponed to aTDC and the highest release rate decreased with advancing the SOI.

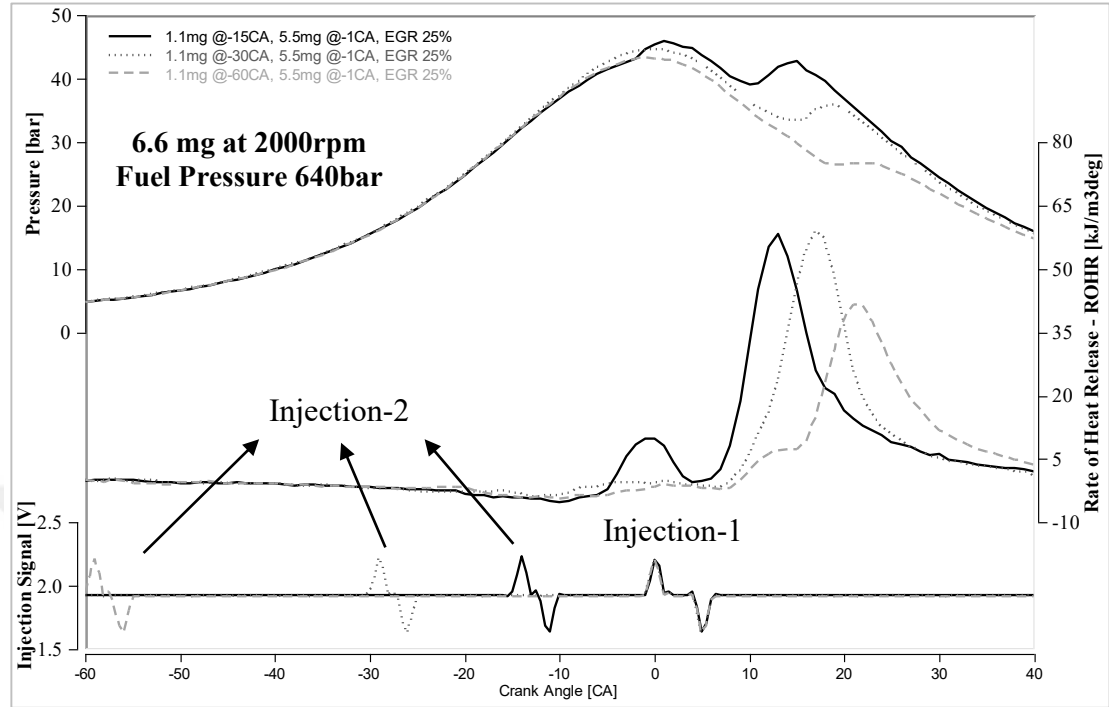


Figure 4.45 : Combustion data of pilot injection adjustments at main SOI -1 CA.

Effect of SOI of the pilot injection on power, SOC, NO_x and opacity was plotted in Figure 4.46-a,b,c,d, respectively. Dotted red line in the plots shows the values of the original condition of the engine that was also shown in Figure 4.43. As these are the original output of the engine, reference label was utilized in the plots.

There are 8 different pilot SOI condition in Figure 4.46-a,b,c,d (-10,-15,-20,-30,-40,-50,-60,-70) to realize the trend in the change of power, SOC, NO_x and opacity.

A continuous power decrease was observed with advancing SOI of 1.1mg fuel injection as can be seen in Figure 4.46-a. Although 17% of the fuel was injected earlier; the effect on power is more than 70%.

Observed high power reduction may be related with the necessary mixture formation duration to ignite the charge. Because early injection of the pilot fuel generate lower heat in the chamber as can be seen in Figure 4.45. Injection of the main fuel to the less heat released chamber need more time to start diffusion combustion. As combustion of the main injection postponed to expansion cycle, lower in-cylinder pressure was observed that result in power reduction.

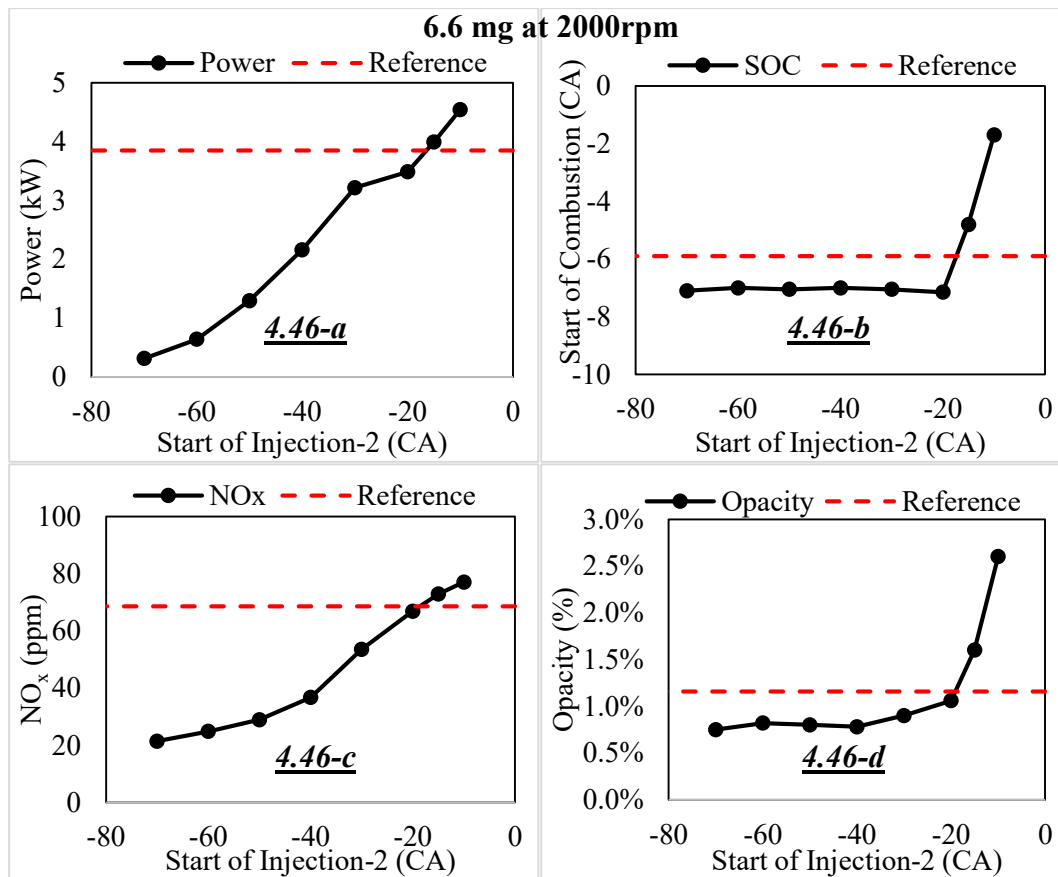


Figure 4.46 : Power, SOC, NO_x and Opacity change according to pilot SOI (case-1).

Figure 4.46-b shows the SOC change according to the alteration of pilot SOI that has quantity of 1.1mg/stroke. SOI of the pilot injection had an impact of SOC from -10 to -20 CA however injecting the pilot injection earlier than 20 CA bTDC had no effect on SOC as can be seen in Figure 4.46-b.

NO_x emission was reduced continuously with advancing the SOI as can be seen in Figure 4.46-c. Reduced in-cylinder temperature and oxygen content inside the combustion chamber prevented the N₂ to react with O₂ and caused reduction in NO_x emissions. Opacity emission was also reduced with advancing the SOI of pilot injection. SOC and Opacity had similar trend. The reduced opacity may be correlated with the more homogenous mixture preparation before SOC.

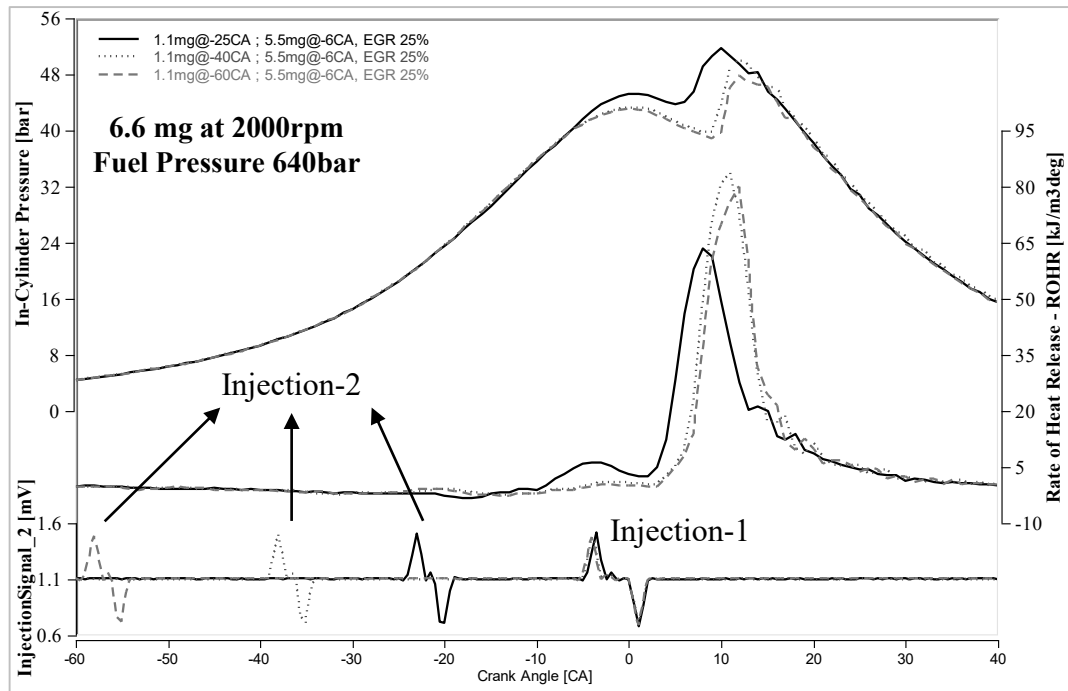


Figure 4.47 : Combustion data of the adjustment of pilot injections while main injection was at -6CA.

The second pilot injection SOI adjustment was performed while keeping the SOI of main injection at -6 CA instead of -1 CA as shown in Figure 4.47. The main injection was performed in the compression stroke, so it can be burned more efficiently compared to the case that main injection was at -1 CA.

Effect of SOI of the pilot injection, while keeping the SOI of main injection at -6 CA instead of -1 CA, on power, SOC, NO_x and opacity was plotted in Figure 4.48. Dotted red line in the plots shows the values of the original condition of the engine that was also shown in Figure 4.43. As these are the original output of the engine, reference label was utilized in the plots.

There are 7 different pilot SOI conditions in Figure 4.48-a,b,c,d (-20,-25,-30,-40,-50,-60,-70) to realize the trend in the change of power, SOC, NO_x and opacity

In the second case, power reduction was also observed however, magnitude of the reduction was reduced from 70% to 15%. As the main injection was performed at -6 CA instead of -1 CA, charge can be prepared that can ignite just before TDC. This reduced the impact of pilot injection on power as can be seen in Figure 4.48-a.

Figure 4.48-b shows the change in SOC according to SOI of the pilot injection where main injection was performed at -6 CA. SOI of the pilot injection had an impact on SOC from -20 to -40 CA. However injecting the 1.1 mg earlier than 50 CA bTDC had

almost no effect on SOC as can be seen in Figure 4.48-b. SOC of the SOIs earlier than 40 CA bTDC were around -3 CA, which were caused by the main injection at -6 CA.

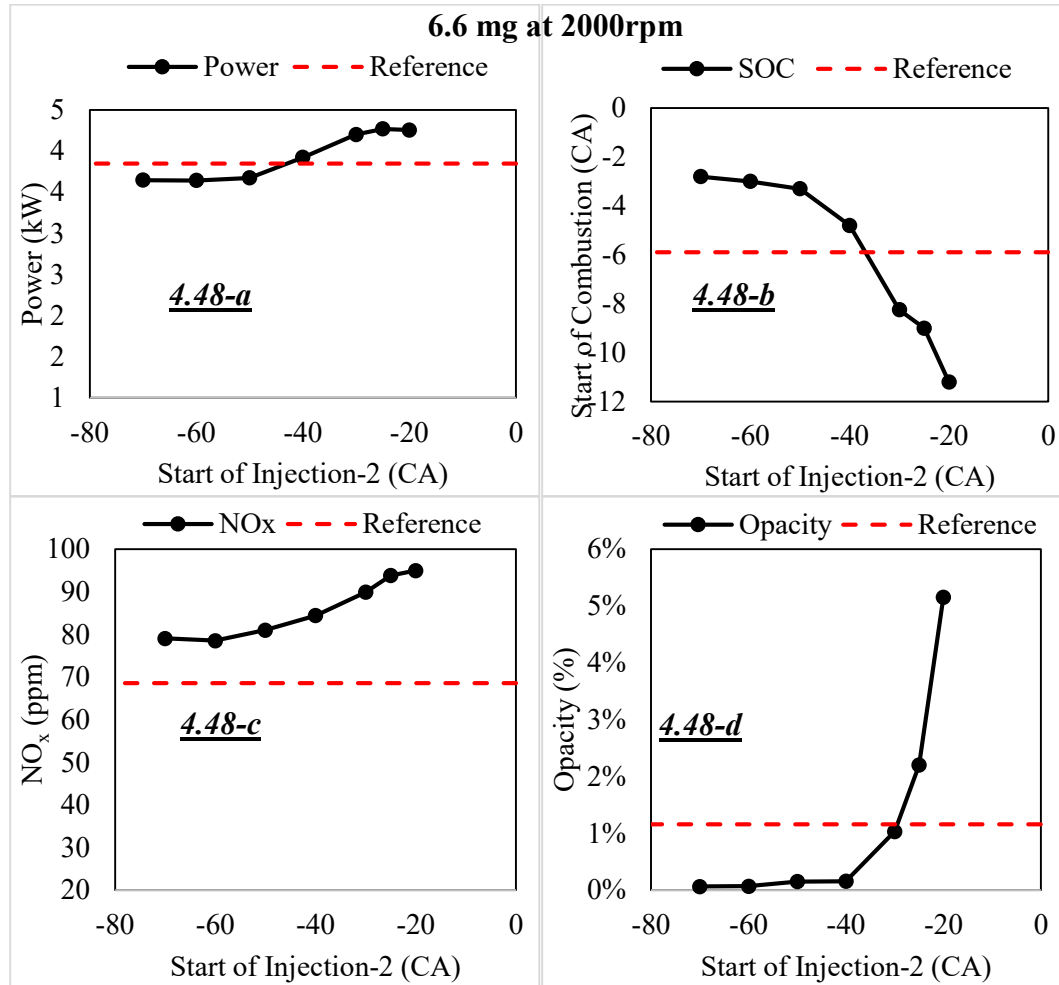


Figure 4.48 : Power, SOC, NO_x and Opacity change according to pilot SOI (case-2).

NO_x emission was reduced continuously with advancing the SOI as can be seen in Figure 4.48-c. However, observed NO_x emission was higher than reference condition. The rapid rise of the in-cylinder pressure just after TDC may lead local high temperature zones that may be the reason of the higher NO_x emission compared to original case.

Opacity emission can be reduced by advancing the pilot SOI. There was one heat release peak at very early injected pilot fuel as can be seen in Figure 4.47, therefore pilot injection was burned at the same time with main injection. Very early injected pilot fuel led more homogenous charge during the combustion, which might be the reason of the reduced opacity.

4.5.2.2 Adjustment of the quantities of the injections

Double injection strategy had one pilot injection and main injection. The pilot injection was performed in an amount of 1.1 mg at -17 CA and main injection was carried out in an amount of 5.5 mg at -1 CA, while engine running under original configured parameter set. Total injection quantity was kept same in all cases. Pilot injection was increased from 1.1 mg to 3.5 mg, while reducing the main injection 5.5 mg to 3.1 mg. The EGR rate, SOI of pilot and main injection were kept as 40%, -6 and -40 CA respectively, which was also indicated at the legend of Figure 4.49.

Figure 4.49 shows the changes at in-cylinder pressure and ROHR for the pilot quantities 1.1, 2.6 and 3.5 mg per stroke. Zoomed section of the injection signal can be also seen in the plot to show the injection duration of different quantities. Green arrows show the start of injection (SOI) and end of injection (EOI) of the 5.5 mg main injection. Injectors have solenoid valve and stay open during the CA between SOI and EOI. Beginning of the first triangle is the SOI and end of the second reversed triangle is the EOI.

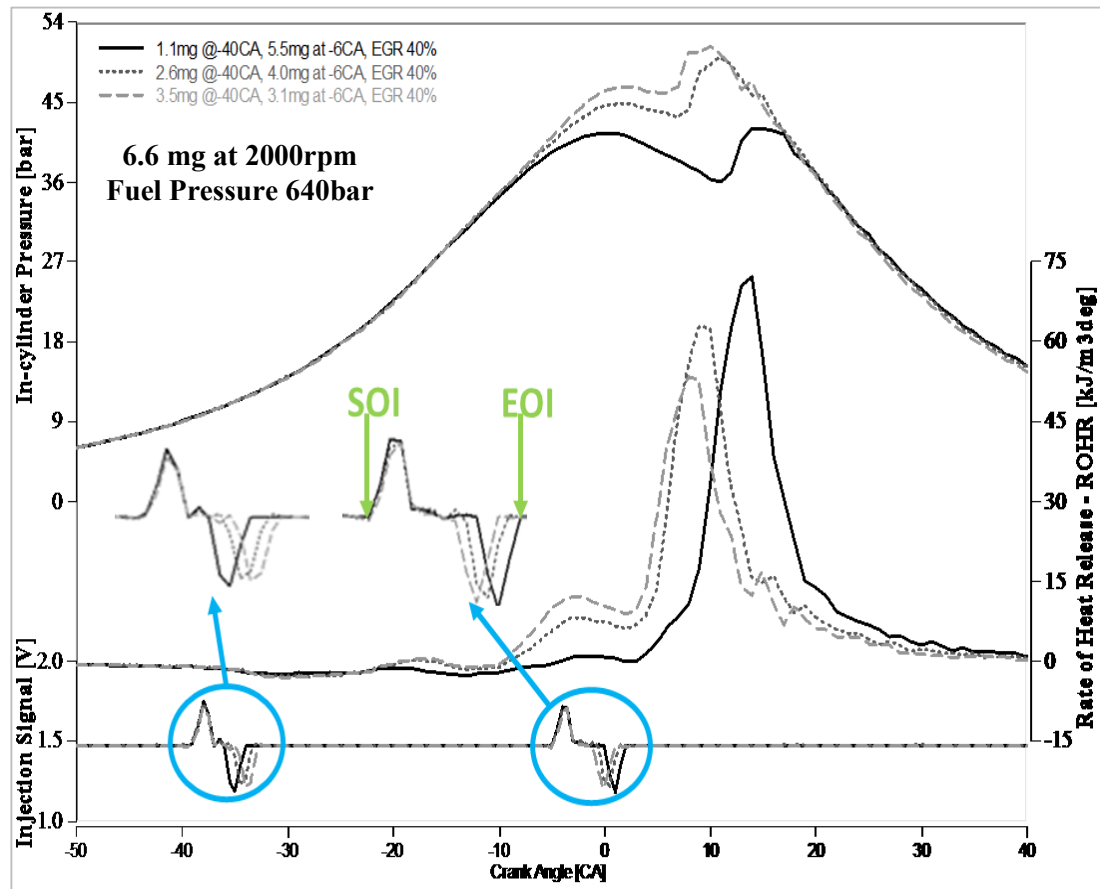


Figure 4.49 : Combustion data of different quantities of injections.

