

University of Wales Trinity Saint David

School of Engineering - M.Sc. Motorcycle Engineering



Tribological Analysis and the effects of Fuel/Oil chemistry on Engine Performance and Emissions



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Abbreviations

AFR -Air Fuel Ratio

API - American Petroleum Institute

ACEA - European Automobile Manufacturers' Association

ASTM - American Society of the International Association for Testing and Materials

BDC - Bottom Dead Centre

BSFC - Brake Specific Fuel Consumption

CFD - Computational Fluid Dynamics

CO - Carbon monoxide

CO₂ – Carbon dioxide

CR - Compression Ratio

ECU - Electronic Control Unit

EGR - Exhaust Gas recirculation

EGT - Exhaust Gas Temperature

ER – Equivalence Ratio

EU - European Union

FM - Friction Modifier

FMEP - Friction Mean Effective Pressure

HC- Hydrocarbon

H₂S – Hydrogen Disulphide

HTHS -High Temperature High Shear

ICE - Internal Combustion Engine

IMEP -Indicated Mean Effective Pressure / Instantaneous Mean Effective Pressure

IPM-Inorganic Particulate Matter

LSPI -Low Speed Pre-Ignition

LTHR – Low Temperature Heat Release

LVEO -Low Viscosity Engine Oil

NO – Nitrogen Monoxide

NO_x - Nitrogen Oxide

OPM – Organic Particulate Matter

PM - Particulate Matter

RON – Research Octane Number

SAE - Society of Automotive Engineers

SPI – Stochastic Pre-Ignition

TDC - Top Dead Centre

UHC – Unburned Hydrocarbon

ULEV- Ultra Low Emission Vehicles

ZEV – Zero Emission Vehicles

1.Introduction

The road transportation sector uses internal combustion engines to power its vehicles for decades, which has led to a growing demand for research and development in engine performance, fuel efficiency, and greenhouse emissions (Ambel, 2017) (Agency, 2020). For an internal combustion engine to deliver adequate results with optimized tailpipe emissions, automotive fuel must be completely burned under high pressure without knocking.

Traditional carbon-based fuels have been the primary source of energy for many decades. However, they are still responsible for many harmful pollutants in the atmosphere, mostly CO₂, CO, HC, and NO_x. Automotive fuel consists of more than 150 chemical compounds. Accordingly, Octane number varies from country to country due to the contrasting composition of chemical compounds that varies from crude petroleum, manufacturers, and seasons. (Registry, 1995).

Pre-ignition is an abnormal combustion process, wherein ignition and combustion are carried out before spark timing and typically not from a spark plug. (E. Singh, 2018) (V. S. Costanzo, 2019) (E. Chapman, 2014). Pre-ignition contributes substantially to the increased cylinder liner temperature and pressure. An optimum cylinder liner temperature has an influence on oil properties like kinematic viscosity and flashpoint. This parameter varies according to the base oil and additives used to improve the performance of the lubricant. The higher cylinder liner temperature over the flash point can lead to the volatility of the lubricant. Studies have examined the relationship between engine oil properties and oil consumption. Lubricants are the only source of lubrication, and it is essential to improve lubricant properties and performance by adding more chemistry to compounds. The choice of base oil and additives can help reduce oil consumption in ICE (Gilles, et al., 2007).

The geometries of optically accessible engines normally require unconventional features, such as enlarged piston clearances and ring packs located lower on the piston. In-cylinder processes are intended not to be affected, but Unburned Hydrocarbon emissions are dependent on the crevices in the top-land area of the piston (Dreizler, 2014) .

The objective of this paper is to understand the characteristics of research octane number (RON) and molecular diffusion of automotive fuel and lubricant on engine performance and emissions, to investigate the influence of molecular diffusion in piston top land crevice volume on hydrocarbon emissions by computational modelling and understand the possibilities and limitations of the CFD software analysis in the direction of research. Understanding particulate emission caused by the combustion of lubricant requires more attention due to the presence of chemical compounds in lubricant. Based on the combustion modelling and simulation, it is possible to figure out the fuel properties and techniques that will enhance the reliability of a naturally aspirated engine.

2. Industry Relevance

Transitioning to zero-emission cars and vans by 2035 delivery plan (Government of UK, 2021) as a part of the green industrial revolution to limit global warming is the primary objective of the government of the UK and other main countries progressing on the United Nations Paris agreement (United Nations, 2015). The potential pathway is achieved by more Ultra-Low Emission Vehicles (ULEVs) and Zero Emission Vehicles (ZEVs) in the next couple of decades. This includes the introduction of more electric vehicles, using of blended fuels, synthetic fuels without carbon, and the combustion of hydrogen in the research and development phase. To achieve zero tailpipe emissions a detailed analysis of hydrocarbon emissions by lubricant and depletion of the atmosphere by oxides of nitrogen and carbon requires a finite approach. Inappropriate thermodynamic operation of internal combustion engines due to pre-ignition, and temperature rise leads to the generation of hazardous by-products other than CO₂ which is committed to reducing by emissions regulatory regime by 2024.

Automotive gasoline exists in the form of hydrocarbon with different volatility (vapor pressure and distillation behaviour) and physicochemical properties which fluctuate according to the research octane number. Gasoline for example, is governed by specifications and standards based on the octane number needed on the market. Engines powered by right fuel requirements produce optimum power, controlled emissions, and fuel consumption (Wibowo CS, 2020) where the design and operation of the ICE are requisite.

Similarly physical and chemical properties of a lubricant have drastic changes from mineral oil to semi-synthetic and fully synthetic lubricants which is dependent on the type of base oil used. It is common for a user to go for cost-efficient lubricant which compromises the reliability and tribological characteristics of an internal combustion engine. Hydrocarbon and other hazardous emissions caused by lubricants are merely dependent upon the composition of chemical compounds in lubricants which requires wide attention from automotive manufacturers. It is important to recognize that the research has a significant level of competency in understanding the chemistry of automotive fuels and lubricants regarding emission, which is important when dealing with ways to regulate depletion of the environment in internal combustion engines and the demands to optimize strict fuel and lubricant specifications in automotive industries.

3.Literature Review

3.1 Automotive emissions

Global warming and human health challenges force researchers to develop new ways to effectively control air pollution. Road transport is primarily responsible for environmental and health risks because fossil fuel-powered internal combustion engines emit harmful gaseous pollutants and particulate matter (C.Dora, 2000).In addition to tailpipe emissions which has been controlled by regulations and introduction of innovations through engineering and technology, emissions from the transport sector also include evaporative and abrasive emissions from tyre wear, clutches, brake pads, and road surface (Ankur Sinha, 2020).

Vehicle emissions are a significant source of gaseous and particulate pollutants, particularly urban air pollution, such as carbon monoxide (CO), nitrogen oxides (NOx), hydrocarbons (HC), and particulate matter (PM) (P. Krecl, 2018) (M.D. Geller, 2006). Mehraj et al carried out a study to understand the level of (SO₂) and (NOX) pollutants by investigating closely meteorological parameters deteriorating air quality. Increased concentrations are guaranteed, and this may be due to exhaust fumes from many types of automobiles that need an essential analysis (BHAT, 2021).The study also found that pollutants are very high in winter and summer compared to monsoon and post-monsoon periods which outlines reaction of pollutants with the rainwater. Oxides of sulphur i.e., sulphur dioxide (SO₂) and sulphur trioxide (SO₃), represented as (SOX), hydrogen Sulphide (H₂S), carbonyl sulphide (COS), carbon disulphide (CS₂), dimethyl sulphide [C(CH₃)₂S] and Sulphates (SO₄) are the most intimidating air pollutants (Swami, 2018).

Depending on their chemical properties, solid contaminants are divided into two types

- 1) Inorganic particulate matter (I.P.M) - These are added to metal oxides, sulphides, and carbonates formed due to the composition of the metal in fuels. example: lead, sulphur. Lead halides are emitted from the exhaust system and condense to form particulates. Aerosol mists are formed due to the condensation of atmospheric SO₂. (Swami, 2018)

There are two possible sources of sulphur which need investigation during the operation of ICE.

- a) Concentration of Sulphur in fuel
 - b) Oxidation of Sulphur in lubricants due to combustion of lubricant.
- 2) Organic particulate matter (O.P.M) - These are polycyclic aromatic hydrocarbons chrysene, benzo-fluoranthene, benzopyrene, and benzdine (carcinogenic in nature). The sources of OPM are automotive fuels and lubricants (Swami, 2018)

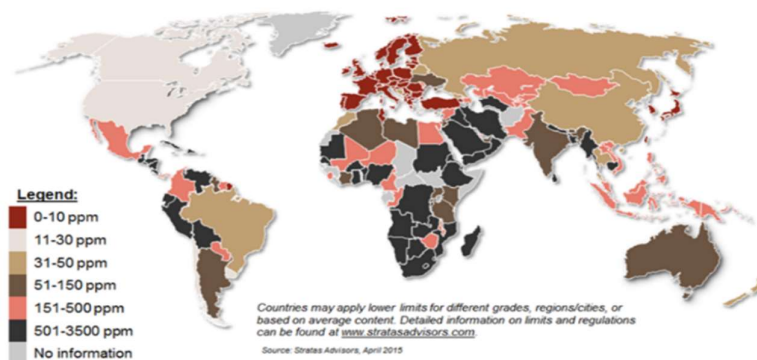


Figure 1 : Global sulphur limit in gasoline (Stratas Advisors, 2015)

3.2 Pre-ignition in Internal combustion engines

The occurrence of abnormal combustion events that lead to high peak pressures and severe knock can be considered one of the main challenges for modern turbocharged and direct injection gasoline engines. These combustion abnormalities have been referred to as Stochastic Pre-Ignition (SPI) or Low-Speed Pre-Ignition (LSPI) (Patrick Haenel, 2019). Knock, these strong impacts cause strong pressure drops, which in some cases can destroy the stock engine (Passow, 2017).

Due to the low-temperature chemistry, low-octane fuels can exhibit significant heat build-up before ignition e.g.: Low-Temperature Heat Release (LTHR) and Intermediate Temperature Heat Release (ITHR). Pre-ignition heat generation affects ignition timing and combustion phase, which are important factors in optimizing the performance of a Gasoline Compression Ignition (GCI) engine (M. AlAbbad, 2019).