Increasing the pilot injection quantity led more heat release during the compression stroke as can be seen in Figure 4.49. However, main injection started to burn just after TDC, which might improve the observed power.

Effect of the injection quantities on power, SOC, NO_x and opacity was plotted in Figure 4.50. Quantity of the pilot and main injection were adjusted at 2000 rpm while keeping the total injection quantity 6.6 mg per cycle. Dotted red line in the plots shows the values of the original condition of the engine (1.1 mg at 17 CA bTDC, 5.5 mg at 1 CA bTDC) that was also shown in Figure 4.43. As these are the original output of the engine, reference label was utilized in the plots.

There are 6 different pilot injection quantity conditions in Figure 4.50-a,b,c,d (1.1, 1.6, 2.1, 2.6, 3.1, 3.5) to realize the trend in the change of power, SOC, NO_x and opacity. There were two trends in the power as can be seen in Figure 4.50-a. First, increasing the quantity of pilot injection (up to 2.6 mg) led to power increase. However, increasing the injection quantity more than 2.6 mg caused a decrease at the power. The use of higher EGR compared to original case was the reason of the lower power than original case.

The ROHR curve of the investigated cases indicated that the heat release of 2.6 mg pilot injection has the optimum shape according to other alternatives since in the 1.1 mg option, heat release occurred very late in the expansion cycle and in the 3.5 mg option heat release occurred very early in the compression cycle. However, power of the reference case was higher than the investigated cases. The use of higher EGR compared to original case was the reason of the lower power than original case.

Figure 4.50-b shows the change of SOC according to the quantity of the pilot injection. The SOC occurred earlier when the pilot injection quantity was high. Because oxygen can react easier with fuel when there is more fuel in the combustion chamber.

The NO_x emission had similar trend with power however, change in NO_x emission was limited, as can be seen Figure 4.50-c. Observed lower NO_x compared to original case can be explained with the high EGR, as the EGR rate was 40%, which is almost twice of the original case.

Opacity emission was also lower than the original case as can be seen in Figure 4.50-d. Because the early injected pilot and main injection enabled more homogenous charge in the chamber. Observed opacity was reduced continuously with increasing

the quantity of the early injected (-40CA) fuel, as more fuel had longer duration to mix with the air in the combustion chamber.

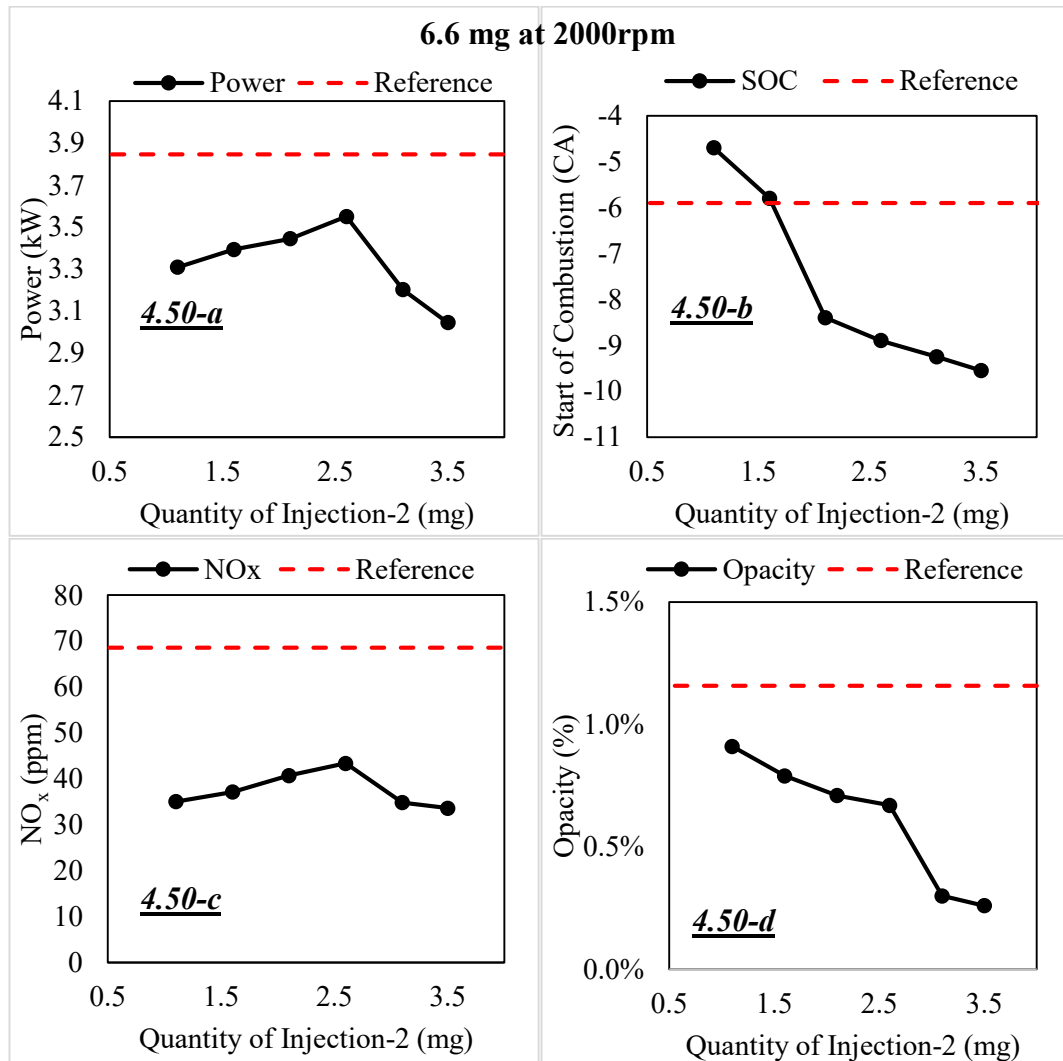


Figure 4.50 : Power, SOC, NO_x and Opacity change according to pilot quantity.

4.5.2.3 Adjustment of the EGR rate

The EGR rate adjustment were performed to investigate the effect of EGR on emissions and performance. As shown in Figure 4.51, 1.1 mg of the fuel was injected at 40 CA bTDC and the rest (5.5 mg) was injected at 6 CA bTDC. As expected lower in-cylinder pressure was observed with the rise of EGR and heat release occurred later in the expansion cycle with lower peak as shown in Figure 4.51.

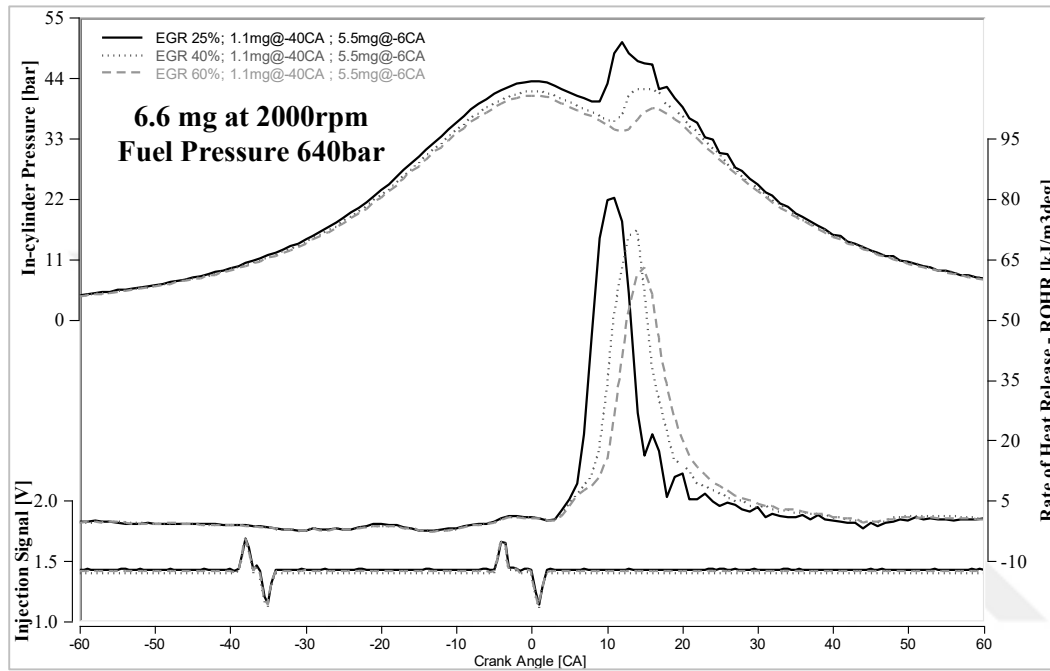


Figure 4.51 : Combustion data of EGR at 22%, 40% and 60%.

Effect of the EGR rate on power, SOC, NO_x and opacity was plotted in Figure 4.52. Dotted red line in the plots shows the values of the original condition of the engine that was also shown in Figure 4.43. As these are the original output of the engine, reference label was utilized in the plots.

There are 5 different EGR rate conditions in Figure 4.52-a,b,c,d (22, 30, 40, 50, 60) to realize the trend in the change of power, SOC, NO_x and opacity.

Rise in EGR caused reduction in power as shown in Figure 4.52-a. As EGR reduced the available oxygen in the combustion chamber, injected fuel could not totally react with oxygen to generate heat and power.

Rise in EGR did not have an impact on the burn duration as shown in Figure 4.52-b. SOC was around 5 CA bTDC and high EGR in the combustion chamber prevent the combustion of the pilot injection. Main injection initiated the combustion, as SOC was

later than the SOI of main injection. Although SOC was same for all conditions, combustion ended later when EGR was higher as can be understood from the ROHR data in Figure 4.51.

On the other hand, EGR is one of the best method to lower the NO_x emission. Figure 4.52-c shows the NO_x reduction with the rise of EGR. As temperature and available oxygen in the combustion chamber reduced with increasing EGR, NO_x formation rate reduced.

Opacity emission was low under all conditions as shown in Figure 4.52-d. Rise in EGR did not increase the opacity, most probably due to achieving homogenous combustion under all EGR cases.

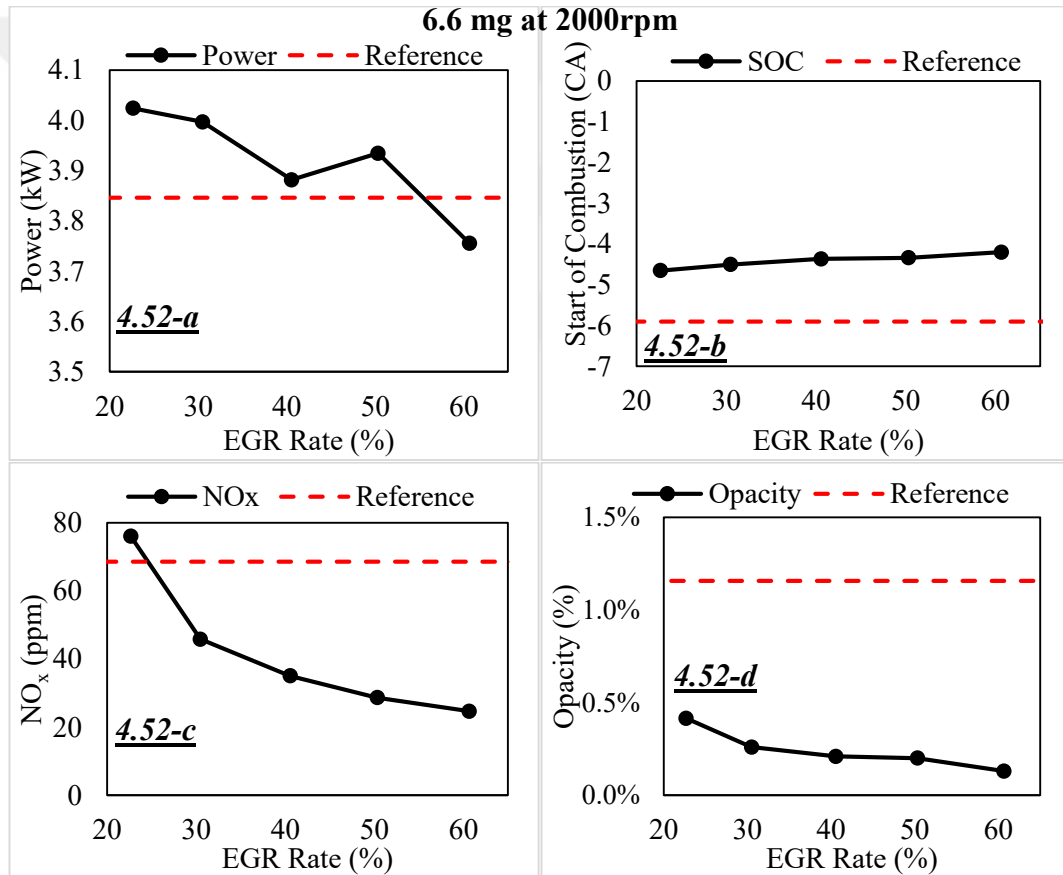


Figure 4.52 : Power, SOC, NO_x and Opacity change according to EGR.

4.5.2.4 Comparison of the original case with selected optimum case

In order to apply advanced combustion methods and define optimum case, 113 different engine conditions were tested for the double injection of the 6.6 mg/stroke fuel at 2000 rpm as shown in Figure 4.43. Results of the adjustments of pilot SOI, pilot quantity and EGR were investigated in the previous sections. Similarly, results of the

all experiments were compared with each other to select the optimum configuration. Later optimum engine condition was compared with the reference engine condition to reveal the variation. Figure 4.53 shows the experiment points of three parameters. Although five parameters were modified, as it is not possible to show them all in one graph, Figure 4.53 was plotted to indicate the range of the parameters. EGR position was shown with grey bubble. X-axis and Y-axis of the figure shows the SOI of main pilot injection respectively. Greater circle of the EGR means that the EGR rate is higher for that SOI combination. For example, at main SOI -4 CA and pilot SOI -40 CA two different EGR rates were tested (labelled point with black rectangular). Inner darker grey bubble represent 22% EGR rate, outer lighter grey bubble represents the 60% EGR rate.

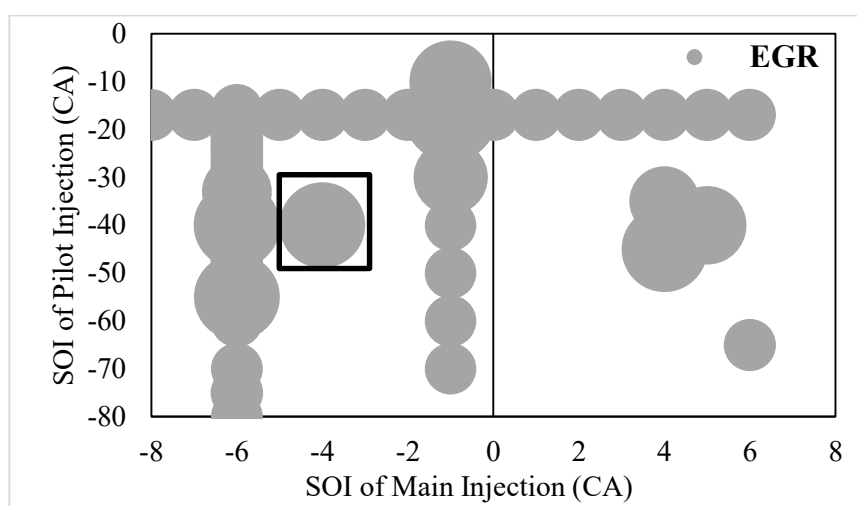


Figure 4.53 : Experiment points of double injection of 6.6 mg/stroke.

Among performed 113 experiments, the optimum case was defined as the case that the highest reduction was observed for both NO_x and opacity while there was an increase at power. Table 4.9 shows the original (reference) and defined optimum engine operation parameters. EGR rate was increased from 22 to 40 percent. SOI and quantity of pilot injection was modified from -17 to -33 CA and 1.1 to 3.0 mg, respectively. SOI and quantity of main injection was modified from -1 to -6 CA and 5.5 to 3.6 mg, respectively. Parameters were modified to achieve low NO_x and soot, without increasing fuel consumption. The proposed combustion is an example of the PCI method, which aim to achieve premixed combustion.

Table 4.9 : Engine parameters of the original case and selected optimum case.

Name of The Case	EGR	Main Quantity	Main SOI	Pilot Quantity	Pilot SOI
Unit	%	mg/stroke	CA	mg/stroke	CA
Original Case	25	5.5	-1.0	1.1	-17.0
Optimum Case	40	3.6	-6.0	3.0	-33.0

In-cylinder pressure, ROHR and injection signal of the original and optimized case can be seen in Figure 4.54. Maximum in-cylinder pressure increased from 46 to 53 bar. As quantity of the pilot injection was increased, ROHR started earlier at optimized case compared to original case. Injected main injection fuel can burn earlier at optimized case than original case.

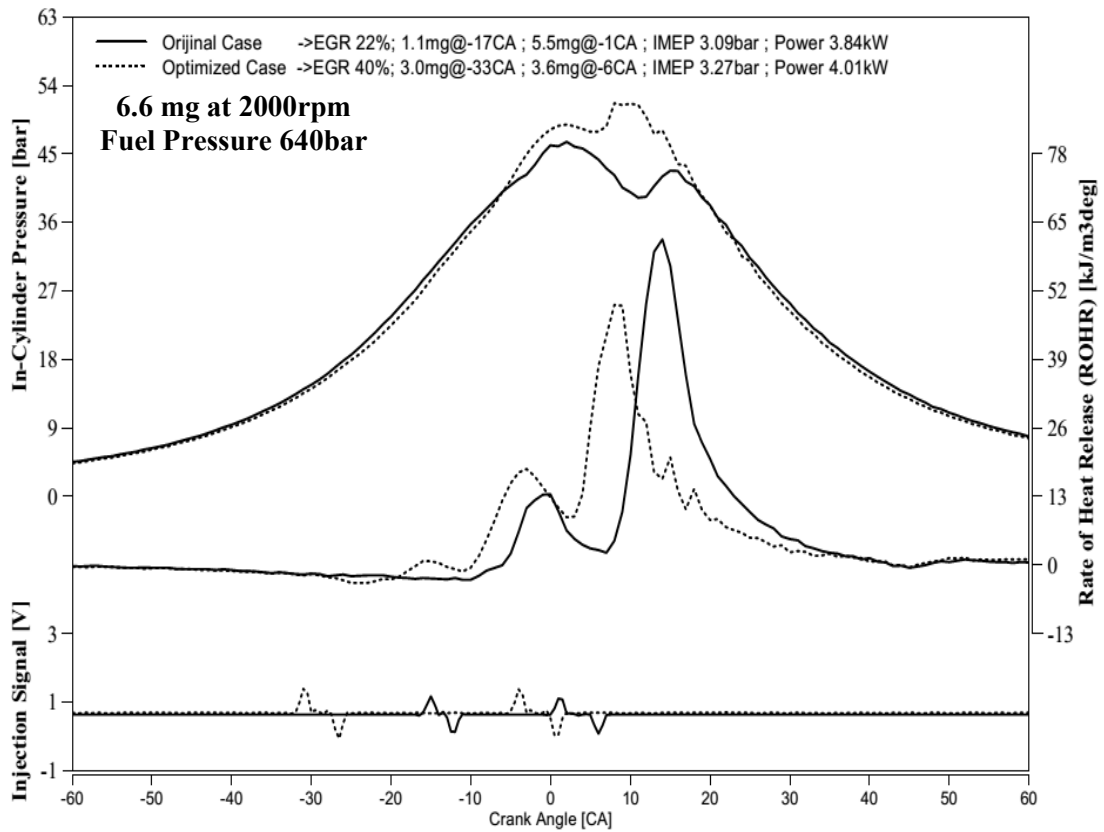


Figure 4.54 : Combustion data of original and optimized case.

Power, NO_x and Opacity measurements of the reference engine conditions were normalized to 100 to able to plot increase/decrease percentage in one graph. As shown in Figure 4.55, power is improved 5% with new setup while reducing the NO_x and PN emissions 38% and 80% respectively.

Therefore, it is concluded that high EGR and earlier pilot injection with higher quantity

compared to original case allow achieving more homogenous mixture before SOC. Combustion mode can be defined as PCI that enables to achieve better NO_x and soot emissions, therefore it has potential to be used for the steady state operation condition of the cycles.

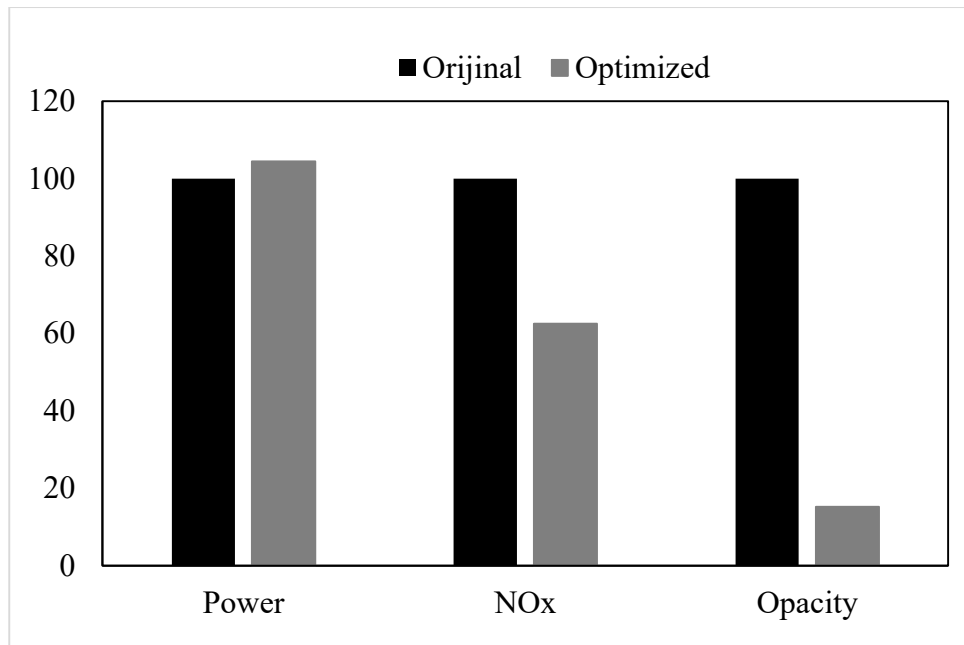


Figure 4.55 : Comparison of the original and optimized case.

Table 4.10 also shows the values of the power, NO_x and opacity that compared in Figure 4.55. As mentioned optimum case improved both NO_x and Opacity while increasing the power for the same amount of injection.

Table 4.10 : Results of the original and optimized case.

Name of The Case	Power	NO _x	Opacity
Unit	kW	ppm	%
Original Case	3.84	62.3	1.9
Optimum Case	4.01	39.0	0.3

4.5.3 Advanced combustion with triple injection strategy

Experiments that were performed with original ECU of the engine without any modification indicate that fuel injected to the cylinder in three parts as shown in Figure 4.56. 6.6 mg injection was divided into 3 parts; as 0.9 mg at -30.3 CA, 1 mg at -17.1 CA and 4.7 mg at 0.2 CA as shown in Figure 4.56. EGR rate was 22 % and the fuel injection pressure was 620 bar. All of the original engine parameters are shown in Figure 4.56 with in-cylinder pressure diagram. Engine power was 2.85 kW under these parameters. 64 ppm NO_x emission and 3.8 % opacity was observed at this triple injection operation condition

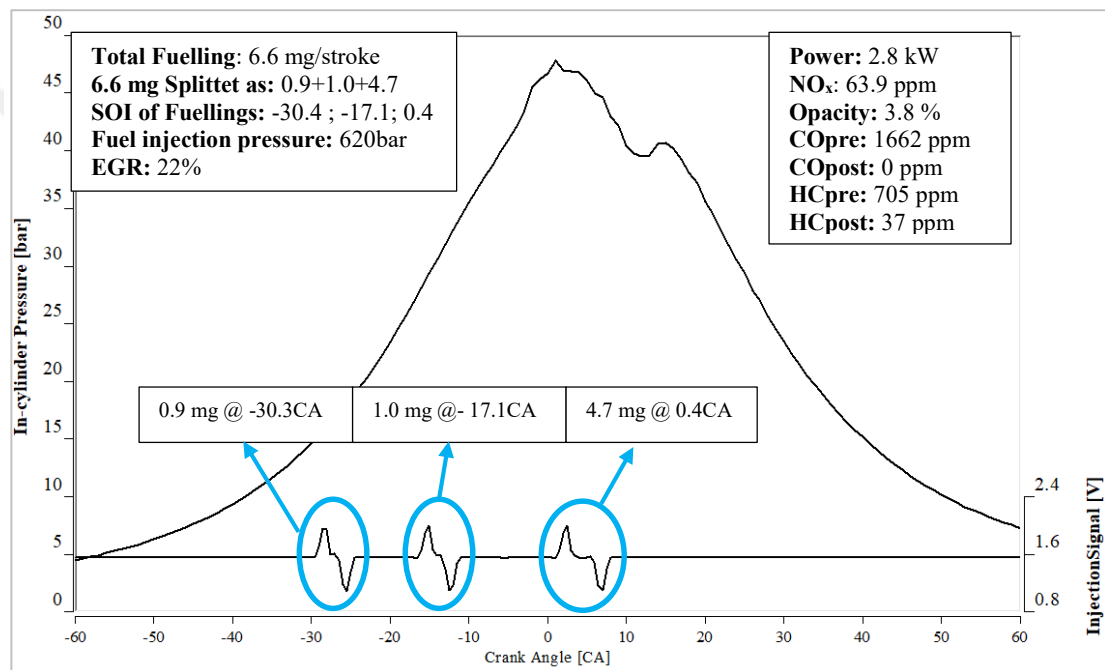


Figure 4.56 : Original engine parameters for triple injection configuration.

In order to investigate the change in combustion, EGR ratio and timing of the fuel injection parameters were modified. In total, four parameters were modified as shown in Figure 4.57.

1. SOI of the main injection (injection-1) was modified from 7 CA aTDC to 9 CA bTDC. Duration between two different injections were kept at least 10 CA to properly control solenoid valve of the injector. Quantity of the main injection was kept 4.7 mg for all cases.
2. SOI of the injection-2 was modified from -10 to -40 CA. Quantity of the injection-2 was kept 1.1 mg for all cases.

3. SOI of the injection-3 (the first injection) was also modified from -30 CA to -90 CA. Quantity of the injection-3 was kept 0.9 mg for all cases.
4. In addition to timing of injections, EGR rate was modified from 20 to 60 % to investigate the effect of EGR on performance and emissions.

There are more than 10,000 operation condition possibilities for the above-mentioned modification range. However, the aim of the study is to investigate the potential of more homogenous combustion under low temperature. Therefore, it is not necessary to test every possible combination. Instead of following a mixed parameter variation with design of experiment method, only one of the parameter was adjusted while keeping the rest same. In this way, effect of each parameter on combustion was defined and testing 174 different conditions were be enough to evaluate the triple injection case of 6.6 mg at 2000 rpm. As plotting all of the condition cause repetition of the similar outcomes, results of the selected adjustments and conditions are presented in this section. The parameters that have a significant effect on the results will be explained in detail in following sub-sections.

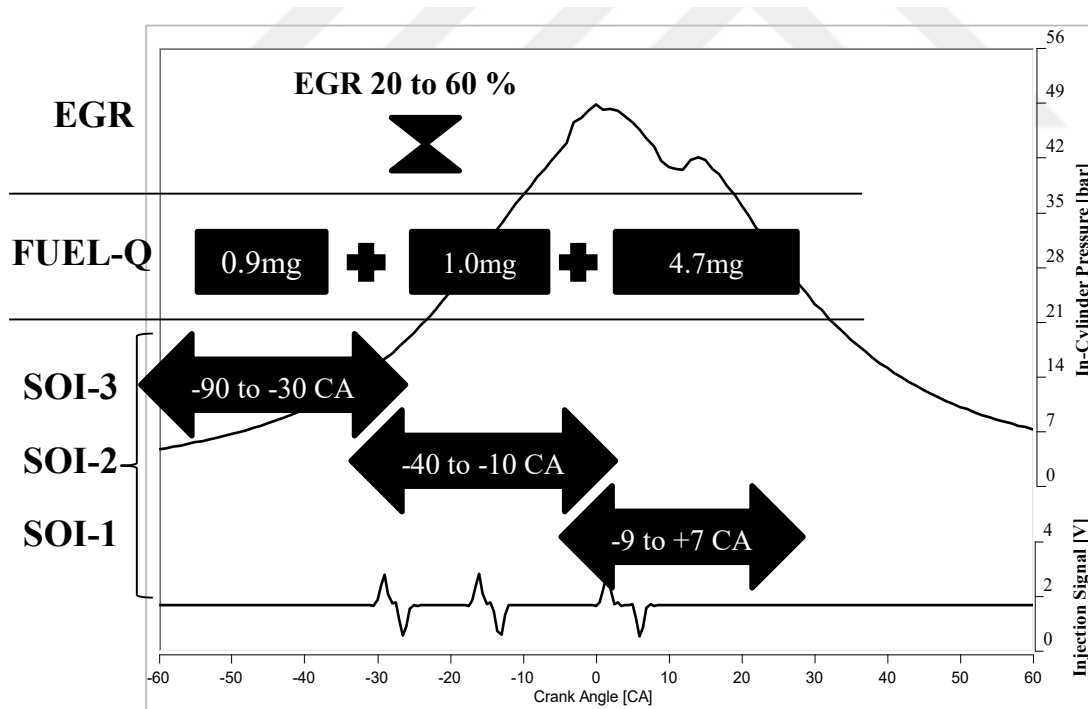


Figure 4.57 : Modified engine parameters and adjustment limits.

4.5.3.1 Adjustment of the injection-1 (main injection)

Adjustment of the start timing of the main injection was performed. Figure 4.58 shows the changes at in-cylinder pressure and ROHR for the SOIs of main injection at -7, 0 and +5 CA. In-cylinder pressure increased with advancing the main injection. Besides, ROHR of the pilot and main injection were combined with advanced main injection.

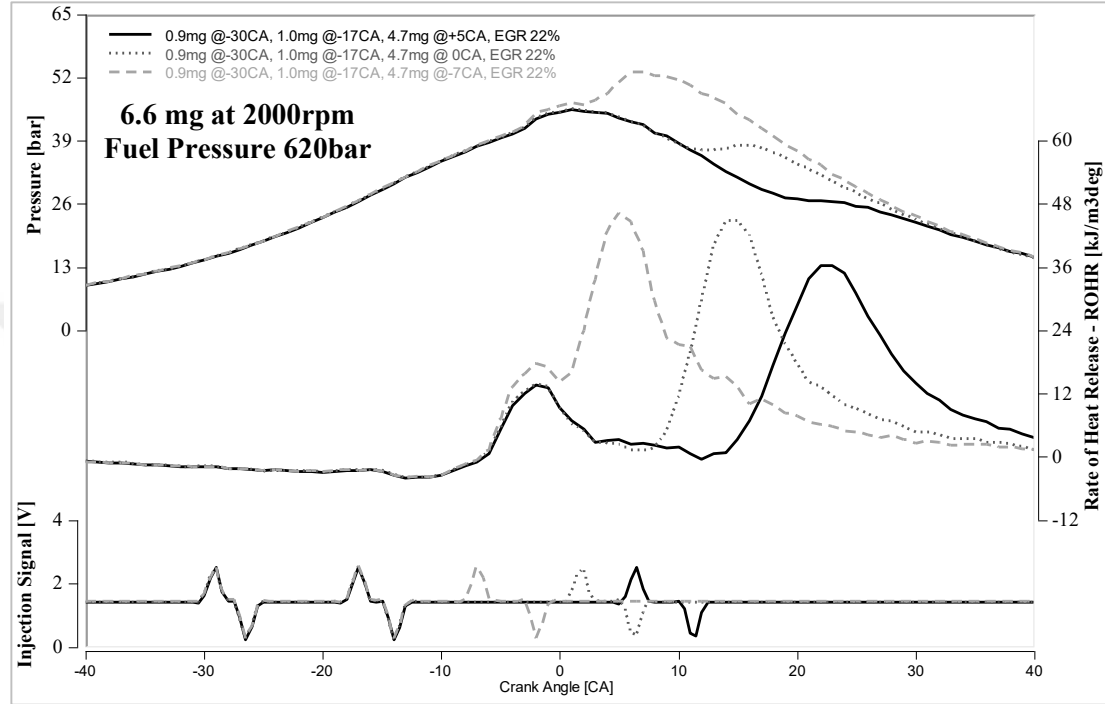


Figure 4.58 : Combustion data of main SOI at -7, 0 and +5 CA.

Effect of SOI of the main injection on power, SOC, NO_x and opacity was plotted in Figure 4.59. Dotted red line in the plots shows the values of the original condition of the engine that was also shown in Figure 4.56. As these are the original output of the engine, reference label was used in the plots.

There are 9 different main SOI condition in Figure 4.59-a,b,c,d (7,5,3,1,0,-1,-3,-5,-7) to realize the trend in the change of power, SOC, NO_x and opacity.

Continuous power increase was observed with advancing SOI of 4.7 mg main fuel injection as can be seen in Figure 4.59-a. Advancing main injection bTDC enabled the start of combustion just after the TDC and the highest in-cylinder pressure was observed between 5-10 CA aTDC. Injecting the main fuel aTDC postponed the ROHR and resulted in lower power and in-cylinder pressure.

Figure 4.59-b shows the SOC change according to the alteration of main SOI that has

quantity of 4.7 mg/stroke. SOC was changed between -6 and -7 CA as can be seen in Figure 4.59-b, therefore adjustment of the SOI of the main injection had limited impact on SOC due to pilot injections.

NO_x emission had similar trend with power as can be seen in Figure 4.59-c. Reduced NO_x can be explained with lower in-cylinder pressure and temperature. On the other hand, high opacity was observed when SOI was earlier than -3 CA. Rapid rise of in-cylinder pressure resulted in high opacity. Studied case had similar trend with conventional diesel combustion. Advancing the SOI led higher power while worsening the NO_x and opacity.

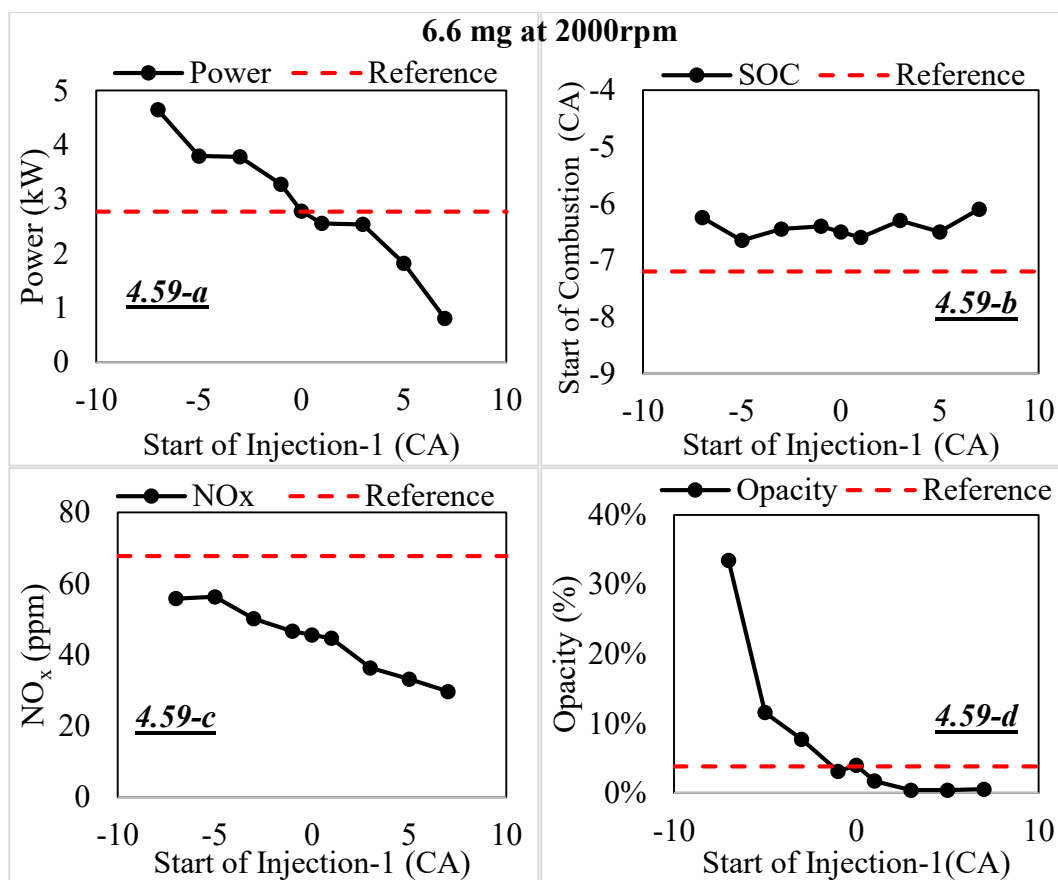


Figure 4.59 : Power change according to SOI of Main Injection.

4.5.3.2 Adjustment of EGR rate

Adjustment of the EGR rate was performed to investigate the effect of EGR rate on combustion. Figure 4.60 shows the changes at in-cylinder pressure and ROHR for the EGR 20, 40 and 60 % with original triple injection configuration. In-cylinder pressure decreased with increasing EGR rate. Besides, ROHR were also decreased and postponed due to reduced lambda with increasing EGR rate.