Fuel	PR	$u_{f,max}$, (m/s)	RON	MON	Source of RON/MON
1-pentene	-21	0.845	90.9	77.1	API
1-hexene	-20	0.835	76.4	74	API
cyclohexane	0	0.78	83	78	API
ethyl benzene	18	0.77	109	97.9	rated
cumene (isopropylbenzene)	19	0.765	113.1	99.3	API
benzene	26	0.84	105	97	API
2-methylbutene	50	0.71	98	82	API
cyclopentane	70	0.782	102.8	85.7	rated
isopentane	75	0.662	93.5	93	API
toluene	93	0.68	117	102	ASTM
p-xylene	95	0.615	113	100.6	rated
isooctane	100	0.667	100	100	ASTM
o-xylene	120	0.615	105.4	88.8	rated
ethanol	-28	0.87	110	91	Thring ³²
m-xylene	125	0.56	117	101.3	rated

Figure 2 :Pre ignition rating, laminar flame speed, MON, RON of different hydrocarbons/fuels (Farrell, 2004)

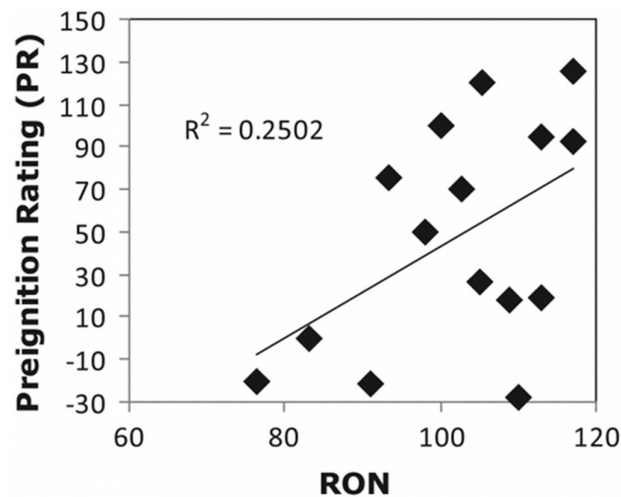
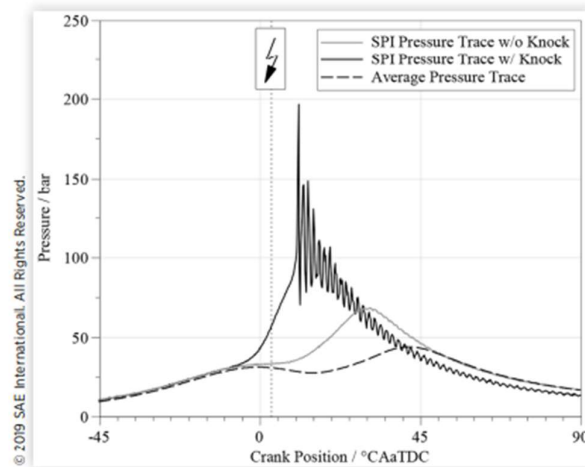


Figure 3:Pre-ignition rating as a function of RON in automotive fuels (Farrell, 2004)

In summary, recent work on pre-ignition in internal combustion engines leading to intense knock, suggesting that pre-ignition is characterized by (Bradley, 2012)

- a) A gas-phase preignition that occurs spasmodically at higher pressure and temperature
- b) An autoignition at even higher pressures and temperatures could produce a very severe, damaging knock
- c) Spraying from injectors strips surrounding surfaces of lubricant
- d) Due to the lower ignition temperature, the lubricating oil and fuel mix with the hot charge and auto-ignite
- e) Gases and particles remaining in the system causing hot spots
- f) Differing pre-ignition ratings for different fuels.



Graph 1 : Examples for a characteristic SPI event with Mega Knock and without Mega Knock (Patrick Haenel, 2019)

The phenomenon of strong shock with pressure swings are clearly visible in the combustion cycle with an early start of combustion. The non-knocking cycle - except for the unintentional early onset of combustion is barely noticeable because it differs from the normal combustion cycle in terms of the pressure rise rate and peak pressure level. To further analyse the combustion properties of the SPI event, Figure 10 shows a comparison of the pressure marks resulting from the spark clearance with the SPI combustion cycle.

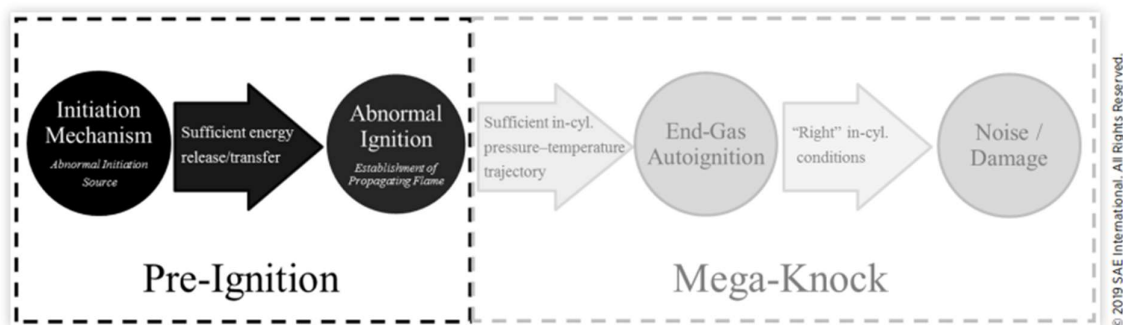
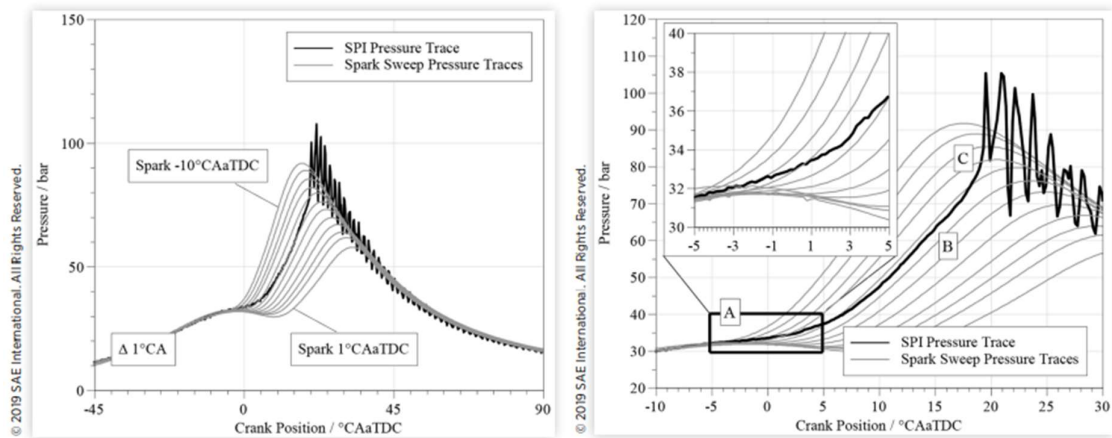


Figure 4 : Definition of Stochastic Pre ignition mechanism steps (Yu, 2017)



Graph 2 : Averaged spark sweep pressure traces with high-octane fuel compared to an example for an SPI pressure trace(left), Pressure rises analysis of the SPI initiation phase (A), deflagration (B) and knock onset phase (C) for an SPI cycle compared to pressure traces resulting from a high-octane race fuel spark sweep (right) (Patrick Haenel, 2019)

The proposed SPI mechanism separates the influencing factors into those that affect SPI initiation, flame development and propagation, and self-ignition (mega-detonation) of the final gas. The following properties can be identified as follows:

- SPI Initiation:** The process of oil-fuelled droplets or unburned hydrocarbon gathering on the lubricant layer of the cylinder wall and in the crevice volume of the top ring land area of an engine can cause them to flake, splatter, blast, evaporate, or blow into the combustion chamber during the gas exchange, expansion, and compression strokes.
- SPI Development:** Once these droplets mix with oxygen, they evaporate and self-ignite, resulting in flame initiation and deflagration. Deposit-borne preignitions can occur in the fuel around flaming deposits either directly or after a heat cycle.
- End-Gas Auto-Ignition:** The early ignition and ensuing deflagration due to SPI events can then lead to strong engine knock (Mega-Knock) according to the same auto-ignition mechanisms that lead to spark knock.

3.3 Research octane number on Automotive fuel performance and emissions

Research Octane Number (RON) is the standard measure of the performance and quality of the fuel. The octane number of fuel has an impact on engine performance and emissions (Sakaguchi, 2000). Higher octane numbers indicate that the fuel can withstand higher compression without premature knocking or detonation. Gasoline formulations are indexed primarily in terms of octane number. Fuel properties of different RON numbers are as follows.

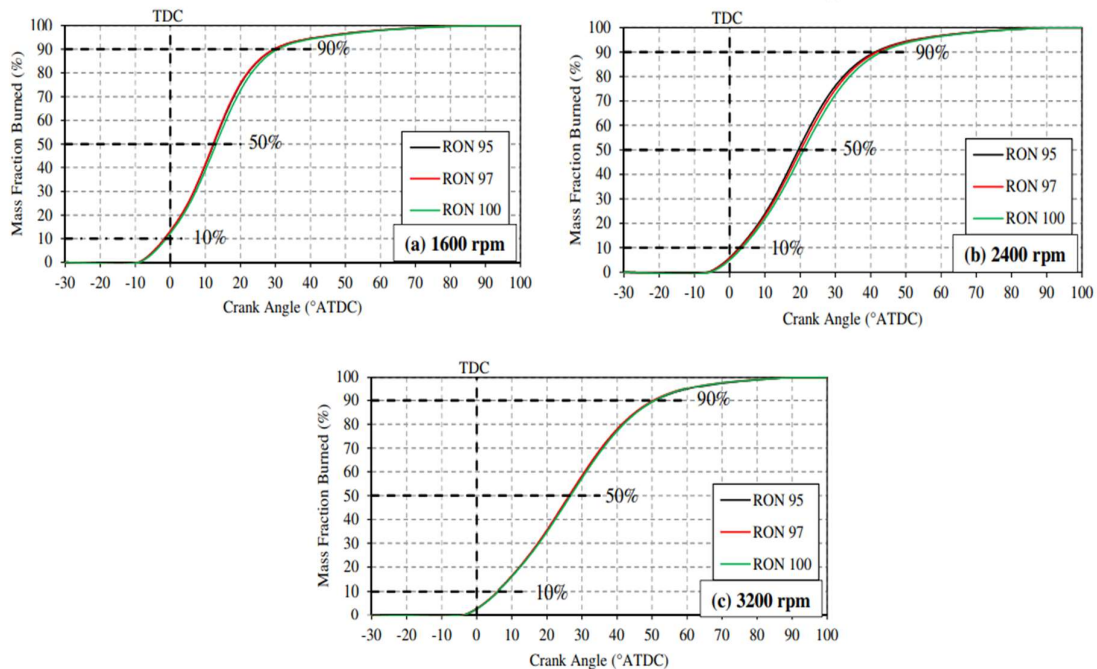
Fuel properties

Fuel type	Octane level	Density @ 25°C, g/ml	Heating value (MJ/kg)
RON 95	95.3	0.7400	43.304
RON 97	98.6	0.7377	43.961
RON 100	100.1	0.7706	43.989

Figure 5: Fuel properties of different Gasoline RON (Ghanaati, 2017)

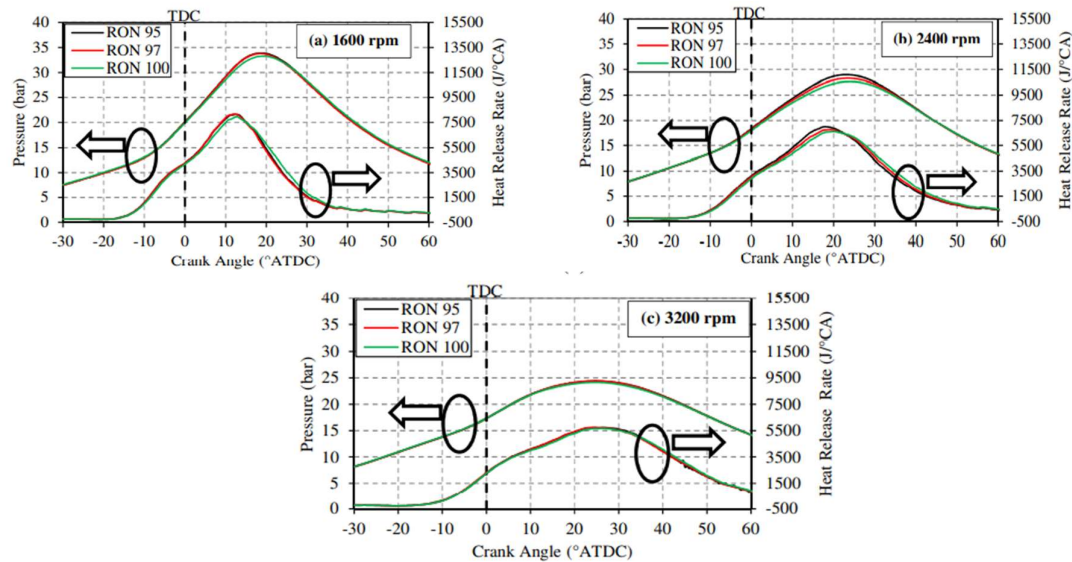
Yew Heng et al experimentally investigated the influence of RON on the performance, combustion, and emission characteristics of spark ignition engines and concluded the following (Yew Heng Teoh, 2018)

- 1) RON 100 has a lower Brake Specific Fuel Combustion compared to RON 95 and RON 97 fuels across all engine speeds. This effectively concludes that the higher the Ron higher the fuel efficiency.
- 2) From a performance perspective, braking torque and braking power for 100-octane fuel are consistently higher than for 95 and 97-octane fuels at all engine speeds.
- 3) CO₂ and NO_x were lower at 95 octanes because more hydrocarbon bonds and higher volatility result in higher flame temperatures in higher octane fuels.
- 4) RON 97 fuel resulted in complete early combustion than RON 95 and RON 100 fuel.



Graph 3: Mass fraction burned as a function of the crank angle at (a) 1600 (b)2400 (c)3200 rpm (Yew Heng Teoh, 2018)

Taken together, the experimental results suggest that the moderate octane fuel of RON 97 is a good candidate to run, given that it focuses on a balance between performance, exhaust characteristics, and combustion. (Akihama, 2004) investigated with a higher compression ratio of 13:1 in spark-ignition direct injection (SIDI) engines and found that a higher RON fuel results in a significant increase in torque under high loads.



Graph 4: Pressure and Heat released rate of different RON fuels at (a)1600 (b)2400 and (c)3200 rpm respectively (Yew Heng Teoh, 2018)

(Mohamad, 2014) conducted experimental studies on a low compression engine (11: 1) with 95 and 97 RON gasoline and found that 95 RON gasoline produced approximately 4.4% more braking torque, braking power, and average effective braking pressure (BMEP) than 97 RON gasoline. For BSFC with 95 RON gasoline, the results showed that 95 RON gasoline was 2.3% higher than 97 RON gasoline. But the 95 RON is 2.3% more fuel efficient when the engine is running at higher speeds and loads.

Emissions of unburned hydrocarbons in exhaust gases are primarily related to the presence of unburned fuel, incomplete combustion, and lubricating engine oil in the combustion chamber (Ghanaati, 2017) . RON 91 gasoline produces higher HC emissions than RON 95 gasoline (Binjuwair, 2015). (Alahmer, 2016) suggest that using a higher quality fuel than the engine requires can have a negative impact on engine performance after testing RON 90,95 duel in a single-cylinder gasoline engine at variable loads.

3.4 Molecular Diffusion and Crevice Volume

When the piston reaches TDC, the crevice volume may take up to 10-20% of the combustion chamber volume of an optical engine. This crevice can contain up to half of the charge (by mass) if these gases remain unburned (Peter Janas, 2015). The crevice volume determines the way heat is released from within the cylinder and therefore the pressure inside the cylinder. These optically accessible engines may have crevice volumes fourfold as large as production engines (Köhler, 2011). Crevice losses are combustion losses that occur mainly because natural gas–air mixtures trapped in the top land crevice regions during the compression stroke do not participate in the combustion (D.E. Nieman, 2019).

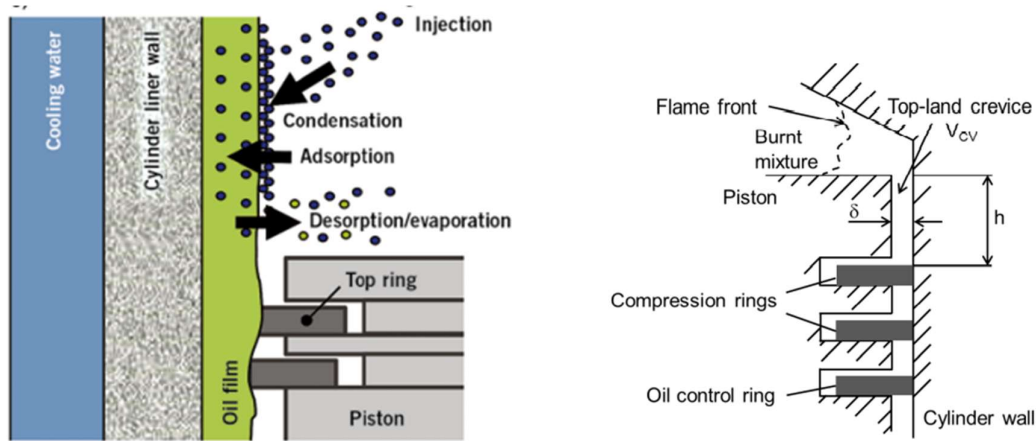


Figure 2: Geometry of the piston and top land crevice with different mechanisms on emissions (API)

Port Fuel Injection (PFI) and Direct Injection (DI) engines can both develop a thin fuel film (micron level) on top of the original oil film (submicron level) after being sprayed with fuel (Qin Zhang, 2018). The ignition of a rogue droplet composed of lubricant oil and fuel is often cited as one of the mechanisms responsible for pre-ignition. A mixture is formed when fuel spray and oil on a cylinder liner interact. Recent studies found that molecular diffusion of fuel with the oil layer in cylinder liner leads to 6 percent of unburned charge accumulated in crevice volume during a single power stroke (Eshan Singh, 2018). By dilution, more unburned gas is pushed into the crevice during combustion.

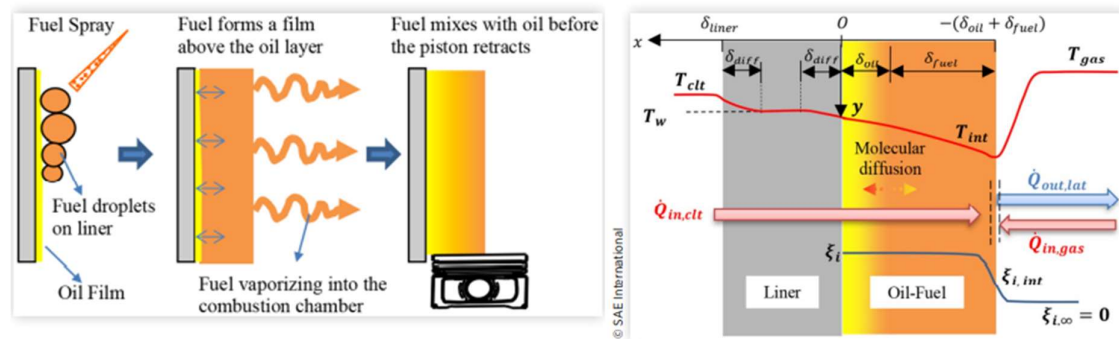


Figure 6: Formulation of fuel-oil film on liner (molecular diffusion) (left) and schematics of heat and mass transfer (right)

(Yu, 2002) concluded that dilution of fuel and oil mixtures leads to

- 1) vaporization of fuel on the surface of the oil
- 2) desorption of fuel from the oil film

The combination of these two mechanisms (vaporization and mass diffusion) makes the overall process of fuel-oil interaction on the liner.

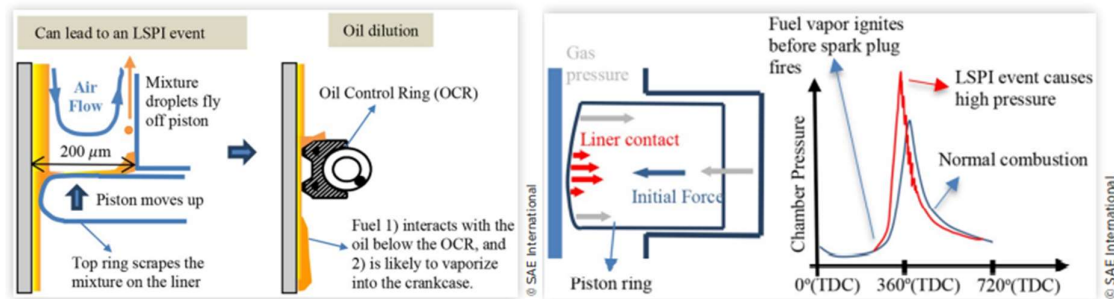
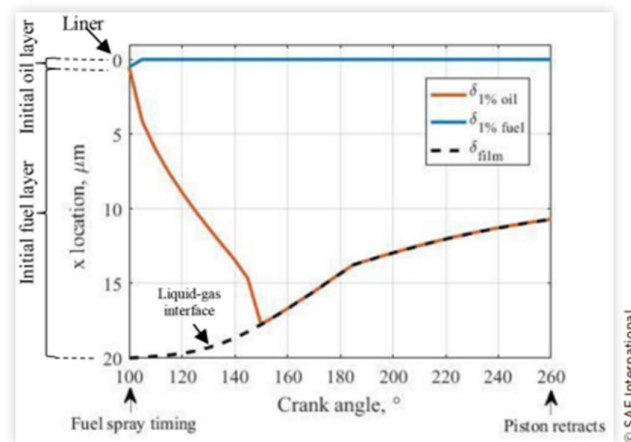


Figure 7: Consequences of fuel-oil dilution and ring-liner contact force (Liu, 2017) (Tian, 2017)



Graph 5 : Location of the penetrating boundary layer for oil and fuel (Tian, 2017)

After numerical simulations (Tian, 2017) concluded that

1. Due to the occupancy of the light species (fuel and oil), the heavy species tend to navigate the movement from the cylinder liner interface resulting in the vaporization at the interface.
2. A considerable amount of the remaining liquid film is residual fuel, which can dilute the local lubricant, increasing the friction between the piston ring and cylinder line contact.
3. Vaporization of the fuel-oil mixture is aided by a high liner temperature that can rise above the flash point of lubricant. Meanwhile, cold starting conditions have the opposite effect, where the mixture is more likely to remain on the liner as the piston retracts.

3.5 Lubrication and Exhaust Gas Recirculation

Engine oil is used in all internal combustion engines and mechanical systems to prevent component corrosion and abrasive wear associated with prolonged contact (Mohsen Tahmasebi Sulgani, 2019). Friction is a prevalent energy loss concern in standard combustion engines and several other industrial processes, and it impedes such mechanisms by generating excess heat, which must therefore be removed via cooling systems, resulting in a significant net power loss (Esquivel-Gaon, 2018). The friction force is determined by the viscosity of the oil and the shearing of the oil film (Taylor, 1993).

Each internal combustion engine component's lubrication regime or mode of lubrication is determined by the lubricant oil viscosity and the working conditions of the tribological pairs in each engine. Due to the variability of these operating conditions, engine components can experience all lubrication schemes in one cycle.