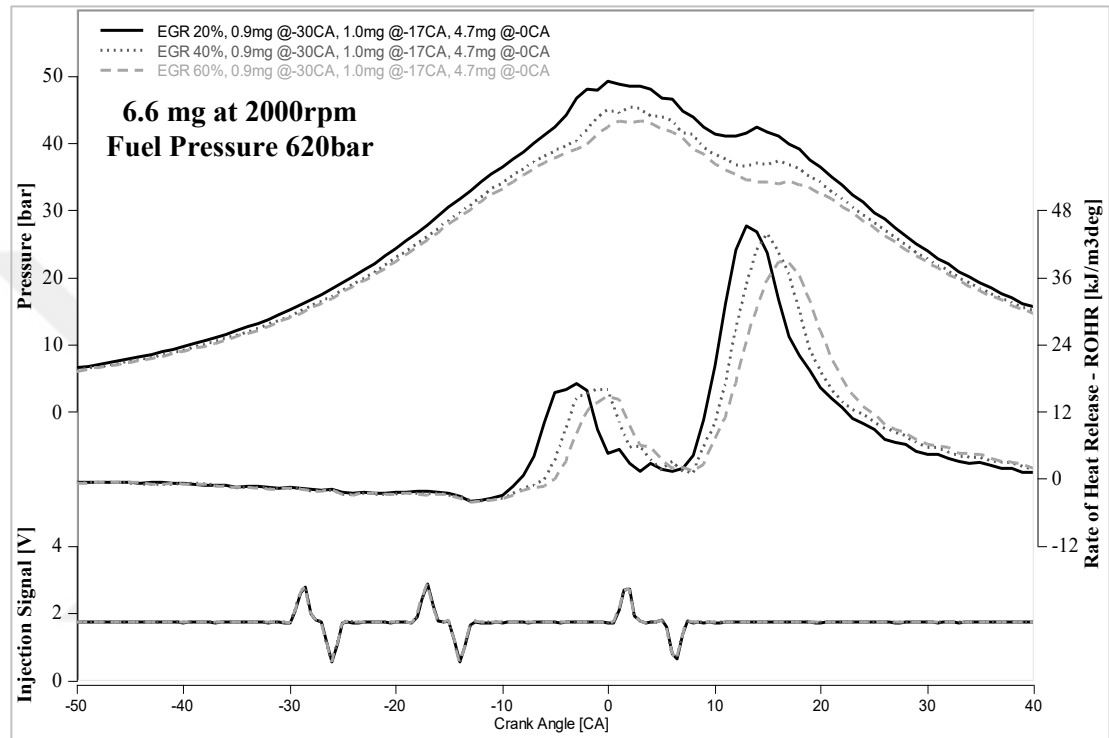


Figure 4.60 : Combustion data of EGR at 20%, 40% and 60%.

Effect of the EGR rate on power, SOC, NO_x and opacity was plotted in Figure 4.61-a,b,c,d. Dotted red line in the plots shows the values of the original condition of the engine that was also shown in Figure 4.56. As these are the original output of the engine, reference label was utilized in the plots.

There are 8 different EGR rate conditions in Figure 4.61-a, b, c, d (10, 17, 25, 40, 47, 51, 55, 60) to realize the trend in the change of power, SOC, NO_x and opacity.

Continuous power reduction was observed with increasing EGR as shown in Figure 4.61-a. Additionally, SOC was delayed with increasing EGR rate. EGR lowered the oxygen content in the combustion chamber and caused longer ignition delay duration. This led delayed SOC and ROHR. Postponing the combustion to the expansion cycle reduced the in-cylinder pressure and power.

NO_x emission reduced up to 22 ppm, which is 65 % lower than the reference condition when the rate of EGR increased from 22 to 60 %. However, opacity emission was increased from 3.8 to 35.2 %. Reduced oxygen content lower the in-cylinder temperature and increase the formation of opacity and lower the formation of NO_x. Therefore, reduced oxygen in the combustion chamber is the reason of the rise in opacity and decrease in NO_x emission.

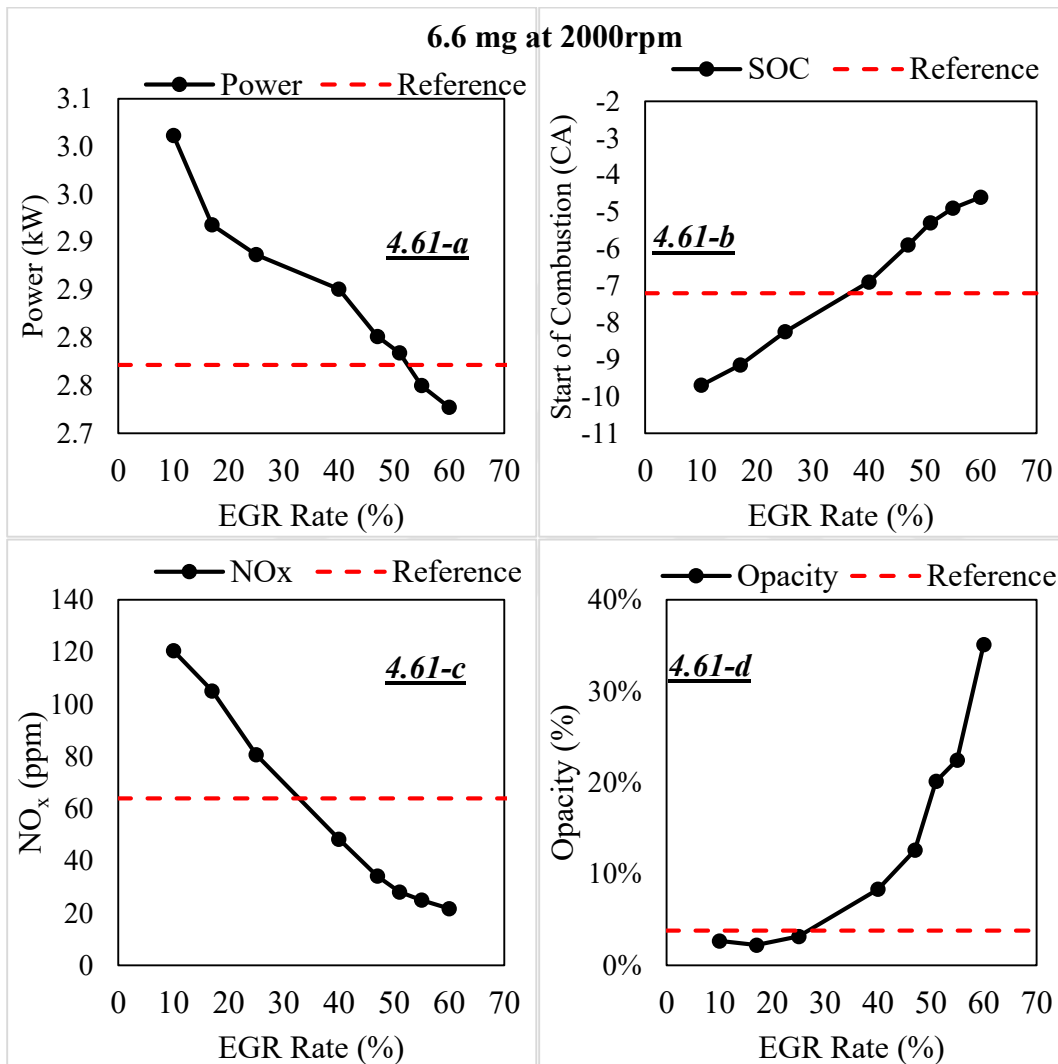


Figure 4.61 : Power change according to EGR Rate.

4.5.3.3 Comparison of the original case with selected optimum case

174 different conditions were tested for triple injection case of 6.6 mg at 2050 rpm to investigate the alternative combustions that has different injection configurations. Figure 4.62 shows the experiment point of three parameters. Indeed four parameters were modified for the triple injection configuration experiments, however it is not possible to show them all in one graph, Figure 4.62 was plotted to indicate the range

of the parameters. EGR position was shown with grey bubble. X-axis and Y-axis of the figure shows the SOI of main and pilot injection respectively. Greater circle of the EGR means that the EGR rate is higher for that SOI combination. For example, at main SOI 0 CA and pilot SOI -70 CA two different EGR rates were tested (labelled points with black rectangular). Inner darker grey bubble represent 22% EGR rate, outer lighter grey bubble represents the 60% EGR rate.

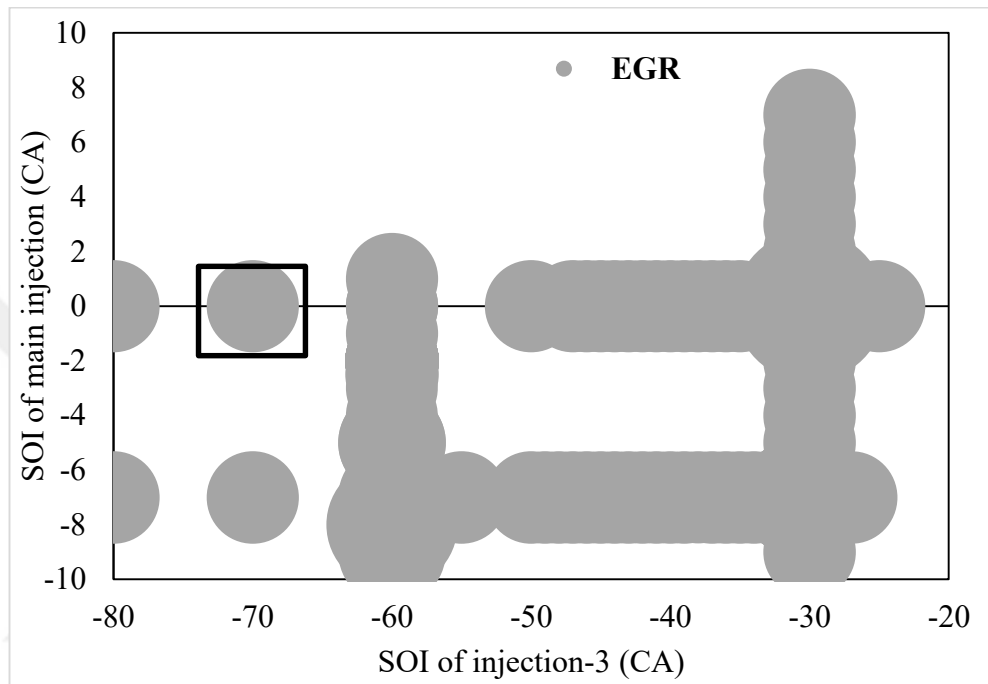


Figure 4.62 : Experiment points of triple injection of 6.6mg/stroke.

Among all conducted 174 experiments, optimum case was defined. Table 4.11 shows the original (reference) and defined optimum parameters of the engine. EGR was increased from 25 to 60 percent. SOI of the injections were modified from 0, -17 and -30 to -8, -30 and -60 CA, respectively. EGR rate and SOI of the injections were modified to achieve more homogenous mixture in the chamber before the SOC to reduce NO_x and soot, without increasing fuel consumption. The proposed combustion is an example of the PCI method, which aim to achieve premixed combustion.

Table 4.11 : Engine parameters of the original case and optimum case.

Name of The Case	EGR %	SOI-1 CA	SOI-2 CA	SOI-3 CA
Original Case	22	0.1	-17.0	-30.0
Optimum Case	60	-8.0	-30.0	-60.0

In-cylinder pressure, ROHR and injection signal of the original and optimum case can be seen in Figure 4.63. While magnitude of the maximum in-cylinder pressure

remained similar, position of the maximum in-cylinder pressure was postponed from 0 to 10 CA aTDC. ROHR started later at optimized case compared to original case. Injected main injection fuel can burn faster at optimized case than original case. There is only one ROHR peak at the optimized case; this showed that all fuel burned at pre-mixed combustion phase. All of the fuel was injected before the SOC at the selected optimum case, whereas main injection was conducted after the combustion of pilot injections at the original case. Therefore optimized case can be represent as an example of premixed compression ignited combustion (PCI).

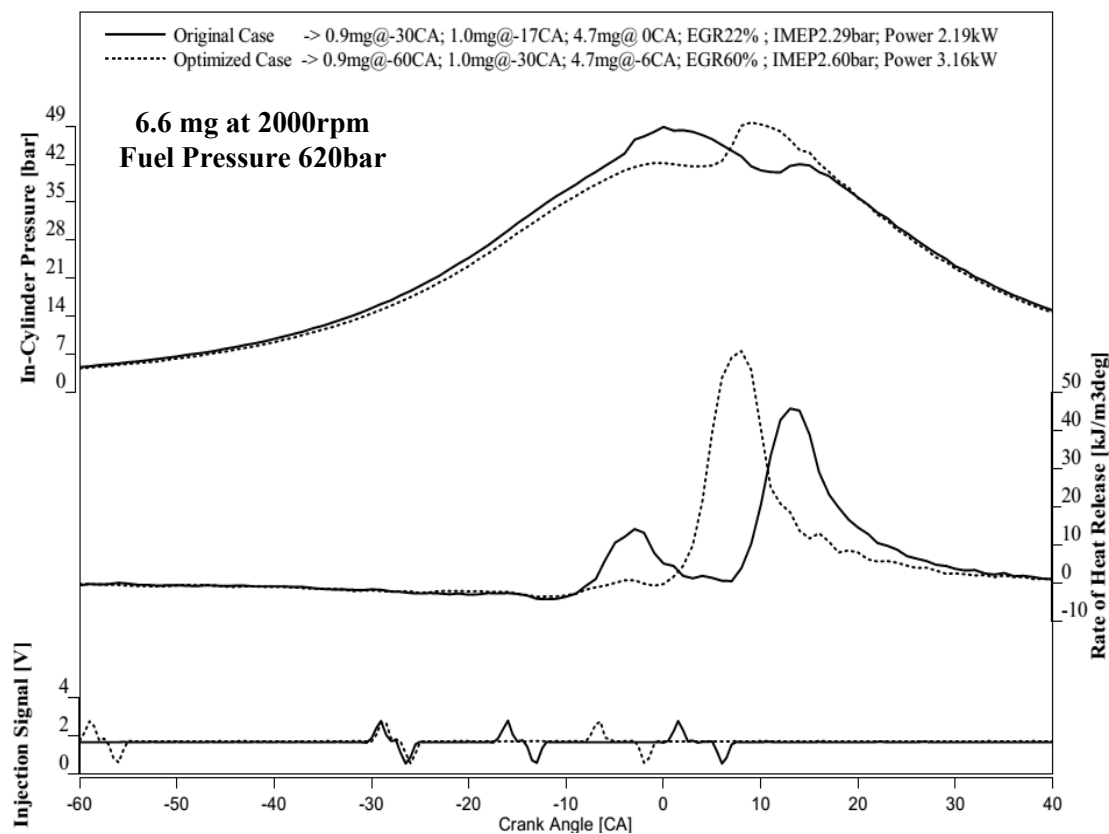


Figure 4.63 : Combustion data of original and selected case.

Power, NO_x and Opacity measurements of the original engine condition were normalized to 100 to able to plot increase/decrease percentage in one graph. As shown in Figure 4.64, the NO_x and opacity emissions can be reduced 35% and 25% respectively with new engine setup, moreover 40% increase was observed at power. Use of high EGR rate and earlier pilot injections compared to original case allow achieving more homogenous mixture before SOC. The high EGR reduced the oxygen content in the combustion chamber and injected pilot fuel could not ignite. Pilot injections was vaporized in the chamber and mix with the air in the combustion chamber up to SOC after injection of main fuel. The optimized case has later SOC and

longer ignition delay duration compared to original case, which improved the homogeneity of the fuel and air mixture in the chamber. Combustion mode can be defined as PCI that enable to achieve better NO_x and soot, therefore has potential to be used for the steady state operation condition of the cycles.

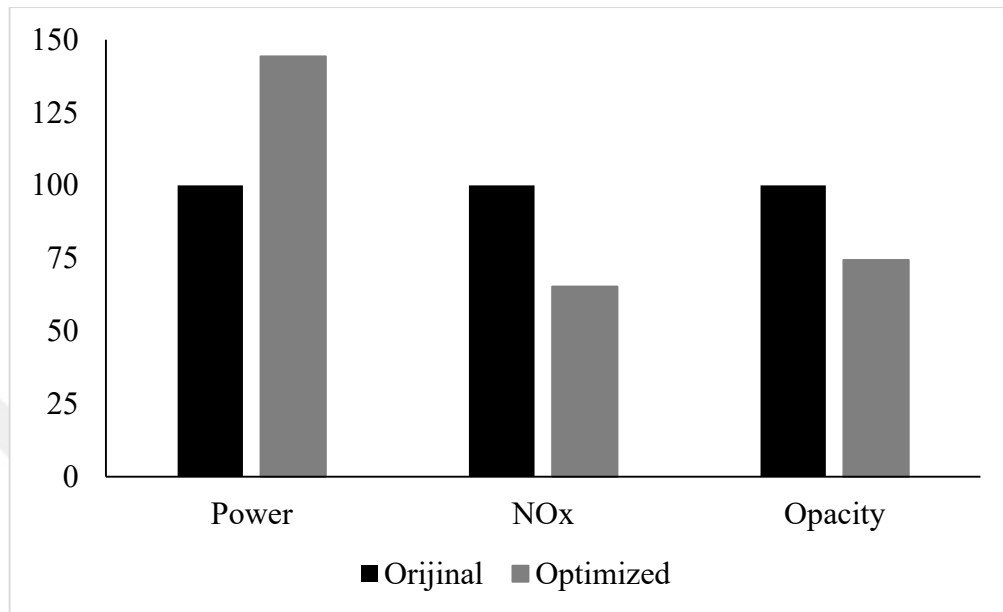


Figure 4.64 : Comparison of the Power, NO_x and Opacity of original and selected case.

Table 4.12 also shows the values of the power, NO_x and opacity that compared in Figure 4.64. As mentioned optimum case improved both NO_x and Opacity while increasing the power for the same amount of injection.

Table 4.12 : Results of the original and optimized case.

Name of The Case	Power	NO _x	Opacity
Unit	kW	ppm	%
Original Case	2.19	63.5	3.7
Optimum Case	3.16	41.4	2.8

4.5.4 Advanced combustion method at medium load condition

After completing optimization study for different injection configurations at the part load condition (6.6 mg), a higher load condition (12 mg) was studied experimentally in engine dynamometer to reveal the applicability of the proposed configuration and to discuss its potential use for hybrid vehicle application. As single injection configuration enabled to achieve the highest power output, original triple injection configuration of the engine was optimized for single injection configuration.

Original engine parameters of the studied case has an injection quantity of 12 mg/stroke. Total quantity was injected in three different times as; 1.0 mg at -33.5 CA, 1.7 mg at -20 CA and 9.3 mg at -2.5 CA in the original case. EGR rate was 32% and fuel injection pressure was 1000 bar under this condition. The original parameters of the studied case was summarized in Figure 4.65. Engine power was 7.8 kW under these parameters. 43 ppm NO_x emission and 3.3 % opacity was observed at this triple injection operation condition.

In order to modify this condition to advanced combustion method, partially premixed compression ignition method was tried by injecting all quantity in one shot. SOI and EGR was modified in a wide adjustment range, in this way combustion, performance and emission maps could be generated according to the EGR and SOI. This enabled to optimize the diesel combustion for range extender application.

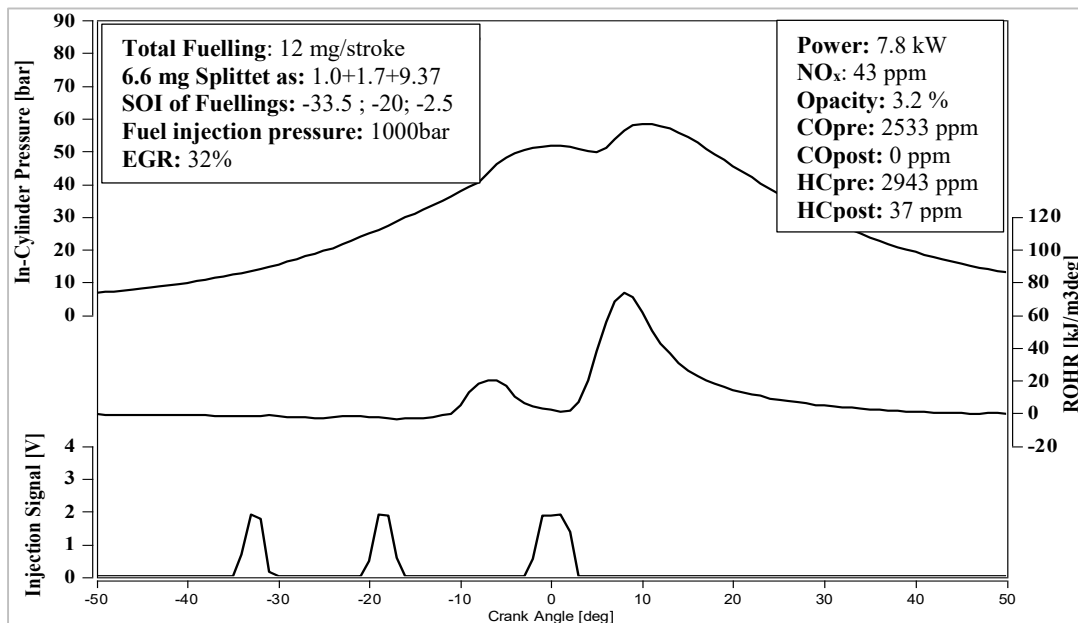


Figure 4.65 : Original engine parameters for 12 mg at 2000 rpm.