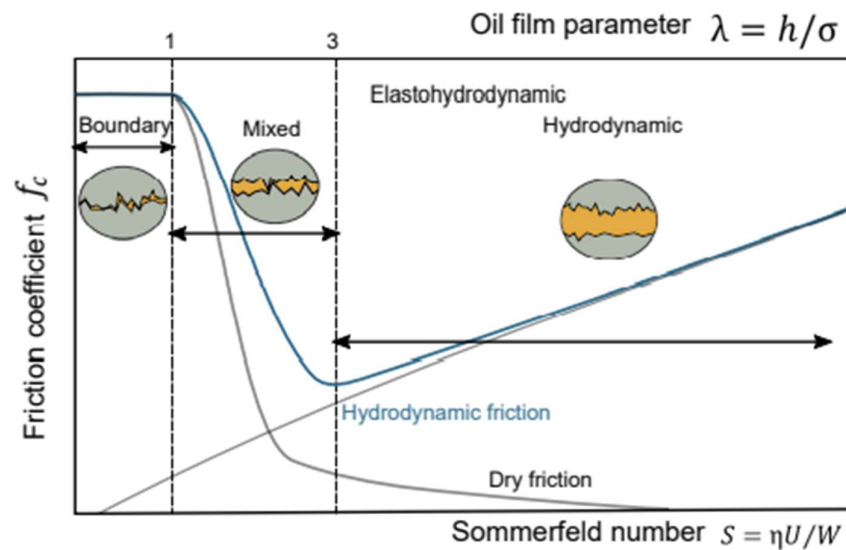


Figure 8: Stribeck curve as a function of Sommerfeld number S and the oil film Parameter (H , 2010)

The lubrication regime is characterized into 4 types

1. Hydrodynamic Lubrication: With this lubrication method, the thickness of the oil film is thick enough to avoid direct contact between the moving surfaces.
2. Mixed Lubrication: This regime is a transition between boundary and hydrodynamic lubrication conditions where there is asperity contact with lubrication happens.
3. Boundary Lubrication: The regime in which asperity contact appears between surfaces in relative motion.
4. Electrohydrodynamic Lubrication: Application of regime at higher loads with a fluid thick film of higher viscosity.

Wang and Stone (X. Wang, 2008) concluded that IC engines emit HC and NOx primarily because of their cylinder wall temperatures, which play a significant role in these emissions. A higher liner temperature tends to reduce HC emissions, while a higher liner temperature tends to reduce NOx emissions. As a result, an optimal liner temperature should be determined based on how HC and NOx emissions are to be balanced in composition

Piston-cylinder liner (%)	Journal bearings (%)	Valve train(%)	References
63%	19%	8%	Hoshi
40-50%	20-30%	7-15%	Taraza
38-68%	20-44%	3-34%	Holmberg
40-55%	20-30%	7-15%	James
45-55%	20-40%	7-17%	Holmberg

Table 1: contribution of tribological pairs on Frictional losses. (M., 1984) (Taraza D., 2000) (Holmberg K., 2012) (Holmberg K., 2014) (J., 2012)

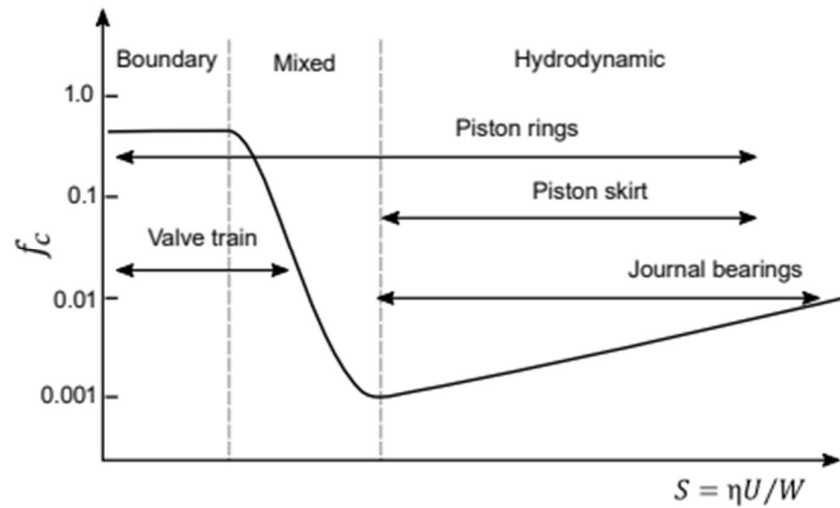


Figure 9: Lubrication regimes of tribological pairs in ICE (L, 2004)

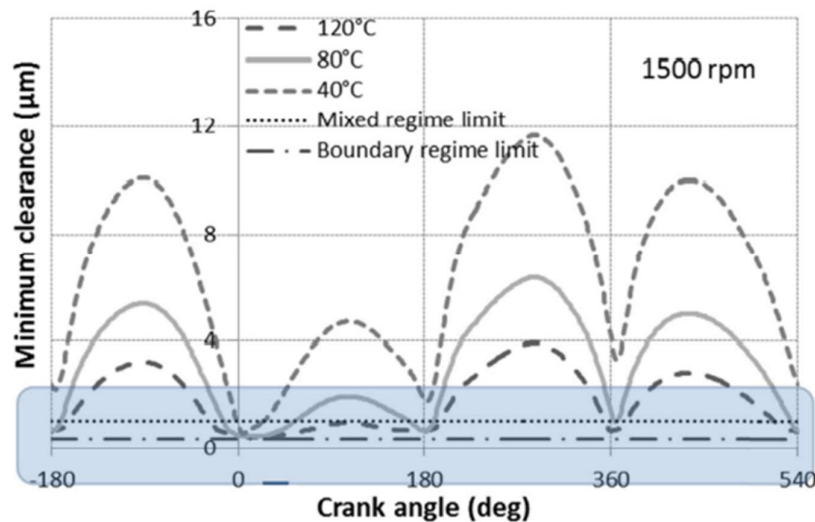


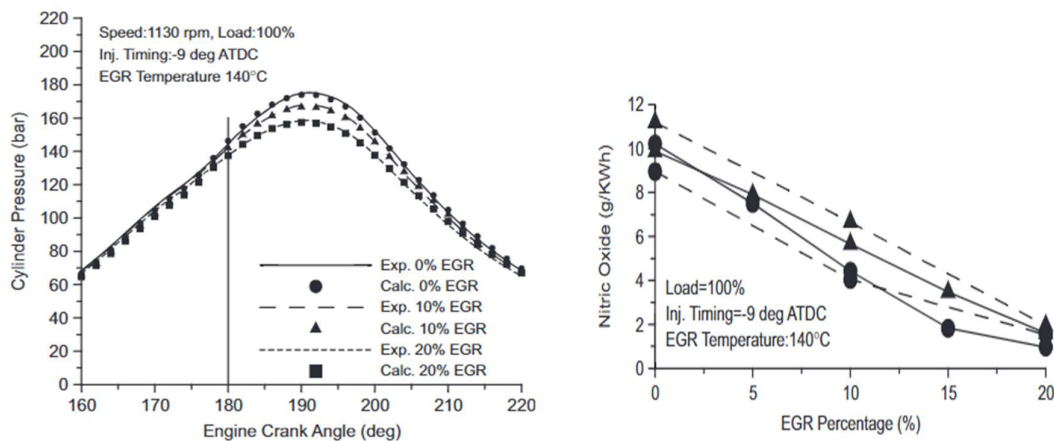
Figure 10: Minimum oil thickness of piston ring as a function of cylinder liner temperature (Rahmani R., 2017)

Table 1 outlines the energy losses in tribological pairs in an ICE. Engine oil is the composition of base oil and additives where the base is mineral or blend with synthetic and purely synthetic lubricant. Polealphaolefins and diesters are widely used in the formulation of multigrade oil with distinctive properties. Base oil is characterized into Group I, II, III, IV, and V.

Oil designation	R	C1	C2	C3	C4	C5
SAE grade	10W30	10W40	10W30	5W30	5W30	5W20
KV@40°C [cSt]	75.23	86.71	56.45	70.60	70.15	43.72
KV@100°C [cSt]	12.05	13.34	9.607	11.85	12.08	7.992
HTHS@150°C [cP]	3.67	3.85	3.17	3.61	3.61	2.7
SAPS level	Medium	Medium	Medium	Low	Medium	Medium
API base oil group	III	III	III	III + IV	III	III
API category	CK-4	CI-4	FA-4	CJ-4	CK-4	-
Friction modifier	[-]	[-]	[-]	Mo	Mo	[-]

Table 2: Characteristics of different grades of engine oils (Moncayo, 2021)

More chemistry is added to improve the performance of lubricants by adding additives. The different types of additives are detergents, dispersants, Antioxidant, Anti-wear, Viscosity Index improvers, and Friction Modifiers.



Graph 6 : Various experimental and calculated cylinder pressure diagrams for different EGR rates (D.T.Hountalasa, 2008)

During diesel engine development, controlling and reducing pollutant emissions to regulatory levels is one of the most challenging problems engineers and manufacturers face (Heywood, 1988). (J.L. Lazaro, 2002) states that the well-known and established method for reducing NO_x emissions is recirculating exhaust gases into the engine intake system. (D.T.Hountalasa, 2008) demonstrated that EGR temperatures have a detrimental effect on brake engine efficiency, peak combustion pressure, and soot emissions.

4. Methodology

The research is broadly divided into two phases. Firstly, the fuel characteristics of research octane number in gasoline are studied using Ricardo WAVE on a Ford Sigma combustion model. Secondly, injection strategies and the influence of EGR on combustion of the diesel is studied using Ansys FORTE R2 combustion modelling.

4.1 1-Dimensional combustion modelling analysis into Research Octane Number (RON) and Air Fuel Ratio of fuel mixtures influence on engine performance and emissions in a gasoline powertrain.

4.1.1 Software

The proposed study on the effects of RON in powertrain performance and emissions are done using a computational simulation environment. In this simulation, Ricardo WAVE software is used to simulate intake, combustion, and exhaust configurations from an initial concept to final design virtually, enabling performance and acoustic analysis. The advantage of using simulation tools like WAVE is that the component can be designed and tested in many different iterations without undergoing physical development or testing. Simulating a real engine is impossible without a precise computational simulation model (Shubham Jadhav, 2020).

During the combustion analysis, Wave simulation software solves Navier-Stokes equations in one dimension, including sub models for combustion and emissions, governing mass, momentum, and energy transfer within compressible gas flows. As part of an engine design process, wave can be used to improve combustion characteristics such as volumetric efficiency, improve and design complex boosting systems, analyse and improve transient response.

With its advanced combustion models for diesel and spark-igniting engines, WAVE is an ideal tool for investigating combustion and emissions. It also contains secondary models specifically designed to study direct injection, turbulent flame propagation, and engine exhaust emissions, which is the first step in research into automotive fuel and its impact on engine performance and emissions. Furthermore, wave includes a suite of exhaust aftertreatment models, which provide chemical kinetic simulations of advanced emission control methods.

4.1.2 Development of Ford Sigma powertrain model in Ricardo wave

4.1.2.1 Powertrain Design

The fundamental components of the WAVE geometry include a series of flow element ducts in intake and exhaust with respect to a 4-cylinder model that navigates the intake air to four throttle bodies of each cylinder followed by exhaust ducts to a muffler and towards outlet or atmosphere. Each component is split constructed to direct the flow of inlet air, exhaust mixture by a series of Y junctions. Apart from the flow elements in the geometry an engine block is attached to four cylinders as a mechanical element that gives necessary analysis to powertrain characteristics using WAVE Post. The design of the developed model is shown in figure 11.

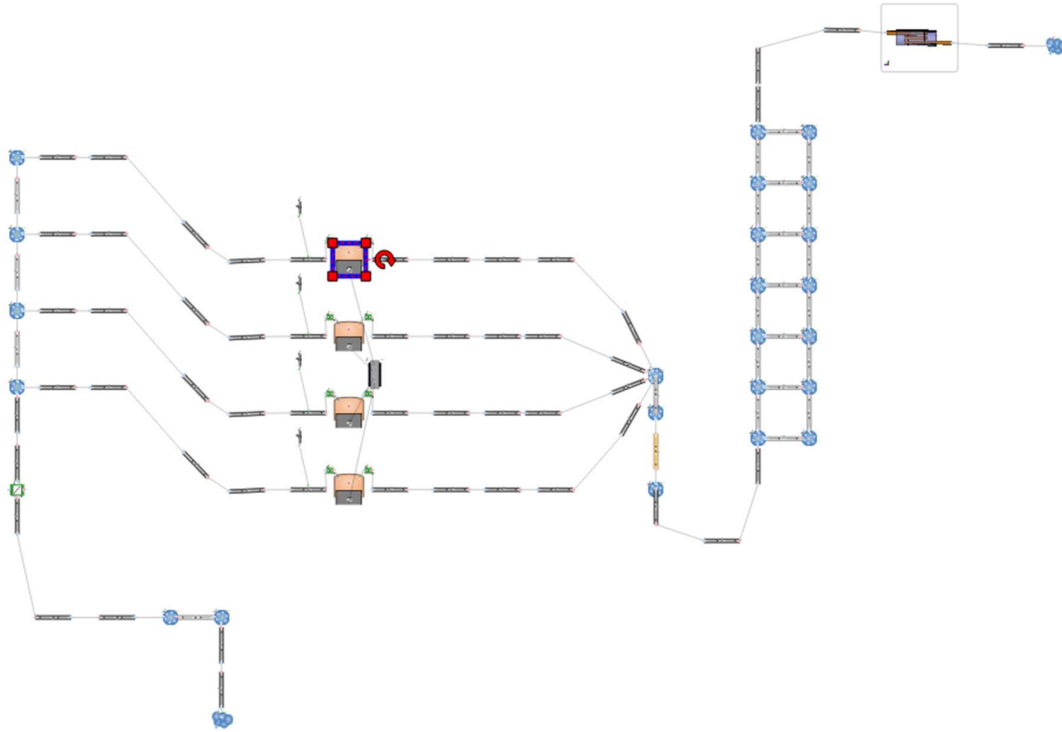


Figure 11 : Developed Ricardo WAVE combustion model of Ford Sigma geometry

4.1.2.2 Design parameters of Ford Sigma

4.1.2.2.1 Intake System

The intake system of the Ford Sigma geometry is constructed using a series of duct flow elements. A total number of 31 ducts of constant and variable design parameters constitutes the intake system. The design parameters of the standard ford sigma geometry elements are mentioned in table 3 and table 4 respectively. Table 3 describes about the design parameters of the ducts before throttle body whereas table 4 describes about the design parameters of flow elements in throttle body before cylinder. According to the simulation the wall friction and wall heat transfer are considered as constants. The initial ambient air conditions are constant throughout the inlet components where pressure is being assumed as 1 bar and wall temperature is 300 kelvins. As the throttle body duct elements nearby to the cylinder (24 to 31), the wall friction is considered as zero meanwhile wall heat transfer is considered as 1.5 due to combustion in cylinder. As the ducts 28 to 31 are 4 inlet pipes attached to the cylinder their wall temperature is considered as 310 kelvins.

Ambient Intake Flow Elements	DUCT 1	DUCT 2	DUCT 3	DUCT 4	DUCT 5	DUCT 6	DUCT 7	DUCT 8	DUCT 9	DUCT 10	DUCT 11
Condition	Intake	Intake	Intake	Intake	Intake	Intake	Intake	Intake	Intake	Intake	Intake
Left/Outlet Diameter (mm)	70	50	140.37	60	60	60	60	60	110	110	110
Right/Outlet Diameter(mm)	50	50	140.37	60	60	60	60	60	110	110	110
Overall Length (mm)	50	100	0	250	150	50	50	150	0	0	0
Discretization Length (mm)	35	35	35	35	35	35	35	35	35	35	35
Bend Angle (deg)	0	0	0	0	90	0	0	0	0	0	0
Wall Temperature (K)	300	300	300	300	300	300	300	300	300	300	300
Pressure (bar)	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
Wall friction	1	1	1	1	1	1	1	1	1	1	1
Wall Heat Transfer	1	1	1	1	1	1	1	1	1	1	1

Table 3: Intake design parameters of the Ford Sigma combustion model before throttle body

Throttle body Flow Elements	DUCT 12 13 14 15	DUCT 16 17 18 19	DUCT 20 21 22 23	DUCT24 25 26 27	DUCT 28 29 30 31
Condition	Intake	Intake	Intake	Intake	Intake
Left/Outlet Diameter (mm)	40	40	40	40	35
Right/Outlet Diameter(mm)	40	40	40	35	35
Overall Length (mm)	100	100	150	100	50
Discretization Length (mm)	35	35	35	35	35
Bend Angle (deg)	0	45	0	0	0
Wall Temperature (K)	300	300	300	300	310
Pressure (bar)	1.5	1.5	1.5	1.5	1.5
Wall friction	1	1	1	0	0
Wall Heat Transfer	1	1	1	1.5	1.5

Table 4:Throttle body flow elements design parameters of Ford Sigma combustion model

Y-junction in the geometry is the conjugation of 2 or more pipes where the ambient mixture is split directed to multiple lines. Y-junction 1,2 represent the ambient inlet ducts and have higher volume which reduces within duct in the direction of travel. Since the geometry of the powertrain contains 4 cylinders Y-junction 3 to 6 is split of inlet duct into 4 throttle bodies of each respective cylinder. Wall temperature, fluid pressure and fluid temperature in the Y-junction is considered the same as in inlet pipes. The geometry design parameters of Y-junction in the inlet system of Ford Sigma are mentioned on table 5.

Intake YJunction	1	2	3	4	5	6
Diameter (mm)	200	200	110	110	110	110
Volume (mm^3)	6000000	2000000	750000	750000	750000	750000
Inlet Duct	2	3	8	9	10	11
Outlet Duct	3	4	9,12	10,13	11,14	15
Opening angle X (deg)	180	180	90	90	90	90
Opening angle Y (deg)	90	90	180	180	180	180
Opening angle Z (deg)	90	90	90	90	90	90
Wall Temperature (K)	300	300	300	300	300	300
Expansion length (mm)	191	64	79	79	79	79
Expansion diameter(mm)	200	200	110	110	110	110
Wall Temperature (K)	300	300	300	300	300	300
Fluid Pressure(bar)	1.5	1.5	1.5	1.5	1.5	1.5
Fluid Temperature (K)	300	300	300	300	300	300

Table 5: Intake Y-junction design parameters of ford sigma combustion model

4.1.2.2.2 Cylinder Geometry

The powertrain parameters of ford sigma engine are mentioned on table 6. The compression ratio of the cylinder is considered as 11.16 as per the manufacturer data where clearance volume is calculated using the data provided. Similarly, all the cylinder geometry parameters are used as values mentioned in manufacturer data.

Cylinder	1	2	3	4
Bore (mm)	79	79	79	79
Stroke(mm)	81.4	81.4	81.4	81.4
Connecting rod length (mm)	137	137	137	137
Compression ratio	11.16	11.16	11.16	11.16
Fluid state Pressure (bar)	1.5	1.5	1.5	1.5
Fluid state Temperatue(K)	300	300	300	300

Table 6 : Cylinder geometry of Ford Sigma combustion model

4.1.2.2.3 Exhaust System

The exhaust system of the Ford Sigma geometry is constructed using a series of ducts and Y-junctions as like intake system. Exhaust system consists of 45 ducts of constant and variable dimensions between cylinder, catalytic converter, and muffler. The combustion turbulence thermodynamic conditions such as temperature and wall heat transfer have variable data that decreases throughout the direction of burned mixture outflow. Table 7 and 8 outlines the design parameters of ducts between cylinder, catalytic converter, and muffler.

Exhaust Flow Elements	DUCT 32 33 34 35	DUCT 36 37 38 39	DUCT 40 41 42 43	DUCT 44 45 46 47	DUCT 48 49 50 51	Duct 52
Condition	Pre Catalyst	Pre Catalyst	Pre Catalyst	Pre Catalyst	Pre Catalyst	Pre Catalyst
Left/Outlet Diameter (mm)	28	28	40	40	40	76
Right/Outlet Diameter(mm)	28	40	40	40	40	76
Overall Length (mm)	90	150	40	80	120	0
Discretization Length (mm)	40	40	40	40	40	40
Bend Angle (deg)	0	0	0	60	0	0
Wall Temperature (k)	400	400	750	750	750	750
Fluid Pressure (bar)	1.05	1.05	1.05	1.05	1.05	1.05
Fluid Temperature (K)	900	900	1100	1100	1100	1100
Wall friction	0	0	1	1	1	1
Wall Heat Transfer	1.5	1.5	1	1	1	1

Table 7: Exhaust flow elements design parameters of Ford Sigma combustion model between cylinder and catalytic converter

Exhaust Flow Elements	DUCT 53	DUCT 54	DUCT 55	DUCT 56 to 61	DUCT 62 to 67	DUCT 68 to 73	DUCT 74	DUCT 75	DUCT 76	DUCT 77
Condition	Pre Muffler	Pre Muffler	Pre Muffler	Pre Muffler	Pre Muffler	Pre Muffler	Pre Muffler	Pre Muffler	Pre Muffler	After Muffler
Left/Outlet Diameter (mm)	50	50	50	50	11.3	108.15	50	50	50	50
Right/Outlet Diameter(mm)	50	50	50	50	11.3	108.15	50	50	50	50
Overall Length (mm)	100	100	40	0	0	0	40	120	300	40
Discretization Length (mm)	40	40	40	40	40	40	40	40	40	40
Bend Angle (deg)	90	90	0	0	0	0	0	90	0	0
Wall Temperature (k)	800	800	600	700	600	600	600	600	600	600
Fluid Pressure (bar)	1.05	1.05	1.05	1.05	1.05	1.05	1.05	1.05	1.05	1.05
Fluid Temperature (K)	1000	1000	800	900	800	800	800	800	800	800
Wall friction	1	1	1	1	1	1	1	1	1	1
Wall Heat Transfer	1	1	1	1	1	1	1	1	1	1

Table 8: Exhaust flow element design parameters of Ford Sigma combustion model between catalytic converter and muffler

Y-junction is the centre of connection of 2 or more ducts where the burned fuel mixture is directed with a turbulence and have significant effects on improving the thermal efficiency and thermal management in the exhaust system. Y-junction in intake system is comparatively less and are used to diverge the flow of ambient air mixture into throttle body whereas a total number of 17 Y-junction are used to control the outlet turbulence of the burned mixture. Table 9 and 10 outlines the geometry or design parameters of the Y-Junction. Y-Junction number 7 consists of the highest number of 4 ducts which merge the burned mixture of all cylinders to a 4 by 1 ratio.