4.5.4.1 In-cylinder pressure – heat release rate

Figure 4.66 a,b,c,d show the combustion data of the EGR rate adjustments (10% to 60%) for the SOIs' 10, 20, 30, and 40 CA bTDC, respectively. The darkest black color was utilized to show the highest EGR rate in each SOI plot. The black color were lighted with reducing EGR rate and the lightest gray tone was used for the lowest EGR rate in each plot. In addition, graphs were drawn using dotted and straight lines, respectively, to show the change in each EGR state more clearly.

Injection duration was the same for all cases as can be seen in Figure 4.66 a,b,c,d, as 12 mg/stroke were injected at 1000 bar in all experiments. In this way, effect of EGR rate was monitored for each SOI case.

Maximum in-cylinder pressure and the earliest start of combustion were observed at the lowest EGR rate condition in all SOI cases as can be seen in Figure 4.66 a,b,c,d. Utilization of high EGR rate for SOIs 10 and 20 CA bTDC postpone the rise in ROHR and in-cylinder pressure to the expansion cycle (aTDC). Therefore, combustion was shifted more into the expansion cycle under heavy EGR and engine generated less power compared to low EGR conditions. For example, maximum in-cylinder pressure was 80 bar at 10 CA aTDC under 10% EGR rate whereas it was 50 bar under 60% EGR rate. Moreover, 10% EGR case has higher in-cylinder pressure and ROHR compared to 60 % EGR case during the all expansion cycle, hence power was higher at low EGR condition than high EGR condition for the SOIs up to 30 CA bTDC.

On the other hand, rise in EGR flow increased the power for the SOIs earlier than 30 CA bTDC. Maximum in-cylinder pressure was higher and combustion started earlier for 10% EGR compared to 60% EGR. SOC and maximum in-cylinder pressure was observed in compression cycle for low EGR flow conditions, which resulted in higher in-cylinder pressure during compression stroke. Thus, engine consumed more energy for compressing the more pressurized cylinder. For example, when SOI was 40 CA bTDC (Figure 4.66-d), maximum in-cylinder pressure was 93 bar at 3 CA bTDC under 10 % EGR rate whereas it was 76 bar at 2 CA aTDC under 60 % EGR rate. Increase in EGR slows down the reactions kinetically and gain more time for transferring heat to cooler zones. In this way, local temperature rises can be prevented and soot can be reduced.

Besides, at SOI 40 CA bTDC, 90% mass was burned at 12 CA bTDC under 10% EGR condition, whereas it was burned at 18 CA aTDC under 60% EGR condition. Therefore, the heat released under 10% EGR led more losses compared to 60% EGR during compression stroke for the SOIs later 30 CA bTDC. As a result, rise in torque with increasing EGR was the reason of the postponed combustion to expansion stroke.

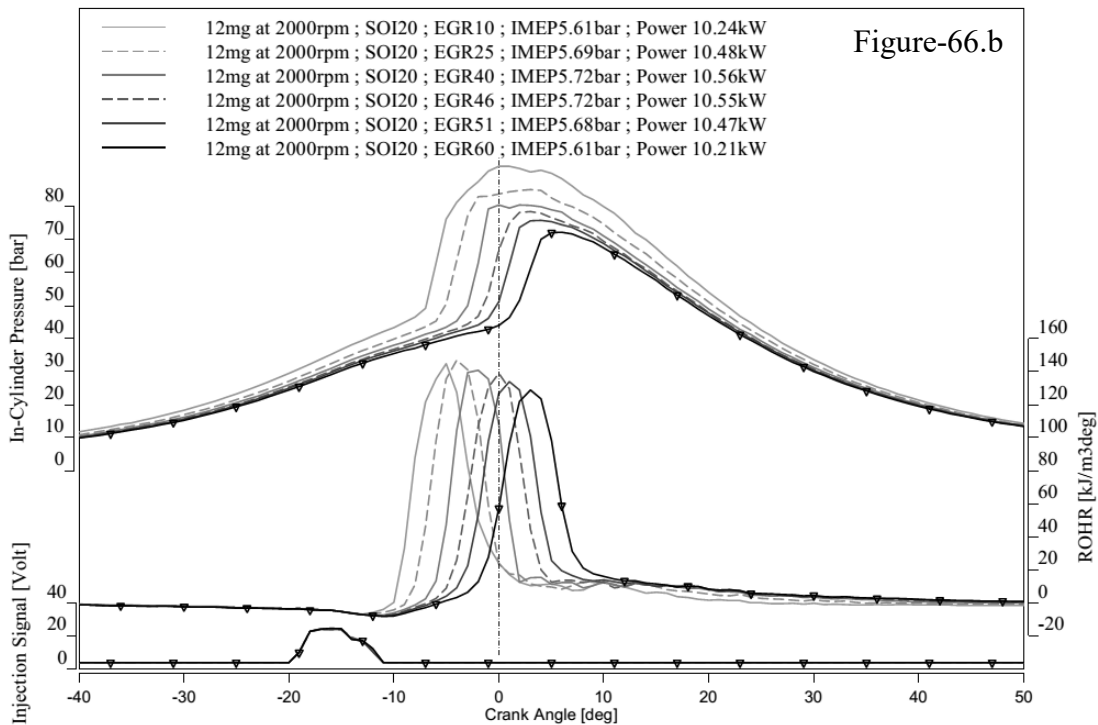
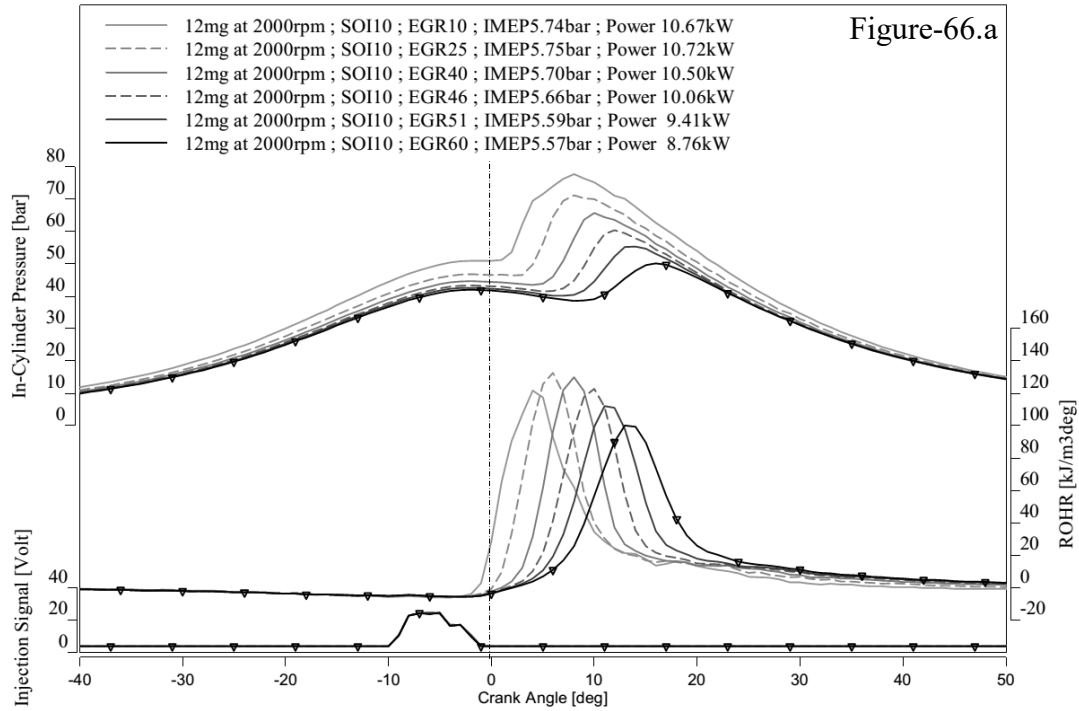


Figure 4.66 : Combustion data of 12 mg at 2000 rpm for SOIs'-10, -20, -30, -40 CA.

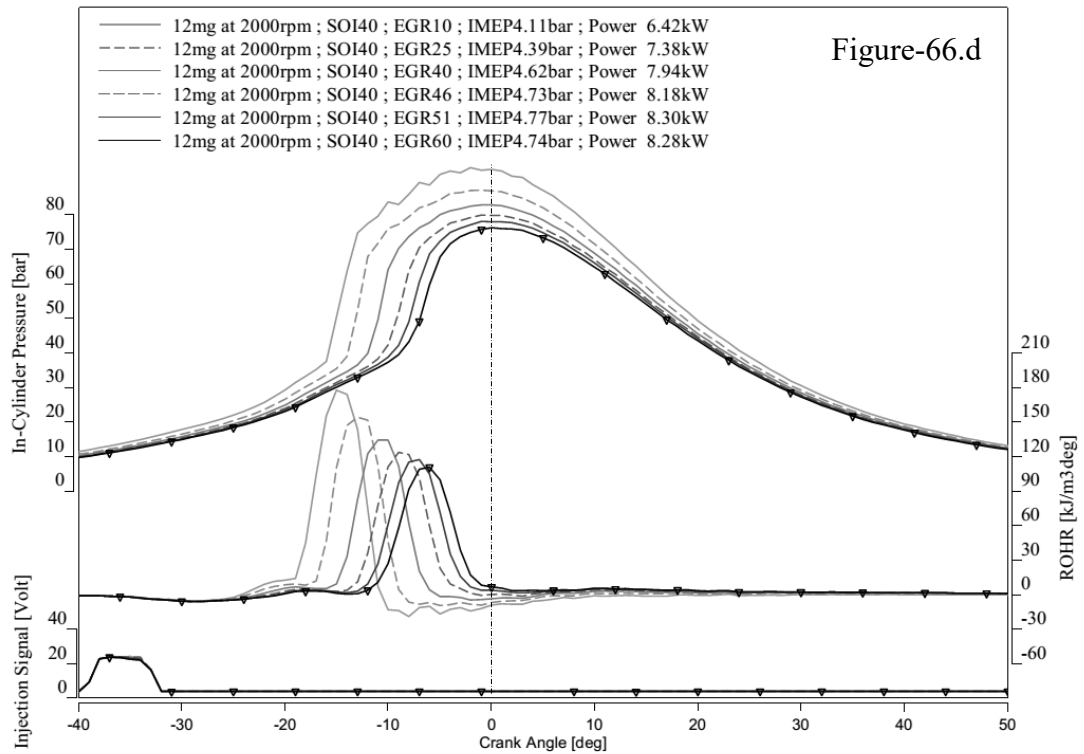
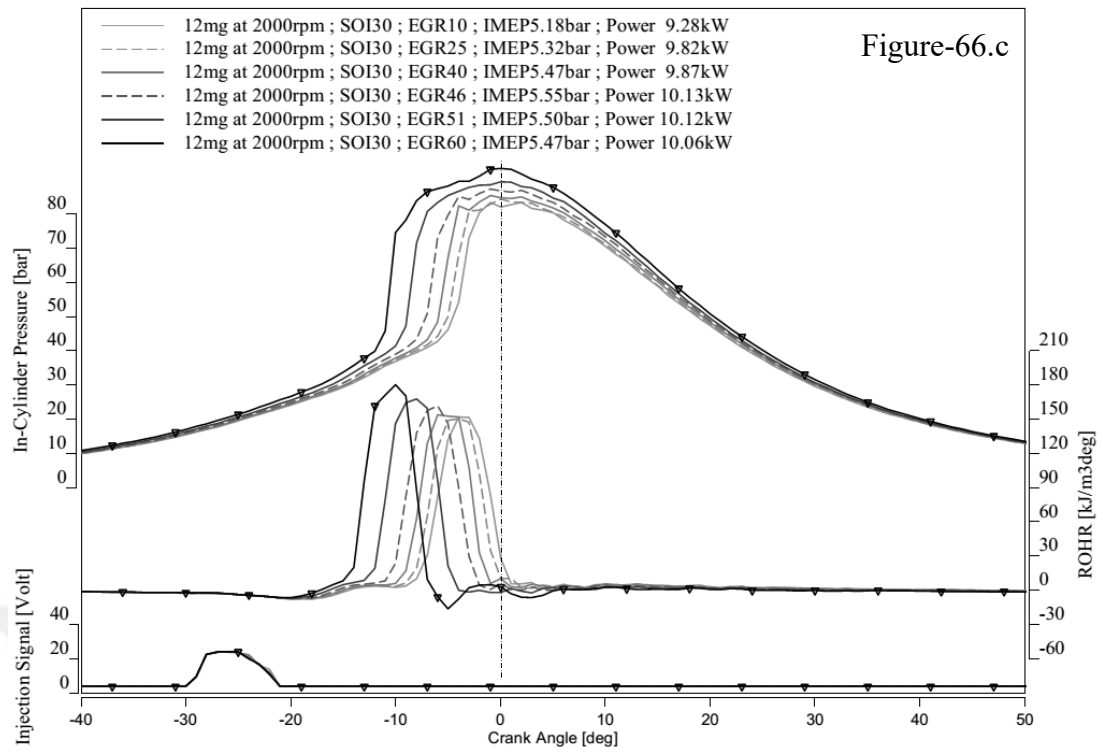


Figure 4.66 (continued) : Combustion data of 12 mg at 2000 rpm for SOIs'-10, -20, -30, -40 CA.

4.5.4.2 Torque

Torque map of the 2000 rpm, 12 mg fuel injection according to SOI and EGR adjustments were plotted in Figure 4.67. Since the torque has a greater variation in numerical value compared to the power, the torque map was drawn instead of the power to better observe the variation according to the SOI and EGR.

The torque changed from 25 Nm to 52.5 Nm according to SOI and EGR as shown in Figure 4.67. The highest torque was observed under 20% EGR rate at 15 CA bTDC SOI and the lowest torque was observed under 10% EGR rate at 50 CA bTDC SOI. Retarding the SOI from 15 to 60 CA bTDC, under the same EGR rate, reduced the torque continuously. In addition, rise in EGR rate reduced the torque as expected from literature up to 25 CA bTDC SOI. However, when SOI is earlier than 30 CA bTDC, the rise in EGR rate increased the torque (like in 6.6 mg case) as opposed to its conventional use. Although rise in EGR rate improved the torque for the SOIs earlier than 25 CA bTDC, the improved torque was still below the observed highest torque under conventional diesel combustion.

The observed torque reduction up to 25 CA bTDC SOI due to rise in EGR rate can be explained with the reduced clean air mass in the cylinder. Besides, the observed torque reduction due to earlier SOI might be the reason of the higher in-cylinder pressure in the compression stroke that increase the pumping loss. On the other hand, the observed torque increase with the rise of EGR for the SOIs earlier than 30 CA bTDC, was also caused due the effect of EGR on in-cylinder pressure. Heavy EGR flow delayed the SOC and rise of in-cylinder pressure as can be seen in Figure 4.66-c and Figure 4.66-d. Combustion was postponed to aTDC, which reduced the losses in the compression stroke and enabled to achieve higher power.

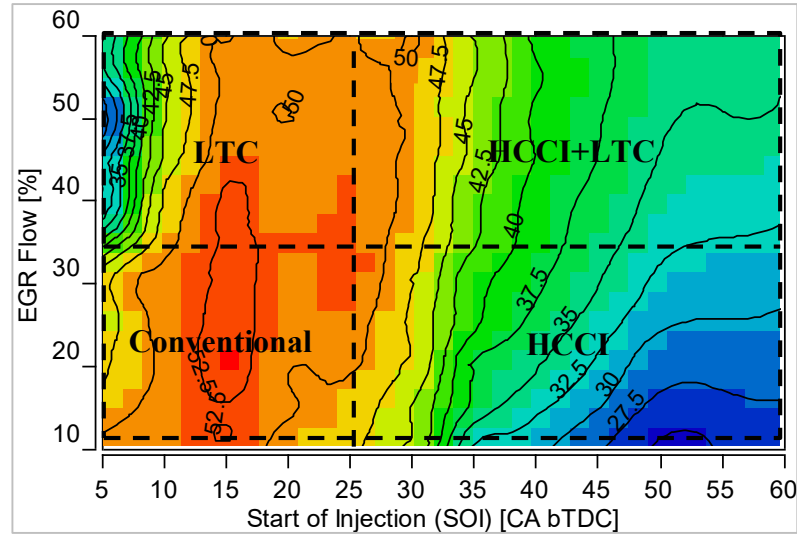


Figure 4.67 : Torque change according to EGR and SOI.

4.5.4.3 NO_x and soot emission

The NO_x emission is the biggest environmental problem of the diesel engines, as it cannot be converted to N₂ and O₂ with 3-way catalyst like in gasoline engines. Figure 4.68 shows the NO_x emission map in ppm unit for the measurements of the 70 different EGR and SOI combination. NO_x emission reduced below 50 ppm for all SOIs when EGR rate is above 40%. Advancing the SOI increased the NO_x emission.

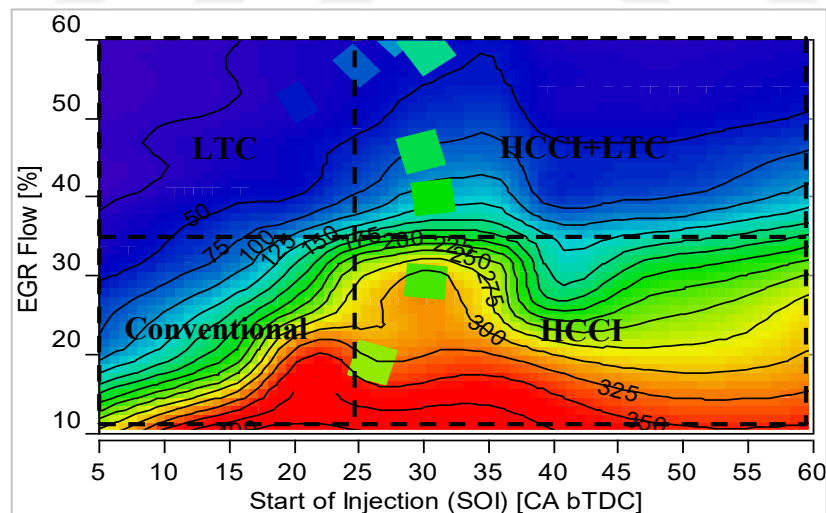


Figure 4.68 : NO_x change in ppm according to EGR and SOI.

However, above 40% EGR rate, improving the homogeneity of the air fuel mixture reduced the NO_x emission. Therefore, EGR is more effective on NO_x emission reduction compared to SOI.

The decreasing effect of EGR rate on NO_x emissions can be explained by the in-cylinder temperature because the rise in the EGR flow reduced the clean air in the

cylinder and combustion efficiency. On the other hand, the increasing effect of early SOI on NO_x emissions can be explained by the high in-cylinder pressure, which also mean high in-cylinder temperature, shown in Figure 4.66-a to d. As there is cleaner air in the cylinder compared to high EGR flow conditions, air-fuel mixture started to burn faster and a steeper rise was observed at in-cylinder pressure and temperature. Moreover, in-cylinder temperature reached earlier to the high temperature and excess O_2 in the cylinder had more time to react with the N_2 .

In addition to NO_x catalyst, Diesel Particulate Filter (DPF) is also being used in diesel engines to comply EURO-6 PM and PN limits although it is not necessary in gasoline engines. Combustion temperature, pressure and available oxygen in the cylinder influence the soot formation process, therefore different compositions of soot are generated under advanced combustion and conventional combustion [106,107]. The stages of soot formation (pyrolysis, nuclei, surface growth, and coalescence and agglomeration) are different for diffusion combustion and premixed combustion, moreover pressure and temperature condition in the cylinder affect the size of the soot [106]. Figure 4.69 shows measured Opacity emission after DPF for 70 different conditions. Opacity was found to be less dependent to EGR compared to NO_x because there was not a continuous trend in opacity according to EGR flow. The highest opacity emission was observed at SOI 15 CA and 35 CA bTDC above 40% EGR rate. Under the other SOI conditions, use of higher EGR rates did not worsen the soot emission. Although the size of soot particulates were not known as measurement were performed with opacity device, observed high opacity under conventional combustion (SOI 10 to 20) and low temperature combustion region (SOI 35 to 40) can be explained with the two different soot formation processes. Observed high opacity under the SOI 10 to 20 CA region was more similar to the conventional particle formation, as supported by the burn duration. However, the high opacity under SOI 35 to 40 CA could be explained with particle formation under advanced combustion, which started with the increase of premixed combustion and resulted in reduced particle size.

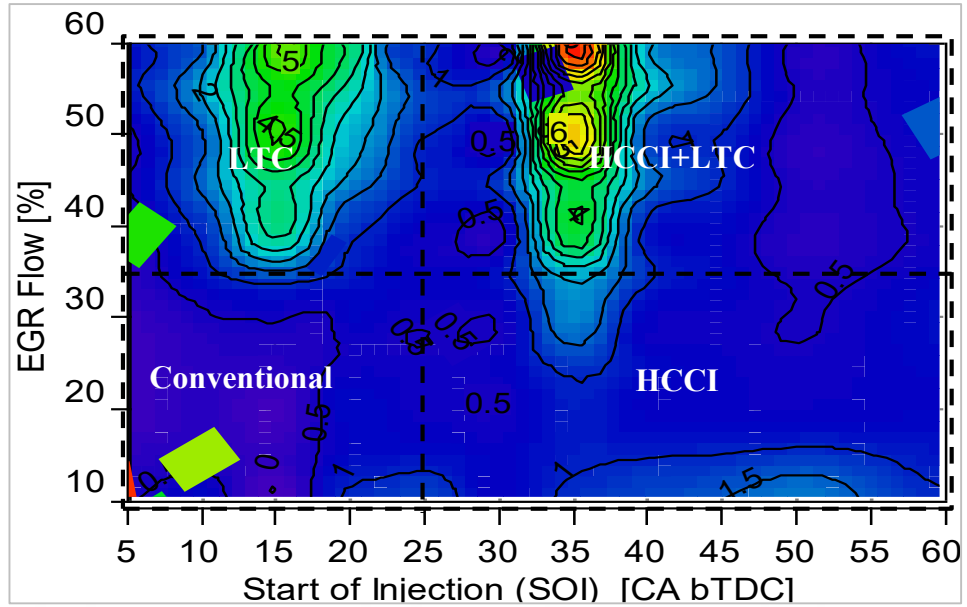


Figure 4.69 : Opacity change according to EGR and SOI.

4.5.4.4 Comparison of the original case with selected optimum case

In order to study high load case of advanced combustion application, 70 different engine conditions were tested for the single injection of the 12 mg/stroke fuel at 2000 rpm. Results of the SOI and EGR adjustments were analyzed to determine the optimum parameter set for range extender application. Table 4.13 shows the engine parameters of the original and selected optimum cases. In the original configuration, fuel was injected in three parts to the combustion chamber as given in Figure 4.70. Triple injection configuration was modified to single injection as shown in Figure 4.70 and rate of EGR was increased from 32 to 60%.

Table 4.13 : Engine parameters of the original case and selected optimum case.

Parameter	EGR	Q-Inj-1	SOI-1	Q-Inj-2	SOI-2	Q-Inj-3	SOI-3
Unit	%	mg	CA	mg	CA	mg	CA
Optimized Case	60	12	-30	0	0	0	0
Original Case	32	9.3	-2.4	1.7	-20	1.0	-33.5

In-cylinder pressure, ROHR, and injection signal of the original and optimized case can be seen in Figure 4.70. Position and magnitude of the maximum in-cylinder pressure was changed, higher and earlier maximum in-cylinder pressure was observed under optimized case compared to the original case. ROHR started almost same crank angle for both cases, however injected fuel was burned faster at optimized case than original case. Although steep rise of in-cylinder pressure enabled to achieve higher

power, noise of the engine was increased due to higher-pressure gradient of the optimized condition compared to original case.

Under the optimized case, all fuel injected at once and SOI was at -30 CA. Therefore, all fuel was injected before SOC, however under the original case, fuel burned at two different time as can be seen in Figure 4.70. Although, premixed combustion was observed under both condition, there was more homogenous mixture at optimized case. Fuel burned in shorter duration under optimized case compared to original case and the optimized case can be defined as PCI because of premixed combustion with single ROHR.

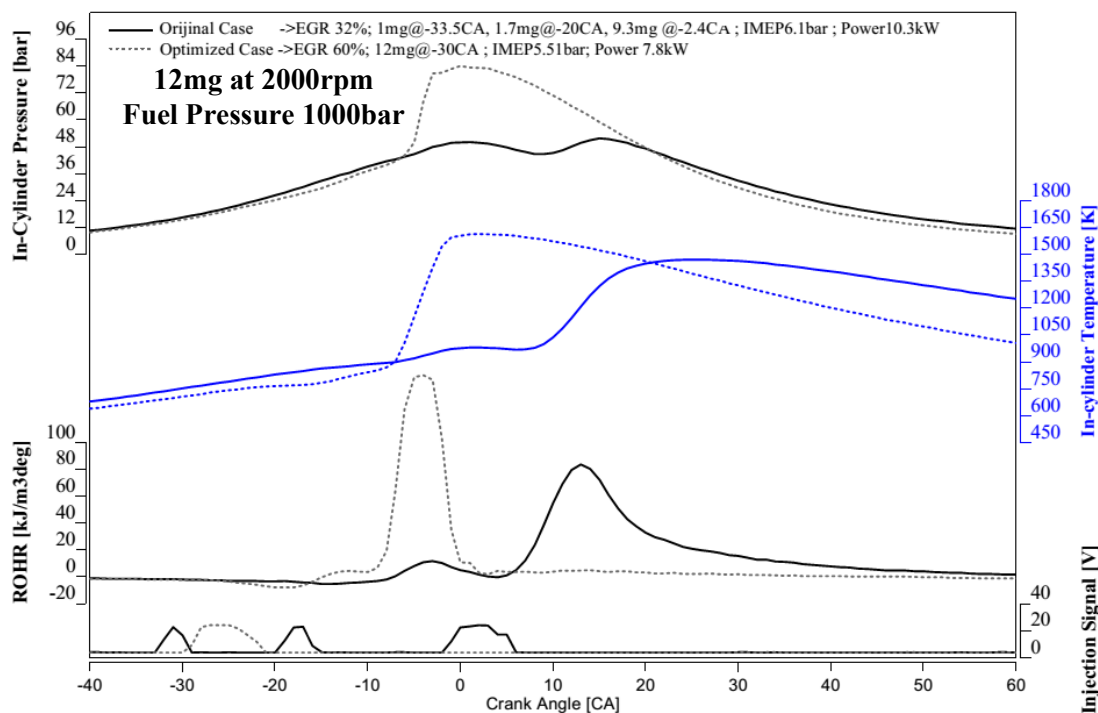


Figure 4.70 : Combustion data of original and selected case.

Figure 4.71 shows the comparison of the Power, NO_x and opacity of the optimized and original cases. Similar to previous injection configurations, power, NO_x and opacity measurements of both selected optimum and original cases were normalized to 100 (power, NO_x, and opacity of the original case were enlarged to 100 and the optimized case was enlarged with same multiplier), to able to plot in one graph. As shown in Figure 4.71, all of the engine outputs; power, NO_x and PN emissions were improved with selected case. The NO_x and opacity emissions can be reduced 32% and 20% respectively with new engine setup, moreover 32% increase was observed at power.

Therefore, proposed combustion method for range extender application can save fuel while reducing the NO_x and opacity emissions.

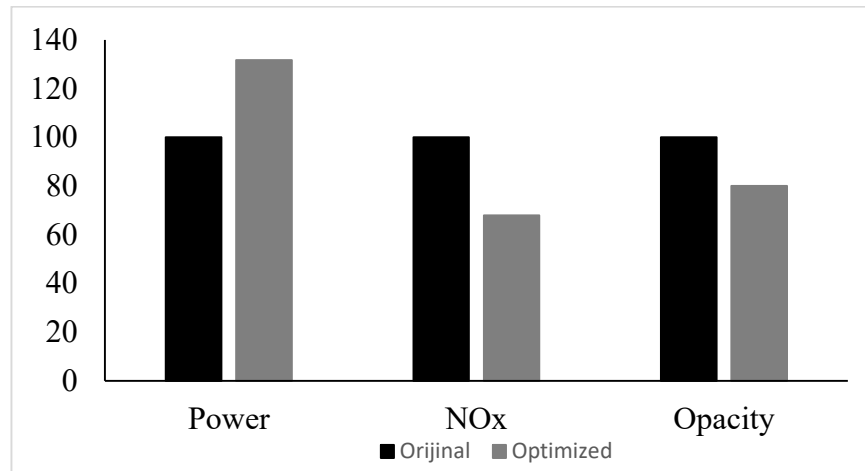


Figure 4.71 : Comparison of the Power, NO_x and Opacity of original and optimized case.

Table 4.14 also shows the values of the power, NO_x and opacity that compared in Figure 4.71.

Table 4.14 : Results of the original and optimized case.

Name of The Case	Power	NO _x	Opacity
Unit	kW	g/kWh	%
Original Case	7.84	3.6	3.2
Optimum Case	10.33	2.5	2.5



5. RANGE EXTENDED HYBRID VEHICLE MODEL AND RESULTS

The last section of the thesis is prepared to show developed injection configuration can be utilized as an alternative for range extended electrical vehicle applications. Hybrid electric vehicles (HEV) is getting more popular due to stringent emission regulations of NO_x and soot. However diesel engines are not preferred for HEV due to necessity of additional after treatment equipment that has a cost more than 2,000\$ to the vehicle [108]. Moreover, urea have to be used to convert NO_x emission properly, this consumable also added an extra cost to the vehicle. Suggested combustion method enable to achieve EURO-6 emission limits with EURO-5 vehicle exhaust layout. This may attract the car manufacturers to utilize diesel engine for hybrid applications.

The range-extended electrical vehicles have both battery and internal combustion engine to supply energy to its traction electric motor. However, required energy at the traction motor of the vehicle is always supplied over the battery. The range-extender engine is coupled with generator to produce necessary energy when the battery charge is below pre-defined level. Engine of the range-extender works only under steady state load conditions such as low, medium and high loads. Because, battery supply the transient energy, which is necessary while accelerating the vehicle or climbing hill. This enable to use developed advanced combustion methods for diesel engine of the range-extended vehicles.

In order to show the potential of the advanced combustion methods for range extended electric vehicles, which will be popular in the future due to stringent emission regulations, energy consumption and emissions were calculated according to results of the optimized and original 12 mg load condition.

Before starting calculation, a vehicle layout for diesel range extended vehicle was created to indicate the modification, which shall be performed on conventional vehicle. After, efficiency of the components, which are necessary for hybrid vehicle, were discussed.

Once range extended vehicle model was defined, required energy was calculated from result of the WLTC experiment of the conventional vehicle that were presented in

Figure 5.1. Afterwards, working duration of the range extended vehicle was calculated from the developed model. Since engine worked at same condition as range extender, emissions and fuel consumption of the engine were also calculated from the same working duration.

Last, comparison of the three cases (1-conventional vehicle, 2-diesel range extender with original parameters, 3-diesel range extender with optimized parameters) were performed for NO_x, opacity and fuel consumption in three different plots. OEMs work on aftertreatment equipments and combustion to reduce the size of exhaust system to make it cheaper to be competitive in the market. Fuel consumption is also an important parameter for marketing, therefore NO_x, opacity, and fuel consumption of the conventional vehicle and simulated range extended vehicle under the WLTC were compared. Two different cases of the simulation were calculated to show the improvement of the proposed combustion method. First, calculation was conducted as the engine run under original engine parameters for range extender application. Second, calculation was performed for the optimized case.

5.1 Assumptions and Calculations of the Range Extended Vehicle Model

Conventional vehicles have less complex powertrains compared to hybrid electrical vehicles. However, EU give incentives for hybrid vehicles, as utilization of battery lowers greenhouse gas emissions and dependency to petroleum fuel.

Figure 5.1 shows the schematic of the components of the conventional vehicle and hybrid electrical vehicle, which has range extender to charge the batteries of the vehicle. Transmission and differential are enough to transmit the rotation and power of the engine to wheels at conventional vehicles, however three more components (generator, battery, traction motor) are necessary for range extended electrical vehicle application.

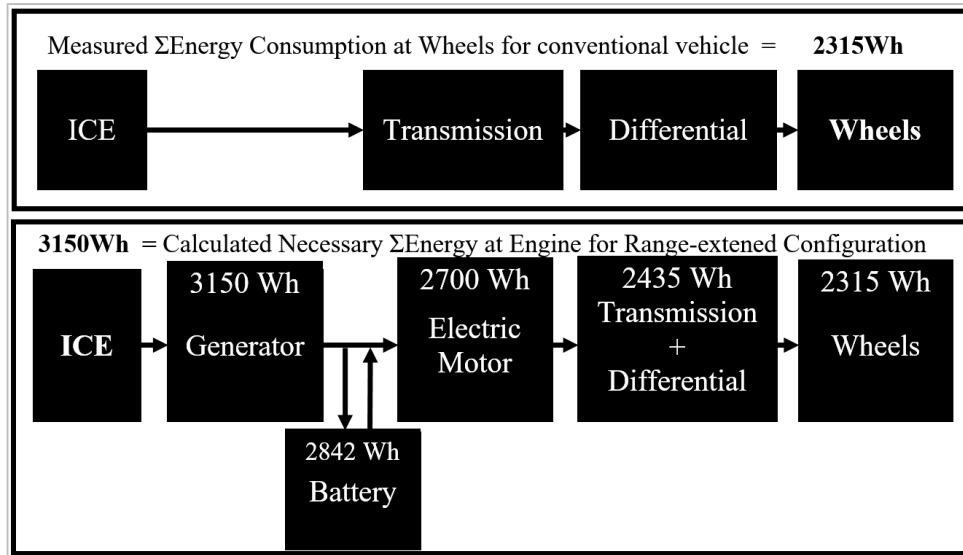


Figure 5.1 : Conventional and Range-extended electric vehicle configuration.

Battery supplies energy to the traction electric motor to rotate the wheels of the vehicle. In addition to the traction electric motor, a generator is necessary. Power output of the internal combustion engine is converted to electric power via generator. Once power of the engine converted to DC power via coupled generator, it can charge the battery of the vehicle. Therefore, internal combustion engine has not any mechanical connection to the wheels. Internal combustion engine creates electricity via coupled generator to charge the battery and the working condition of the engine shall not be changed according to velocity of the vehicle. Figure 5.2 shows an example of the range extender hybrid electrical vehicle. An electric traction motor applies power directly to the front wheels of the vehicle. Battery of the vehicle is between the front and rear wheels, under the seats. Range extender engine and the generator is at the rear side of the vehicle.

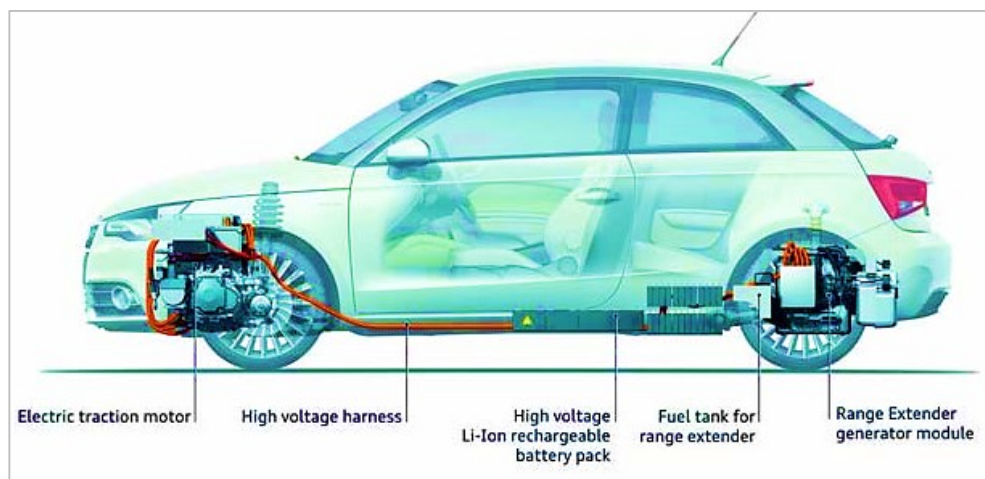


Figure 5.2 : Example of the range extended electric vehicle.

Energy consumption of the range extended electrical vehicle was calculated via developed model. In order to develop a proper model, efficiency of the components has to be defined. Efficiency of the electric motor, the inverter, and the transmission change according to speed and power of the vehicle. Besides, internal resistance of the battery change according to state of charge and temperature. Efficiency values for the electric motor, inverter, transmission and battery of the configured diesel range extended vehicle were taken from the literature [109–111]. Additionally, energy that can be recovered by regenerative braking during deceleration conditions of the WLTC were not considered while calculating the necessary energy, because the results of range extender configuration with advanced combustion method are much better than conventional configuration even without these assists.

Figure 5.3 shows the efficiency map of permanent magnet electric motor that is utilized for electric vehicle applications. As electric motor can be controlled via inverter, the map also includes the efficiency of the inverter. Permanent magnet type electric motors are utilized on electrical vehicles as they have higher efficiency with smaller dimensions compared to asynchronous electric motors.

Similarly, generator of the range-extender unit is selected from permanent magnet type electric motors. Therefore, efficiency of the generator is similar to the efficiency of the traction motor.