Exhaust YJunction	7	8	9	10	11	12	13	14	15	16
Diameter (mm)	80	90.2	76.4	50	50	50	50	50	50	50
Volume (mm ³)	268083	319000	229000	84150	84150	84150	84150	84150	84150	84150
Inlet Duct	48 to 51	52	Catalyst	55	56	57	58	59	60	61
Outlet Duct	52	Catalyst	53	56,62	57,63	58,64	59,65	60,66	61,67	68
Opening angle X (deg)	90	-90	90	90	90	90	90	90	90	68
Opening angle Y (deg)	90	0	180	180	180	180	180	180	180	180
Opening angle Z (deg)	180	90	90	90	90	90	90	90	90	90
Wall Temperature (K)	900	900	900	700	700	700	700	700	700	700
Expansion length (mm)	80	50	50	42.857	42.857	42.857	42.857	42.857	42.857	42.857
Expansion diameter(mm)	80	90.2	76.4	50	50	50	50	50	50	50
Fluid Pressure(bar)	1.05	1.05	1.05	1.05	1.05	1.05	1.05	1.05	1.05	1.05
Fluid Temperature (K)	1100	1100	1100	900	900	900	900	900	900	900

Table 9: Exhaust Y-junction design parameters of Ford Sigma combustion model

Exhaust YJunction	17	18	19	20	21	22	23
Diameter (mm)	108.15	108.15	108.15	108.15	108.15	108.15	108.15
Volume (mm ³)	393686.4	393686.4	393686.4	393686.4	393686.4	393686.4	393686.4
Inlet Duct	62	63	64	65	66	67	68
Outlet Duct	69	70	71	72	73	74	75
Opening angle X (deg)	180	180	180	180	180	180	180
Opening angle Y (deg)	-90	-90	-90	-90	-90	-90	-90
Opening angle Z (deg)	90	90	90	90	90	90	90
Wall Temperature (K)	600	600	600	600	600	600	600
Expansion length (mm)	34	34	34	34	34	34	34
Expansion diameter(mm)	17.32	17.32	17.32	17.32	17.32	17.32	17.32
Fluid Pressure(bar)	1.05	1.05	1.05	1.05	1.05	1.05	1.05
Fluid Temperature (K)	800	800	800	800	800	800	800

Table 10: Exhaust Y-junction design parameters of Ford Sigma combustion model

4.1.2.3 Simulation analysis case sets

The analysis is conducted on wide range of RPM from idle condition assumed as 1000 RPM to maximum power at 6000 RPM of the powertrain as mentioned in table 11. Ford Sigma manufacturer data obtained for simulation reference data was studied for a lambda of .9 and 1.1 is considered for all cases in the Ricardo WAVE. Case 6 at 6000 RPM is the analysis used for validation of the powertrain data and simulation data. Before The temperature of inlet and exhaust valves, cylinder head, liner and piston, catalytic converter is assumed to be increasing with reference to RPM. The mentioned data in case set is a reference for Ricardo wave prior to simulation where analysis was conducted at different RPM and wide range of study on fuel burn duration and air-fuel ratio was done to understand the powertrain characteristics with respect to engine performance and emissions. The research was done for an AFR of .9 and 1.1. This was assumed from a stoichiometric AFR as 14.7 since the fuels that are used for research have varying research octane number that can vary the stoichiometric mass of fuel in grams.

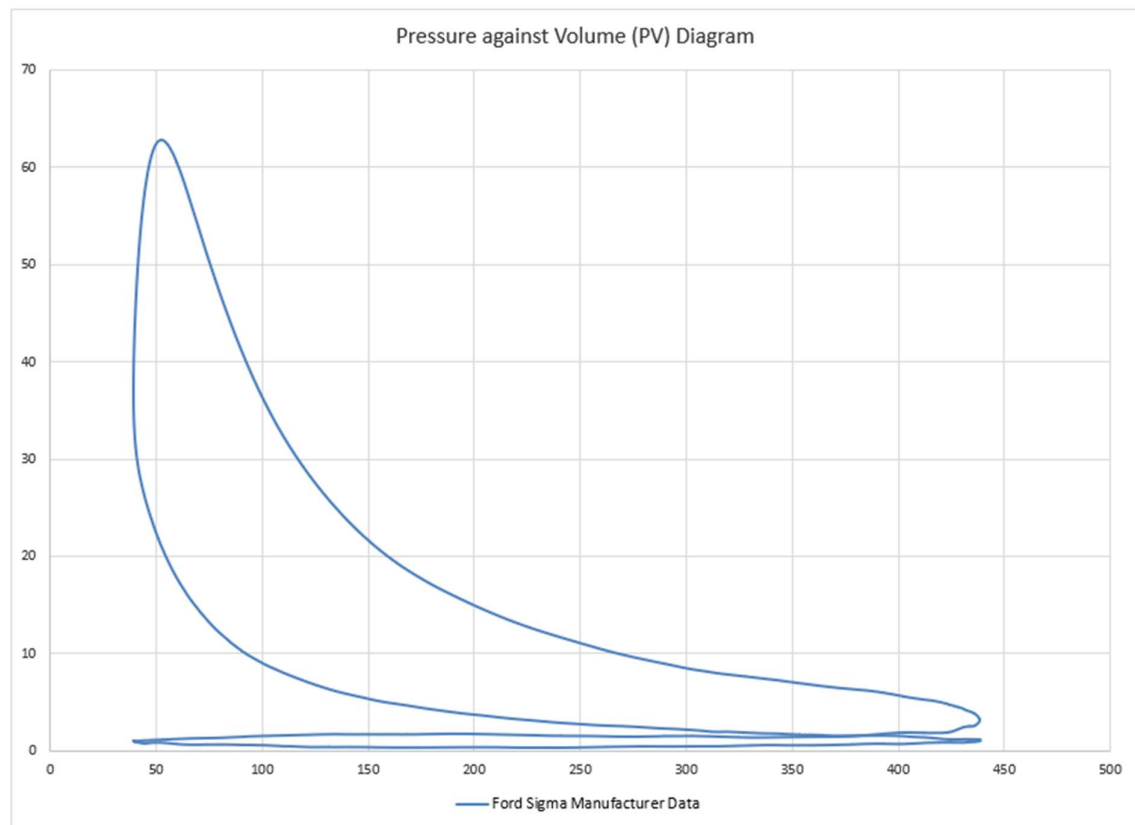
Simulation Case sets	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
RPM	1000	2000	3000	4000	5000	6000
Air-Fuel Ratio ($\lambda=0.9$)	13.23	13.23	13.23	13.23	13.23	13.23
Burn Duration (deg)	28	29	29.5	30	31	32
50 % Mass fraction Burn Duration (deg)	9	9	9	8.5	9.5	11
Catalytic Converter Temperature (K)	790	830	900	950	1025	1075
Exhaust Valve Temperature (K)	365	366	370	375	377	380
Cylinder Head Temperature (K)	550	580	595	620	630	635
Inlet Valve Temperature(K)	312	316	318	320	320	320
Liner Temperature (K)	540	570	580	595	600	616
Piston Temperature(K)	500	530	550	580	585	595
Throttle Angle (deg)	90	90	90	90	90	90

Table 11 : Pre evaluated combustion case sets of Ford Sigma combustion model strategies for Ricardo WAVE simulation

4.1.3 Validation of Ford Sigma manufacturer data with computational model

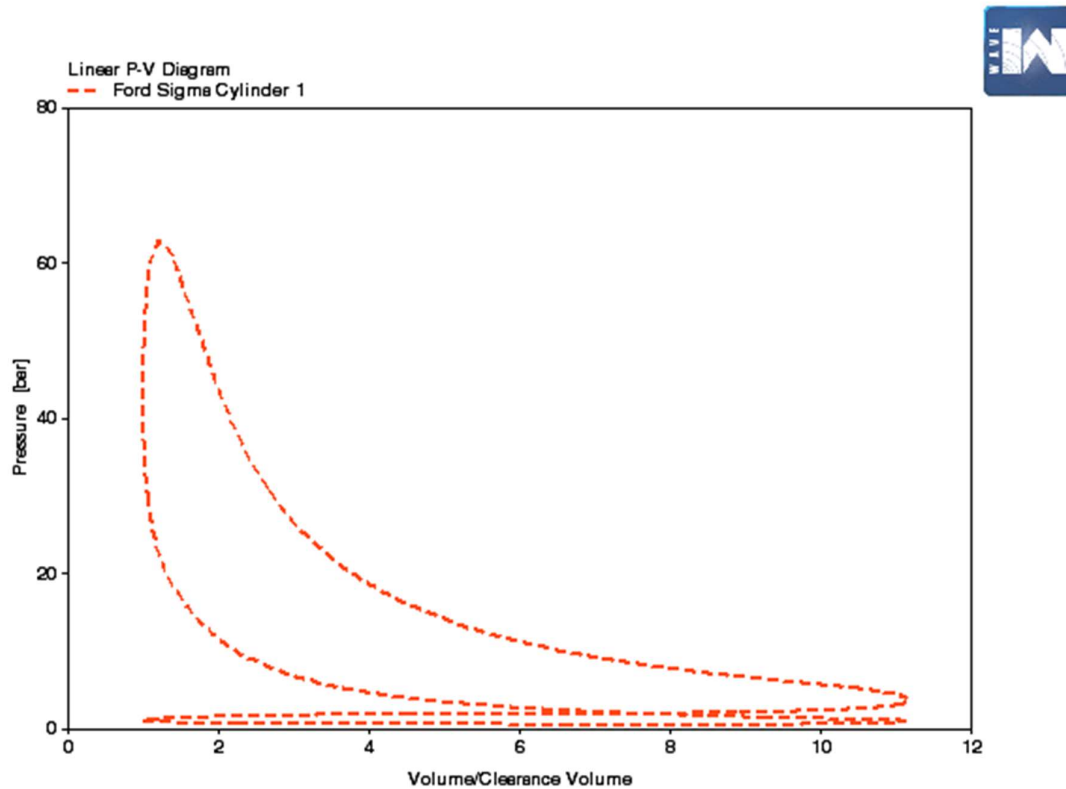
The pre mentioned case sets as in table 11 with the developed geometry of the ford sigma is run by Ricardo WAVE requesting the simulation output of pressure volume diagram and logarithmic pressure volume diagram to find the fit of manufacturer data with the simulation output in phase 1.

4.1.3.1 Linear Pressure-Volume (P-V) Diagram



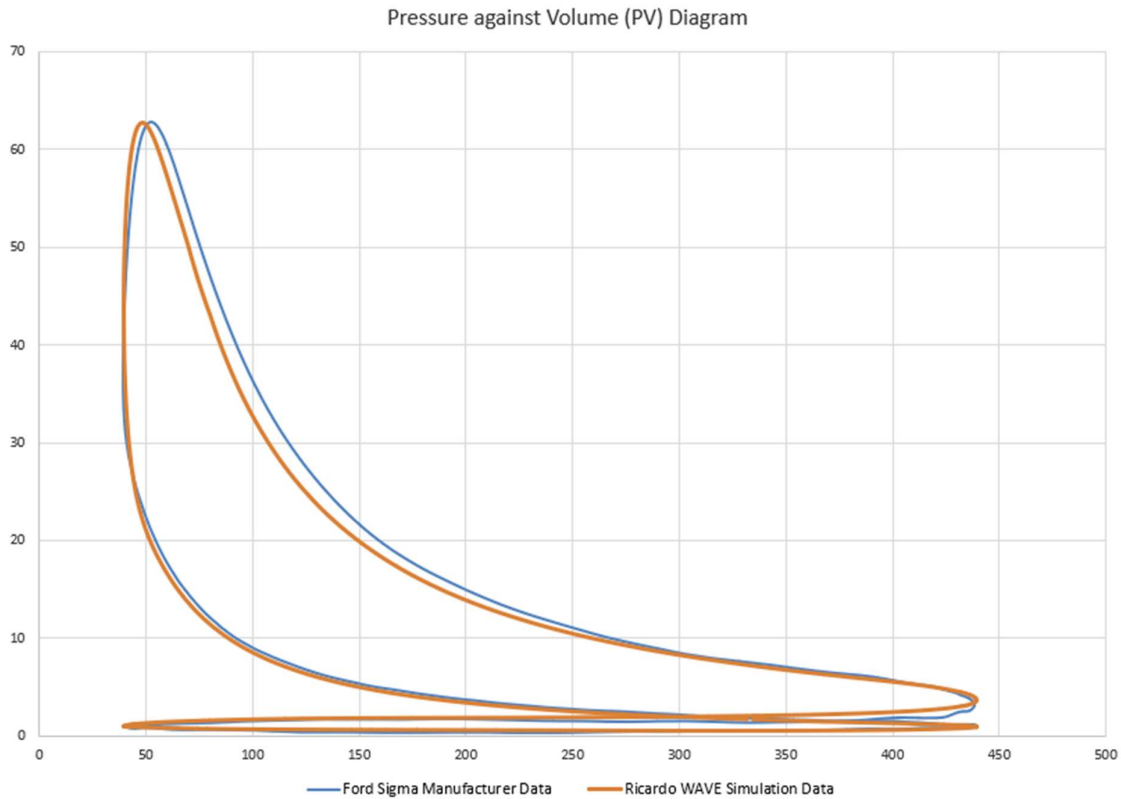
Graph 7 : P-V(pressure-Volume) diagram from the manufacturer data for 720-degree cycle

The manufacturer data of ford sigma for a 720-degree cycle is mentioned on graph 7. A maximum cylinder pressure of 62 bar is obtained from the graph at the end of compression stroke. X axis denotes the volume of cylinder whereas the Y axis denotes the cylinder pressure in bar.