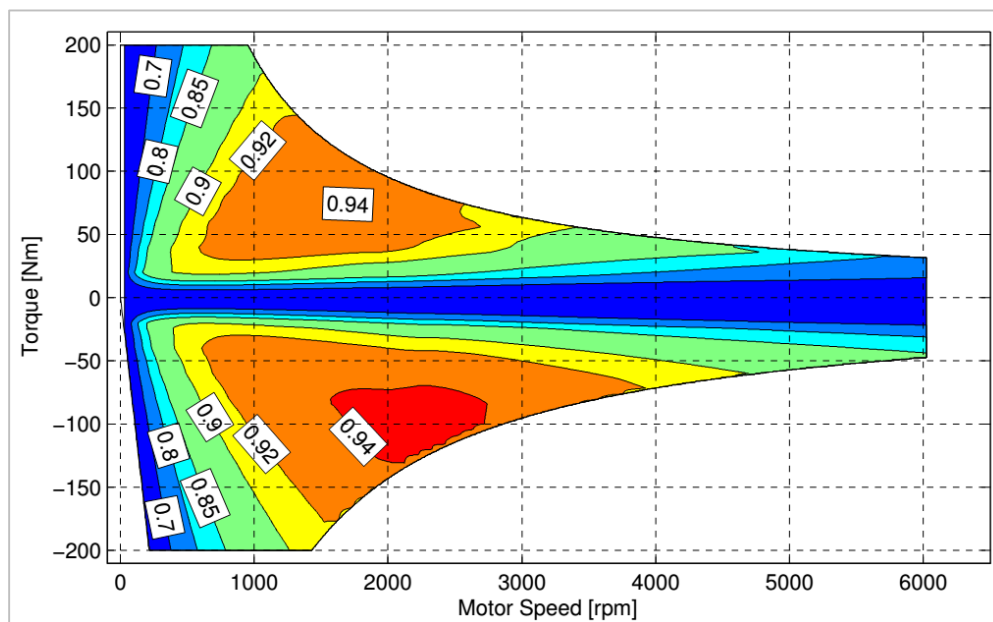


Figure 5.3 : Efficiency of a typical permanent magnet electric motor (including inverter) [109].

Figure 5.4 shows the efficiency change of the high voltage battery of an electrical vehicle. Internal resistance of the battery changes according to the state of the battery and discharge or charge rate of the battery. However, as can be seen from Figure 5.4 efficiency of the battery is over 90% overall the map.

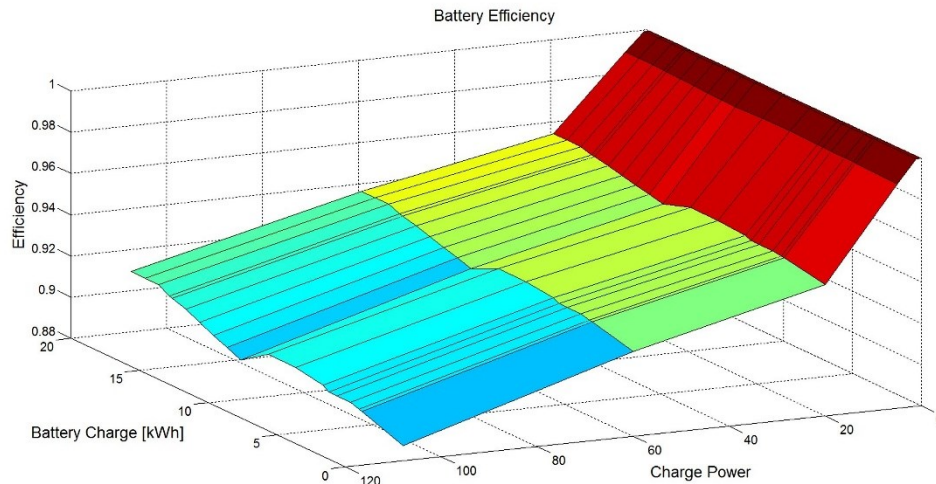


Figure 5.4 : Efficiency map of a typical electrical vehicle battery.

Figure 5.5 shows the efficiency of the transmission efficiency of a typical electrical vehicle. In order to reduce the dimensions of the electric motors, speed of them increased to 10000 rpm. As electric motor run at high speeds, a transmission is needed to lower the speed. Since electric motors can supply the same power under all speeds, one fixed ratio transmission is enough. Because of fixed ratio and coupled differential, most of the transmission suppliers (ZF, Borgwarner) can satisfy an efficiency over 96% in the whole working range of the vehicle.

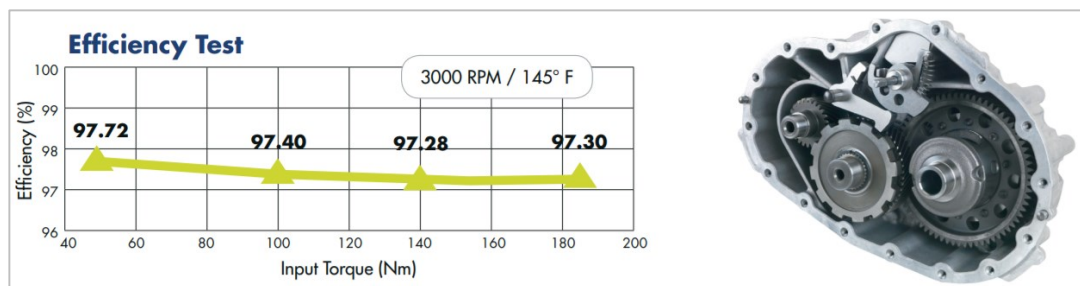


Figure 5.5 : Efficiency of the transmission and differential for electrical vehicles.

5.2 Calculation of Developed Range Extended Vehicle Model

First, energy consumption at the wheels of the conventional Ford Fiesta was calculated by summing the multiplication of the measured force and velocity for each 0.1 second

of the WLTC test (equation (5.1)). Figure 5.6 shows the velocity and power measurement from the test bed PC-2.

$$\Sigma E_{wheels} = \Sigma_i^n F_{wheel.i} * V_{wheel.i} \quad (5.1)$$

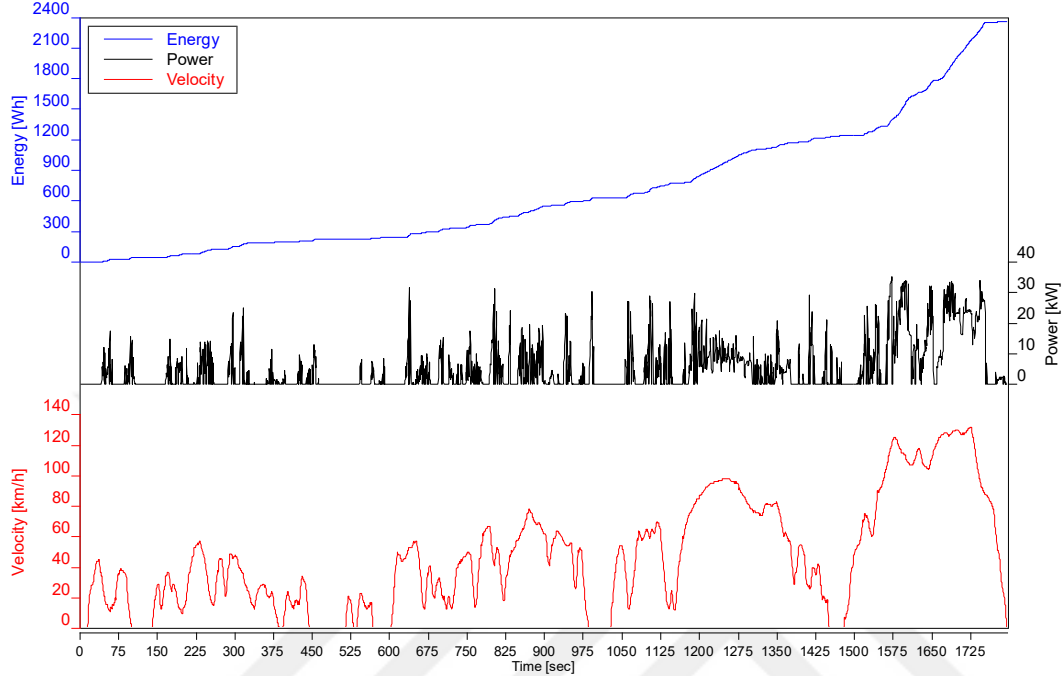


Figure 5.6 : Energy consumption of the conventional vehicle during the WLTC.

After finding the necessary energy at the wheels to run WLTC with test vehicle, the total energy that range extender should generate was calculated by evaluating the efficiency losses (equation (5.2)) of the necessary conversion components, which are shown in Figure 5.1 and Figure 5.2. First, internal combustion engine generates the power. Then, coupled generator to the crankshaft of the engine generates electricity; during this conversion, an energy loss is inevitable. The efficiency of the coupled generator was selected as 90%, which is the worst efficiency of the motor and inverter set in Figure 5.3. Second energy loss is during the charging and discharging of the battery. This efficiency can be selected as an average of 95% from Figure 5.4. Third energy loss is at the traction motor of the vehicle. As traction motor has similar characteristics with generators, efficiency of the traction motor and its inverter can be utilized as 90% from Figure 5.3. The last energy loss is at the transmission. As transmission generally has high efficiency ratio, 97% efficiency can be utilized at the model while calculating the energy loss.

$$E_{REX} = \frac{E_{wheels}}{\eta_{trans} * \eta_{motor} * \eta_{battery} * \eta_{generator}} \quad (5.2)$$

In this calculation case, the necessary power and energy is known at the wheels of the vehicle. Therefore, calculation was performed from wheels to the ICE of the range extender. All defined energy losses were added on to the measured energy at the wheels.

Besides, regenerative braking during deceleration conditions of the WLTC were calculated from the measured brake power during the WLTC test. Results can be seen in Figure 5.7, approximately 600 Wh of energy can be recovered via regenerative braking. However, it may not possible to gain all this energy, due to high deceleration necessity at hard braking cases.

Calculated total energy conversion losses were ~850 Wh. Although some part of the losses can be covered via mentioned regenerative braking, it was not extracted from the necessary power of the engine. In this way, the worst case of the range extended vehicle was compared with original vehicle. Nevertheless, the results of range extender configuration with advanced combustion method are much better than conventional configuration even without regenerative braking assist.

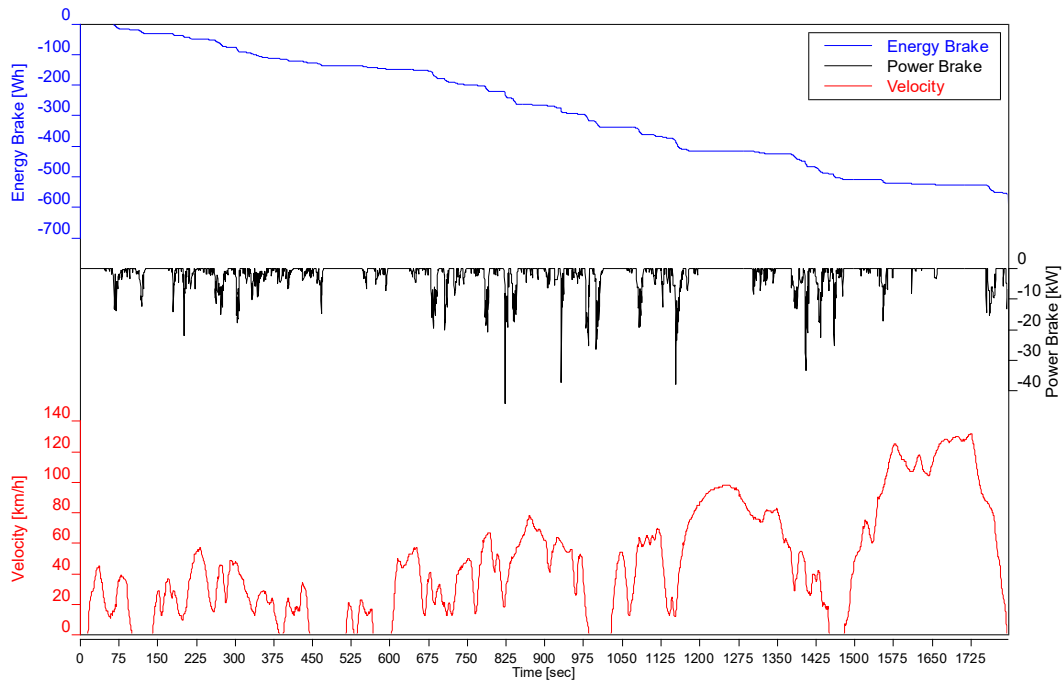


Figure 5.7 : Energy that can be recovered via regenerative braking.

After calculating the necessary energy that have to be generated via internal combustion of the range extender unit, necessary working duration of the engine was calculated. As the advanced combustion experiments were performed at 10 kW, work duration of the diesel engine as range extender was calculated by dividing necessary energy to engine power (equation (5.3)). Additionally, working duration of the original condition of the engine was also calculated to compare difference of direct utilization of the engine as range extender with proposed method.

$$t_{REX\ work} = \frac{E_{REX}}{P_{REX}} \quad (5.3)$$

Figure 5.8 shows the working duration of the engine as range extender for proposed advanced combustion method at 10 kW and original engine parameters that created 7.8 kW. Engine has to run 1100 seconds according to the model under advanced combustion condition as can be seen in Figure 5.8. However, engine has to run 1445 seconds under the original configuration of the engine. Both conditions create 3150 Wh of energy to be able to run the vehicle under the WLTC.

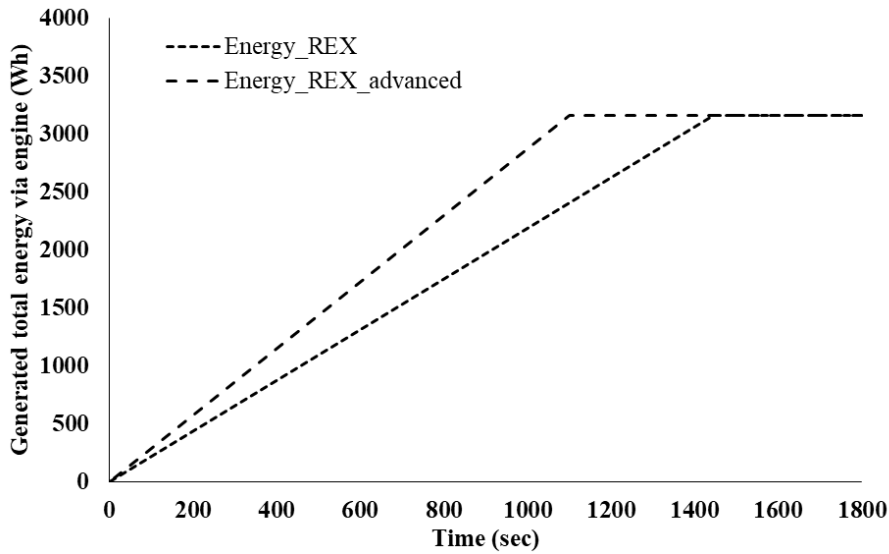


Figure 5.8 : Engine working duration for two cases.

Once the necessary engine operation duration was determined from necessary energy, fuel consumption, NO_x, and Opacity emissions were calculated by multiplying the dynamometer measurements with the time period. Equations (5.4), (5.5) and (5.6) were utilized respectively to calculate the total emissions and fuel consumption of the two cases. Results of the two different range extender cases were compared with the WLTC results of the conventional vehicle.

$$\Sigma NO_x = t_{REX\ work} * NO_x\ at\ working\ condition \quad (5.4)$$

$$\Sigma Fuel\ Cons. = t_{REX\ work} * Fuel\ Cons.\ at\ working\ condition \quad (5.5)$$

$$\Sigma Opacity = t_{REX\ work} * Opacity\ at\ working\ condition \quad (5.6)$$

5.3 Results of the Model and Comparison with the Conventional Vehicle

Engine parameters of the selected optimum case for 2000 rpm, 12 mg fuel injection condition were 30 CA bTDC, 60% and 1100 bar for SOI, EGR flow and fuel injection pressure respectively. Measured torque, NO_x and opacity were 49.3 Nm, 2.45 mg/h and 0.257 k⁻¹ respectively for selected case. Whereas, measured torque, NO_x and opacity were 37.5 Nm, 3.60 mg/h and 0.321 k⁻¹ respectively for original case. As 12 mg/stroke is constant for both configuration, fuel consumption was measured almost same for both condition from AVL-735S fuel consumption device, 2.819 kg/h and 2.824 kg/h respectively for selected and original cases.

Figure 5.9 shows the comparison of the fuel consumption of the three cases. Original vehicle consumed 1125 ml of diesel under WLTC. If diesel engine was utilized with its original configuration as range extender, engine can charge the battery in longer duration compared to the selected optimum case. Because engine power of the original case (7.8 kW) is lower than the power of the selected optimum case (10.3 kW). Engine shall run 1446 seconds under original case to supply necessary power to the battery of the vehicle and fuel consumption will be 1134 ml of diesel under WLTC, according to the model. As can be seen from the Figure 5.9, engine shall work 1100 seconds to create necessary energy for battery under selected condition. 861 ml of diesel is enough to run WLTC when engine run with the parameters of the selected optimum case, according to the created model.

Figure 5.10 shows the comparison of the NO_x emission of the three cases. Original conventional vehicle emit 8607 mg of NO_x under WLTC. As test vehicle is equipped for EURO-5 emission limit and optimized for the NEDC, NO_x emission is higher than EURO-5 under WLTC testing. NO_x emission of the range extender case with the original parameters of the engine is calculated as 1,085 mg for the whole cycle. When engine run with selected optimum case NO_x emission can be reduced 35%. Total NO_x

emission is calculated as 703 mg for the whole cycle when engine run with proposed advanced combustion method. As the total distance of the WLTC is 23.26 km, engine emits only 30 mg/km NO_x under WLTC, which is lower than the 80 mg/km limit of EURO-6.

Figure 5.11 shows the total opacity emission according to time scale of the WLTC for the three cases. Opacity emission was acquired after the DPF with 1 Hz for the whole cycle. Cumulative sum of the opacity emission for the conventional vehicle under the WLTC is measured 947 k-1. Opacity emission can be reduced via range extender application as can be seen in Figure 5.11. Optimized engine parameters to achieve advanced combustion enable to lower the Opacity emission 69% compared to conventional vehicle and 36% compared to the range extender application of original engine operation condition.

Proposed combustion method allow achieving 23%, 91% and 69% improvement at fuel consumption, NO_x, and opacity emissions respectively compared to the conventional vehicle. Table 5.1 shows the values for fuel consumption, NO_x, and opacity emissions for original vehicle, range extended vehicle with 1.5 liter diesel engine that has conventional combustion system and range extended vehicle range extended vehicle with 1.5 liter diesel engine that has advanced combustion system. Moreover, range extender application of the selected optimum case enabled to reduce fuel consumption, NO_x, and opacity emissions 24%, 35% and 36% respectively compared to range extender application of the original engine operation case.

Table 5.1 : Fuel consumption, NO_x and opacity comparison of three cases.

Parameter	Fuel Consumption	NO_x	Opacity
Unit	ml/cycle	mg/cycle	k⁻¹/cycle
Original Case	1125	8607	947
REX Original Engine	1134	1085	464
REX Advanced Combustion	861	703	297

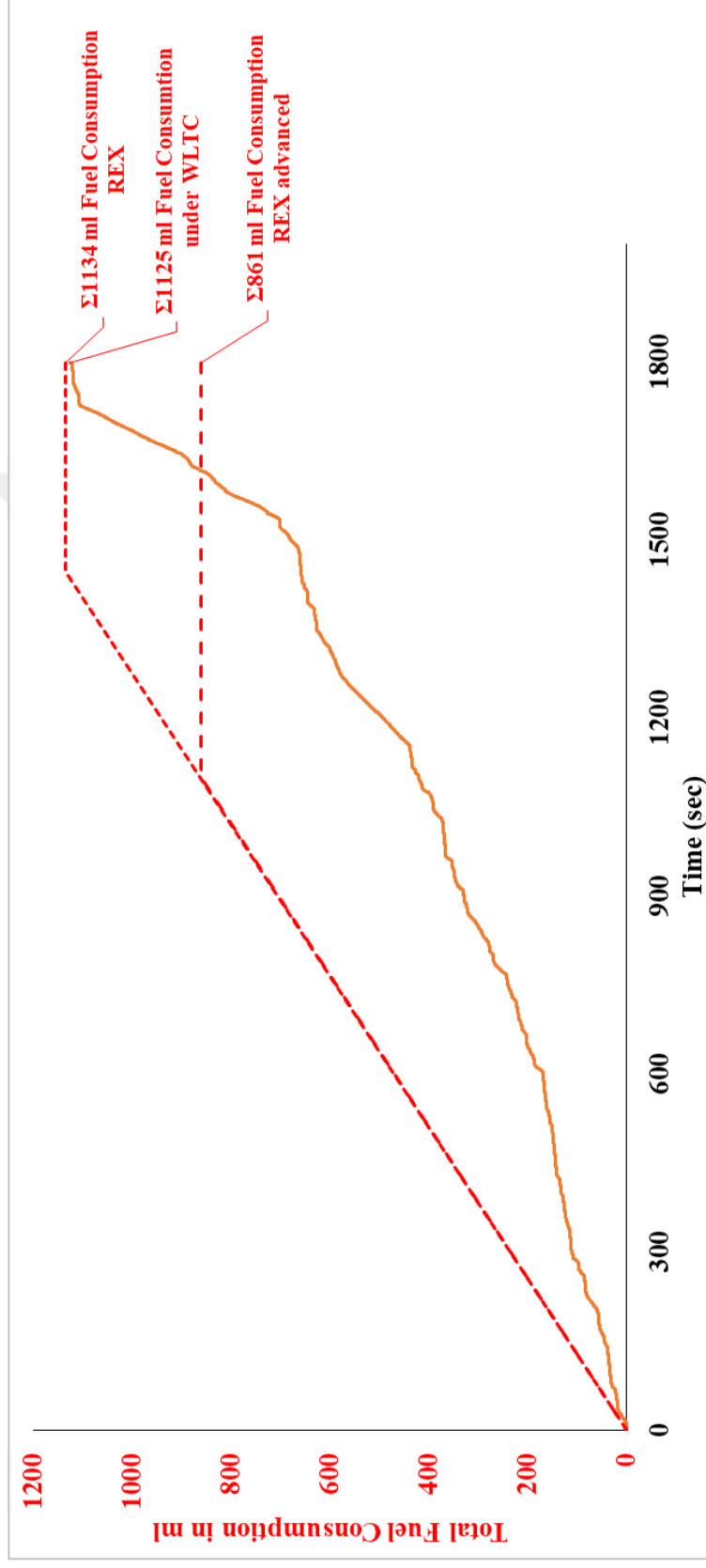


Figure 5.9 : Comparison of the fuel consumption of the conventional and two different range-extended configurations of the Ford Fiesta.

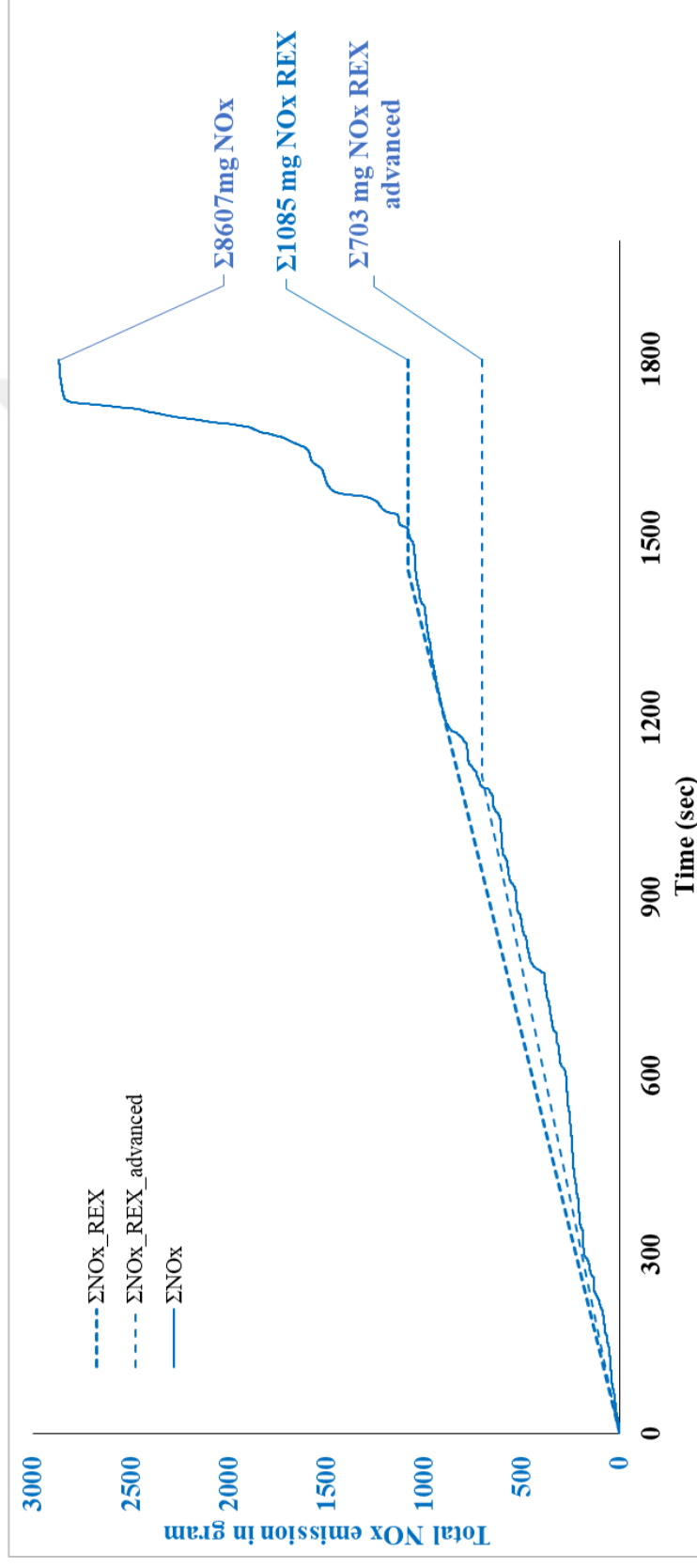


Figure 5.10 : Comparison of the NO_x emission of the conventional and two different range-extended configurations of the Ford Fiesta.

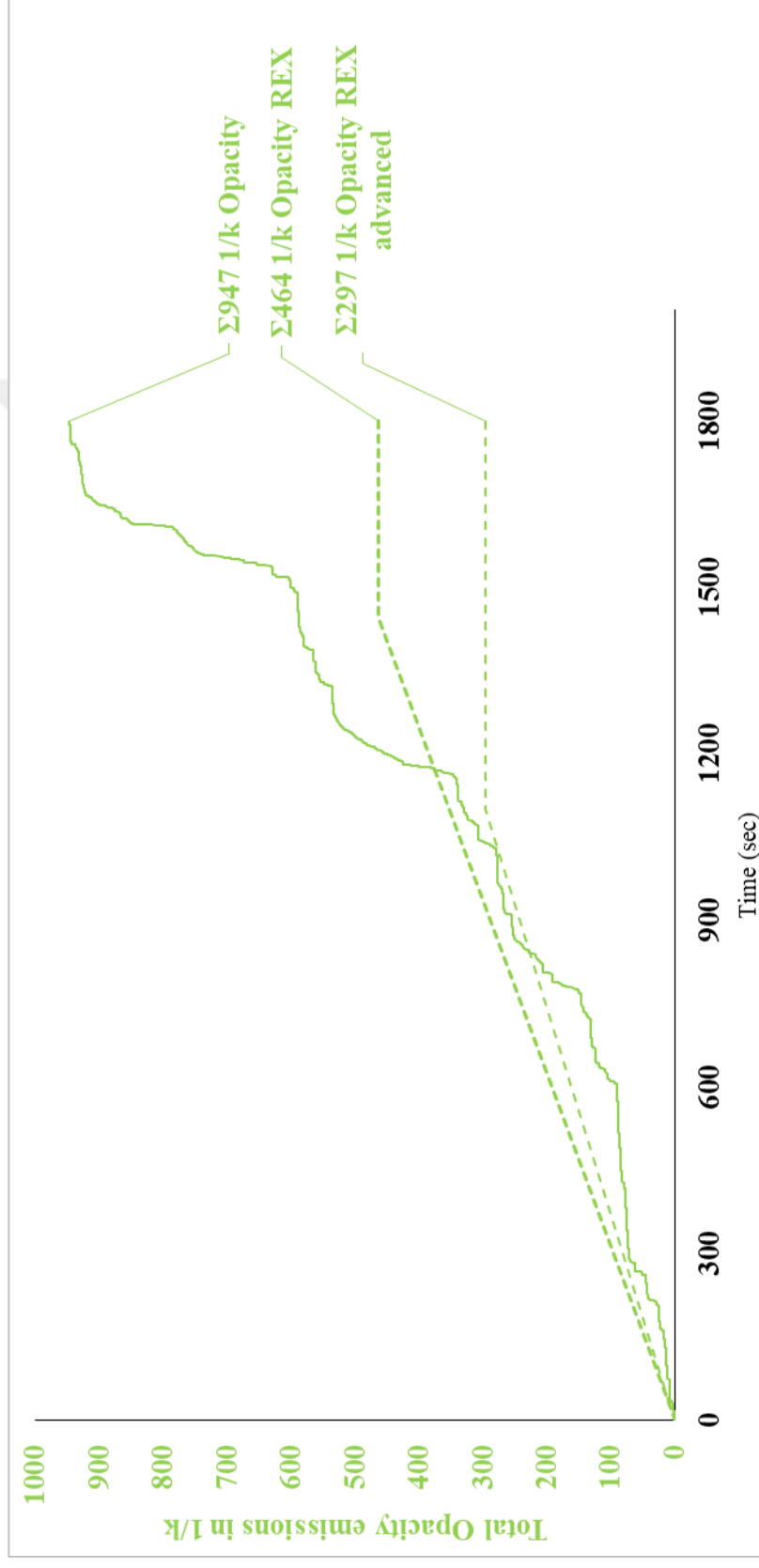


Figure 5.11 : Comparison of the opacity of the conventional and two different range-extended configurations of the Ford Fiesta.



6. CONCLUSIONS

Under this thesis study, experiments were performed both at chassis dynamometer on the vehicle and at engine dynamometer on the engine. First group of experiments were performed at chassis dynamometer without modifying engine control parameters with pure diesel and biodiesel blended diesel. After completing transient regulation cycles and steady state full load and part load experiments, engine was tested at engine dynamometers by modifying fuel injection parameters and EGR rate. All experiments and their results are summarized in the Table 6.1 at the end of conclusion.

In the conclusion section, important outcomes, which were derived from the represented results, were summarized. Conclusion of the experiments is given in two groups. First, outcomes of the full load, NEDC, cruise speed of NEDC and WLTC are presented. Then the results of the advanced combustion methods are presented.

In the first four group of experiments, a Euro-5 capable passenger car with 1.5 liter diesel engine was instrumented to be tested at chassis dynamometer under steady state and transient regulation cycle with pure diesel and biodiesel-blended diesel. The main goal of this part was to evaluate the effects of biodiesel on emission, performance and combustion characteristics, as well as to explain the changes in emissions by the changes in combustion parameters. Meanwhile, new and old regulation cycles were compared both load distribution and biodiesel sensitivity point of view.

Based on the conducted experiments and associated analyses and evaluations, the following conclusions are drawn;

- Fuel volume was kept constant for all fuels during the FL experiments and power is found to decrease with biodiesel addition. Besides, to keep the vehicle velocity identical (at NEDC and WTLC experiments), higher amount of biodiesel is found to be necessary compared to diesel. Therefore, utilization of biodiesel increases fuel consumption, mainly due to lower GCV of biodiesel than diesel.
- NO_x emission increased with biodiesel use due to higher oxygen content and shorter ignition delay duration of biodiesel, which caused rapid rise of in-cylinder

temperature for biodiesel blended fuel.

- The use of 10% biodiesel blend ratio has the most balanced benefit as reduced CO, HC and PN respectively 30%, 20% and 25% while 5% and 2% increase was observed at NO_x emission and fuel consumption.
- Biodiesel is also suitable for advanced combustion methods due to its higher oxygen content and shorter IDD and BD compared to pure diesel. However, fuel injection parameters should be optimized according to effects of biodiesel on combustion characteristics.
- The use of biodiesel caused a higher increase in NO_x and CO₂ emissions whereas a higher decrease in CO, HC, and PN emissions under the WLTC (compared to the NEDC), respectively, due to higher velocity profile and exhaust temperature of the WLTC. Therefore, the WLTC are more sensitive to biodiesel-blended fuels than the NEDC. Besides, engine operated under more distributed and higher load condition under the WLTC compared to the NEDC, which caused 3% higher fuel consumption under the WLTC than the NEDC.

The aforementioned improvements at CO, HC and PN may allow the OEMs to reduce the size of DOC and DPF in the exhaust system. Moreover, OEM may request to use biodiesel-blended diesel for certification tests.

In the fifth group of experiments, a 1.5 liter diesel engine was tested in an engine dynamometer at 2000 rpm with 6.6 mg and 12 mg fuel injection operation condition by modifying fuel injection parameters and EGR flow. Main objective of these experiments were to evaluate the effects of fuel injection parameters and EGR rate on emission, performance and combustion parameters, in order to investigate the improvements and potentials of the different advanced diesel combustion methods. Besides, evaluated best case for 12 mg operation condition was considered as the operating state of the range-extended configuration of Ford Fiesta that has 1.5 liter diesel engine. Calculated emissions were compared with the measured emissions of test vehicle.

Based on the analyses and evaluations of the performed experiments, following conclusions are drawn;

- Splitting injection quantity reduced the power output of the engine.

- Retarding fuel injection after TDC reduced the power output of the engine.
- Very early injection of the fuel reduced the NO_x and soot emissions, however greatly reduced the power output of the engine.
- Rise in EGR prolonged the IDD and BD because of reduced O₂ in the cylinder.
- Increasing of EGR flow for the SOIs earlier than 25 CA bTDC improved the power because of retarding combustion to the expansion cycle.
- Rise in EGR reduced the NO_x emission under any SOI condition due to reduced temperature and O₂ in the cylinder.
- Opacity is found to be dependent to the in-cylinder temperature and formation of soot varied both according to homogenization level and burn duration.
- Optimizing the engine according to proposed injection strategy under high EGR is found to improved power (20%), NO_x (30%), and Opacity (20%) emissions compared to the current EURO-5 complying version of the vehicle.
- Advanced diesel combustion methods are mainly clustered the naming of HCCI and LTC. Combination of these methods have the potential for improving emissions and performance compared to conventional combustion.
- Diesel range-extender configuration of the test vehicle that works under developed advanced combustion method has the potential to full-fill the latest EU emission regulations with lower fuel consumption.

The aforementioned improvements in NO_x and soot emission without compromising fuel economy may attract OEMs to design diesel range-extended vehicles with their existing diesel engine without investing new engine development cost. In turn, diesel hybrid vehicles may regain popularity, as exhaust system will be affordable. With these improvements, fossil fuels can be used more efficiently and environment friendly compared to conventional diesel system.

In further studies, additional improvements at emissions and fuel consumption can also be achieved by investigating the effects of fuel injection pressure and boost pressure as these parameters have direct impact on the homogenization level of the air-fuel mixture and combustion process.

Table 6.1 : Details of the each group of experiment.

Experiments	1st Set of Experiment	2nd Set of Experiment	3rd Set of Experiment	4th Set of Experiment	5th Set of Experiment		
				5.1	5.2	5.3	5.4
Target	* Experimental setup validation * Compare diesel biodiesel blends	* Compare biodiesel blends	* Define applicable engine operation points for ACM	* Compare WLTC - NEDC * Define ACM with single injection configuration * Evaluation of B30 for ACM	* ACM with double injection configuration	* ACM with triple injection configuration	* ACM under medium load contion with single configuration
Operation Condition	Full-load	NEDC	Cruise Speed of NEDC	6.6mg at 2000rpm	6.6mg at 2000rpm	6.6mg at 2000rpm	12mg at 2000rpm
Variables	2000, 2500, 3000, 3500, 4000 rpm	UDC, EUDC, NEDC	15, 32, 35, 50 km/h	Main SOI -45 to 0 CA EGR 0 to 60 %	Main SOI -8 to 8 CA Pilot SOI -70 to -10 CA Main-Q 3.8 to 5.5 mg Pilot-Q 1.1 to 3.8 mg EGR 0 to 60 %	Main SOI -9 to 7 CA Pilot2 SOI -40 to -10 CA Pilot1 SOI -90 to -30 CA EGR 0 to 60 %	Main SOI -60 to 0 CA EGR 0 to 60 %
Fuel Type	B00, B05, B10, B20, B30, B50	B00, B05, B10, B20, B30	B00, B05, B10, B20, B30	B00, B30	B00	B00	B00
Test Environment	Chassis Dynamometer	Chassis Dynamometer	Chassis Dynamometer	Engine and Chassis Dynamometer	Engine Dynamometer	Engine Dynamometer	Engine Dynamometer
Results	HC, CO, PM ↓ Fuel consumption ↑ NOx ↑	HC, CO, PM ↓ Fuel consumption ↑ NOx ↑	HC, CO, PM ↓ Fuel consumption ↑ NOx ↑	Power ↑ NOx , Opacity ↓ B30 < B00	Power ↑ NOx , Opacity ↓	Power ↔ NOx , Opacity ↓	Power ↑ NOx , Opacity ↓
Evaluation	Reduced BD and SOC has adverse effect O ₂ content has positive effect on homogenisation	B10 has potential for reducing size of exhaust components	Transient nature of WLTC has adverse effect on NOx	Low GCV of biodiesel reduce power	Diluted chamber prevent the ignition of the early injected fuel. Preparation of lean mixture prevent soot formation. Low O ₂ content reduce combustion temperature and prevent NOx formation. □		

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PUBLICATIONS/PRESENTATIONS ON THE THESIS

- **Kaya, T.;** Kutlar, O. A.; Taşkiran, Ö. O.; Evaluation of the Partially Premixed Compression Ignition Combustion with Diesel and Biodiesel Blended Diesel at Part Load Condition, In *7th International Conference on Fuels, Combustion, Engines and Fire*; Antalya, Turkey, 10-12 March 2019
- **Kaya, T.;** Kutlar, O. A.; Taşkiran, Ö. O.; Evaluation of the Effects of Biodiesel on Emissions and Performance by Comparing the Results of the New European Drive Cycle and Worldwide Harmonized Light Vehicles Test Cycle, *Energies* 2018, 11(10), 2814, <https://doi.org/10.3390/en11102814>
- **Kaya, T.;** Kutlar, O. A.; Taşkiran, Ö. O.; Investigation on the Emission, Performance and Combustion Characteristics of Diesel Engine Fuelled with Biodiesel-Diesel Blends, In *14th International Combustion Symposium*; Karabük, Turkey, 25-27 April 2018; pp, 52–60
- **Kaya, T.** Araç Emisyon Sorununun Altında Yatan Gerçek, *Bilim ve Teknik*, 55, 30-35, Ocak 2016