Graph 8 : Linear P-V Diagram output of the ford sigma combustion model generated in Ricardo WAVE

The Ricardo WAVE output for the developed geometry is mentioned on graph 8. As the simulation is run with respect to the manufacturer data for a lambda of .9 at 6000 rpm, fuel burn duration of 29 degrees with 50 % burn duration of 10 degrees, a 720-degree cycle of pressure volume diagram outlines the peak pressure of 62 bar close to that of manufacturer data. X axis outlines the ratio of cylinder volume to clearance volume which is a fundamental pre-setting according to Ricardo WAVE output. The output data of pressure and volume are recorded to Microsoft excel. The ratio of cylinder volume and clearance volume is converted to cylinder volume to test the fit of simulated output with real manufacturer data. The merged pressure volume diagram of both manufacturer dyno data and simulated data are shown in graph 9.

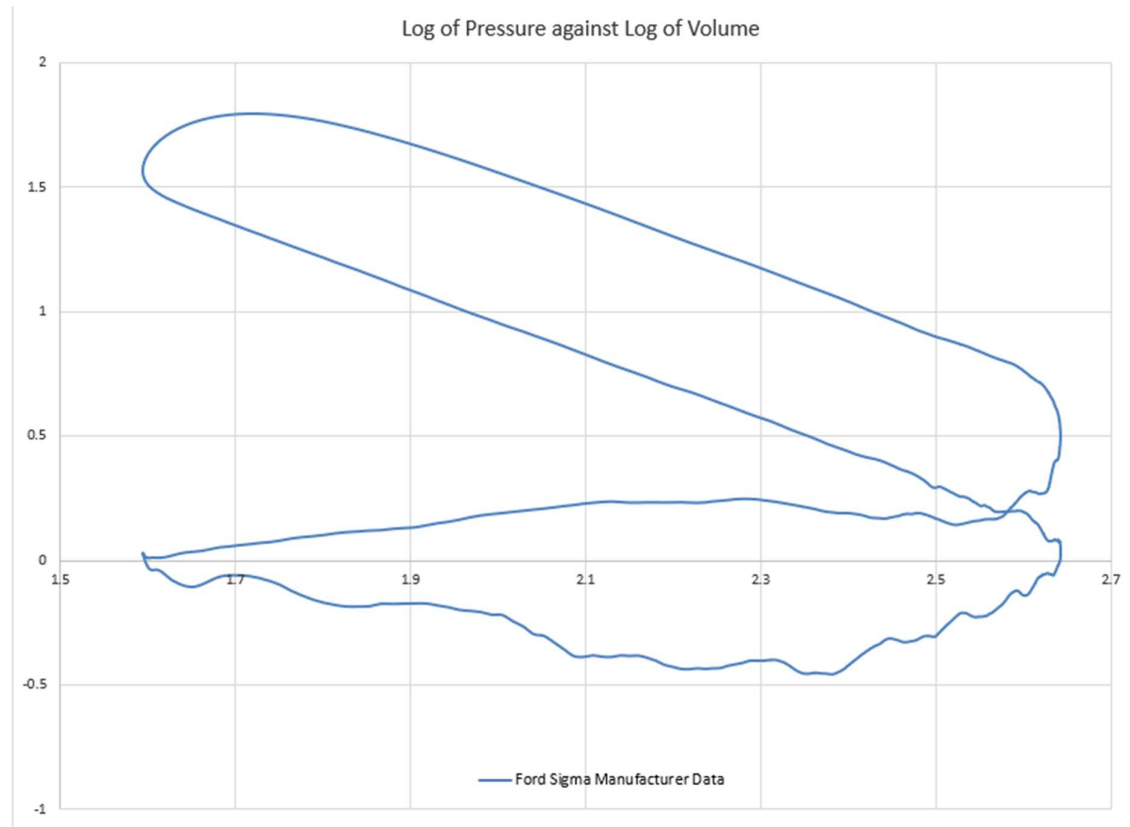


Graph 9: Merged P-V diagram (Pressure-Volume) of manufacturer data and Ricardo WAVE combustion model output

The observations from graph 9 outlines that the computational model have an identical fit throughout the cycle compared to manufacturer data. Key observation from pressure-volume diagram is that the fit of compression and combustion before piston reaches TDC is ideal while peak pressure obtained for the computational model is 62.72 bar to that of 62.782 bar of manufacturer data. It is noted that the peak pressure and pressure drop in computational model is slightly earliest for the case sets used for finest fit of combustion. More understanding about this phenomenon can be understood from chapter 4.1.3.2 which discuss about the validation of log pressure-volume diagram. Fuel properties can have a considerable influence in the combustion. The fuel used for computational model is indolene, an unleaded gasoline which has a Research Octane Number (RON) of 87 (Yu Cheng Liu, 2012). As mentioned in (Yu Cheng Liu, 2012), indolene is essential for engine testing since gasoline composition varies with regions and seasons. The intent of using indolene for engine testing is to replicate the emission characteristics of gasoline in the real world.

4.1.3.2 Log P-V (Pressure – Volume) Diagram

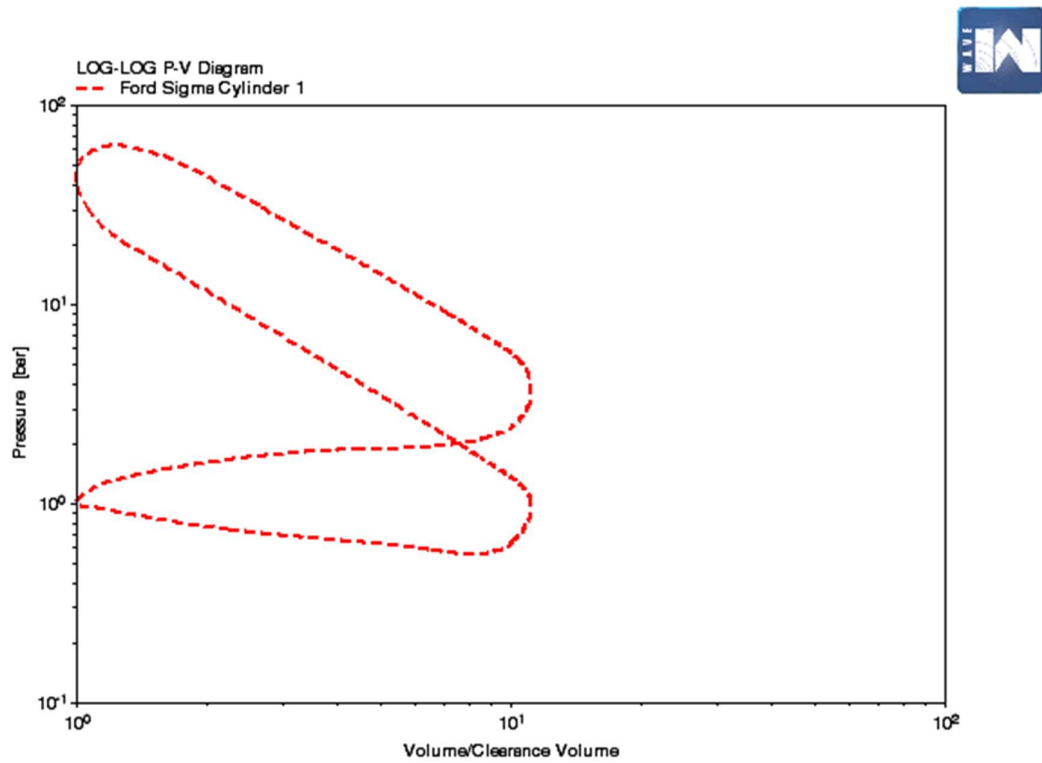
Like the validation of P-V diagram in Phase 1 (4.1.3.1), log P-V demonstrate an impulsive approach plotting of an ideal gas otto cycle to understand the efficiency of the 720-degree pressure volume relation inside an engine (Kan, 2014).



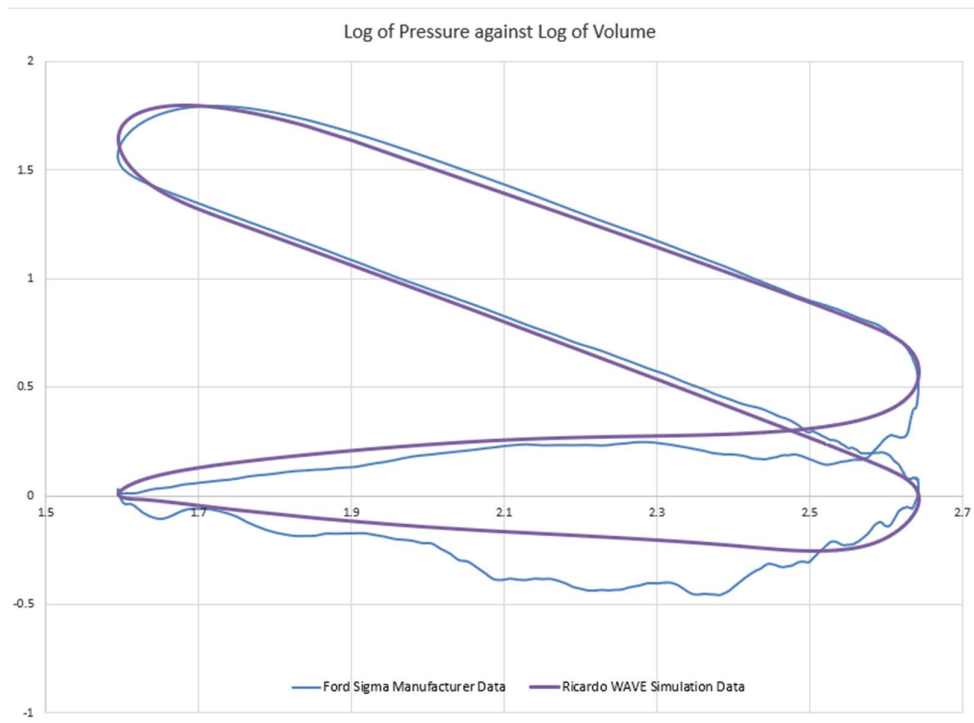
Graph 10: Log P-V diagram of the ford sigma manufacturer dyno data

The manufacturer log pressure-volume data of the ford sigma powertrain is plotted on graph 10. Y axis denotes the log of pressure and X axis denotes the log of volume which is being converted into logarithmic values from the pressure-volume data. The advantages of plotting a logarithmic P-V diagram are that it gives a finite understanding about the efficiency and work done by a system, fit of start of combustion and end of combustion between compression and expansion strokes (Kan, 2014).

The log P-V diagram requested on the cylinder WAVE model for a simulation analysis of 6000 RPM are mentioned on graph 11. The log values obtained from the computational model are exported to excel as a function of log pressure in Y-axis and log volume in X-axis. The manufacturer data and the computational data are merged as mentioned in graph 12 to understand the reliability of the WAVE combustion model.



Graph 11 : Log P-V (log of pressure vs log of volume) Diagram of the Ricardo wave combustion model



Graph 12: Log P-V (log of pressure vs log of volume) Diagram of the manufacturer data and Ricardo WAVE combustion model

As we mentioned the reliability of computational model to manufacturer data in linear pressure-volume diagram, log pressure volume diagram gives a more detailed review of all the 4 strokes in the cylinder. During suction stroke log pressure goes negative due to movement of piston from BDC to TDC as a process of intake of air fuel mixture. At the end of compression stroke, combustion in computational model begins fraction earlier to that of actual data, meanwhile pressure peaks up in computational model where the burn profile is linear for actual data. As the realistic closest fit is obtained from the computational model, further research into effects of research octane number in fuel on engine performance and emissions is carried out in the developed WAVE model.

4.2 3-Dimensional combustion modelling analysis into AFR, EGR and its influence on thermodynamic properties and emissions in Ford Lynx diesel powertrain.

4.2.1 Software

The second part of the research to study the possible effects of injection strategies, exhaust gas recirculation (EGR) and influence of AFR on thermodynamics and combustion emissions are conducted using a 3-dimensional engineering simulation software. ANSYS Forte R2 software provides a platform to research into the combustion modelling and analysis with state-of-the-art chemistry and meshing, it is possible to simulate the engine model accurately and efficiently.

For validated gas phase and surface chemistry, ANSYS Forte incorporates the proven ANSYS Chemkin-Pro solver technology where engine performance and design improvements can be predicted quickly and accurately in each iteration. The iterations in forte R2 as a chemistry solver is dependent on the number of chemically active species and reactions that are calculated from the selected chemistry set.

Automatic mesh generation and streamline workflows save users time, navigating better design solutions more quickly. The key features of ANSYS Forte R2 in current research includes automatic mesh generation, multicomponent fuel models, validated spray modelling with chemkin-pro solver technology (PADT simulation, 2022).

4.2.2 Development of Ford Lynx combustion model in ANSYS Forte R2

The combustion model of the Ford lynx specifications is created using a diesel engine sector mesh tutorial model of variable specifications. The tutorial geometry is imported into sector mesh generator for meshing. The basic engine geometry parameters are input into the sector mesh generator as mentioned on table 12 where engine bowl/piston is created as a moving boundary. As shown in figure 12, Ford Lynx cylinder model mesh generation is done in a 45-degree periodicity to study the combustion, which was later imported to Forte project as shown in figure 13.

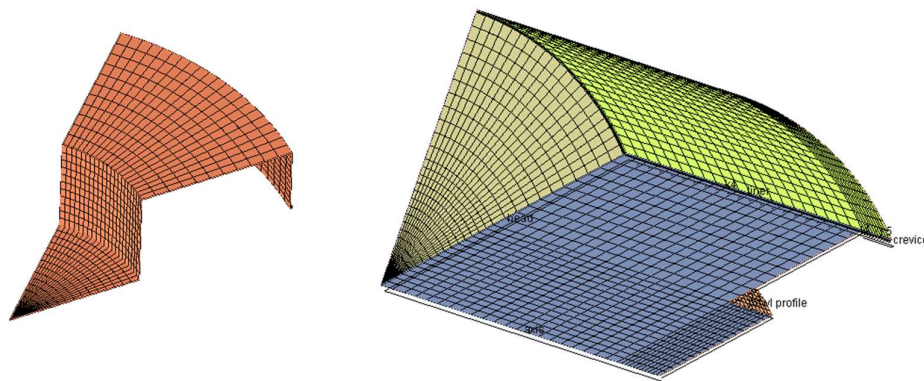


Figure 12 : Mesh generation of (a) Bowl/Piston profile (b) 45-degree periodicity profile of cylinder model on Forte sector mesh generator

Geometry Specifications	Ford Lynx	Forte R2 Validation Model	Ford Lynx
Cylinder Bore	82.5 mm	Injected fuel Mass(Stoich)($\lambda=1$)	0.039 g
Cylinder Stroke	82 mm	Start of Injection	-8 CA
Connecting Rod Length	120 mm	Injected mass Duration	25 degrees
Compression Ratio	19.4	Injection type	Pulsed
Clearance Volume	23.82 cm ³	Initial Pressure	2.49 bar
In Cylinder Volume	438.25 cm ³	Initial Temperature	323 K
Bowl depth	15.5 mm	Simulation RPM	4000
Bowl radius	18.8 mm	Mixture Properties	Mole Fraction
Calculated Squish Distance	1.24 mm	Oxygen (O2)	0.197
Crevice Height	5 mm	Nitrogen (N2)	0.719
Crevice Width	.35 mm		

Table 12 : Ford Lynx computational model parameters used for validation at 4000 RPM

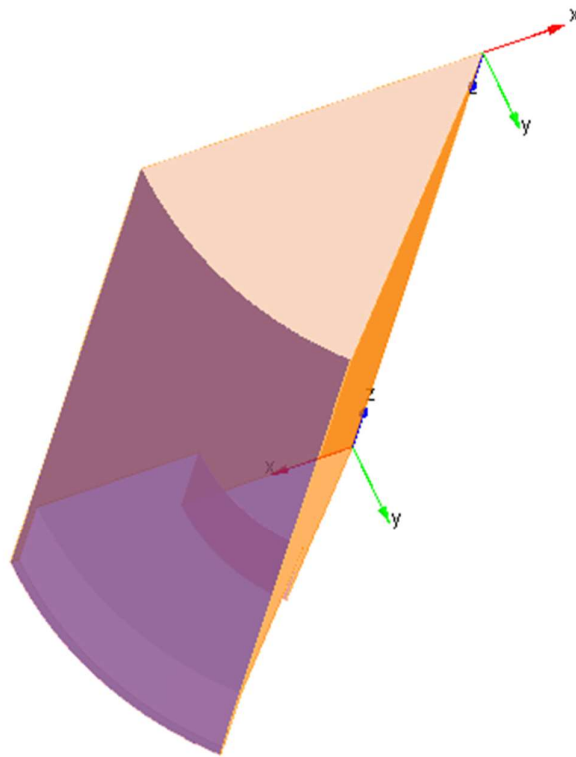
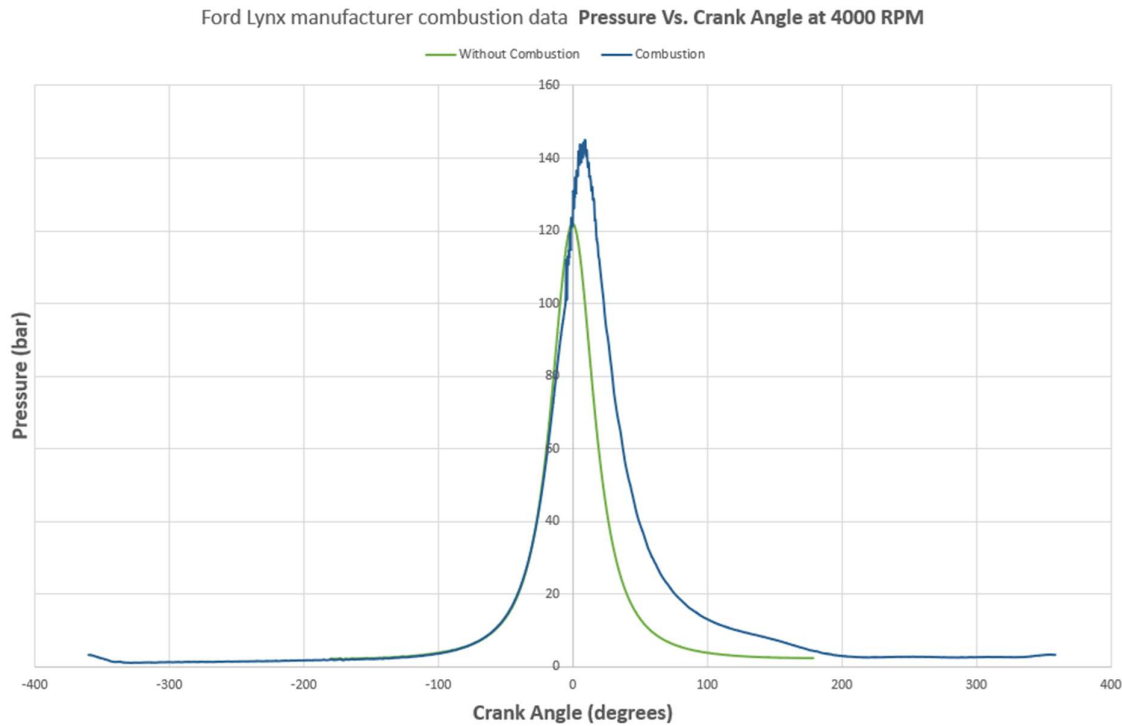


Figure 13 : Imported cylinder geometry of Ford Lynx from sector mesh generator to Forte project.

4.2.3 Validation of Ford Lynx computational model with manufacturer data

The research is done on the injected fuel mixture n-dodecane ($C_{12}H_{26}$). After a series of simulation run changing the initial pressure and temperature, injection strategies like start of injection and injection duration, squish distance and bowl parameters, the validation is done for a stoichiometric combustion of diesel fuel, when injected .039 grams at a crank angle of -8 degrees for a total duration of 25 degrees.

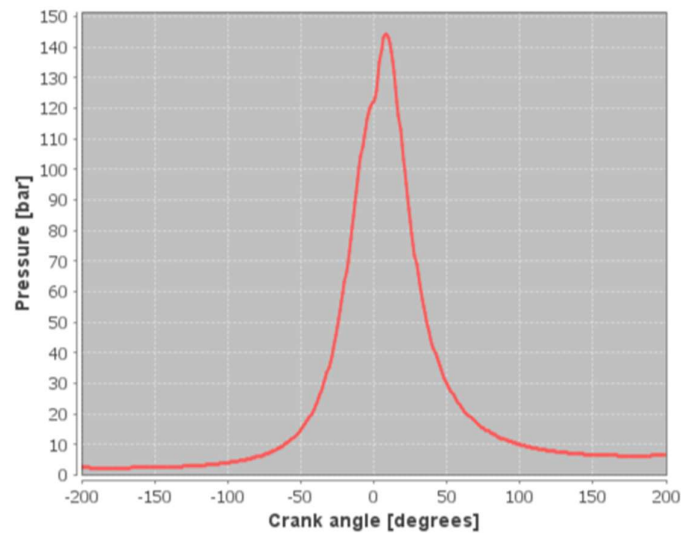


Graph 13 : Ford Lynx manufacturer data for combustion pressure at 4000 RPM and calculated in cylinder pressure without combustion

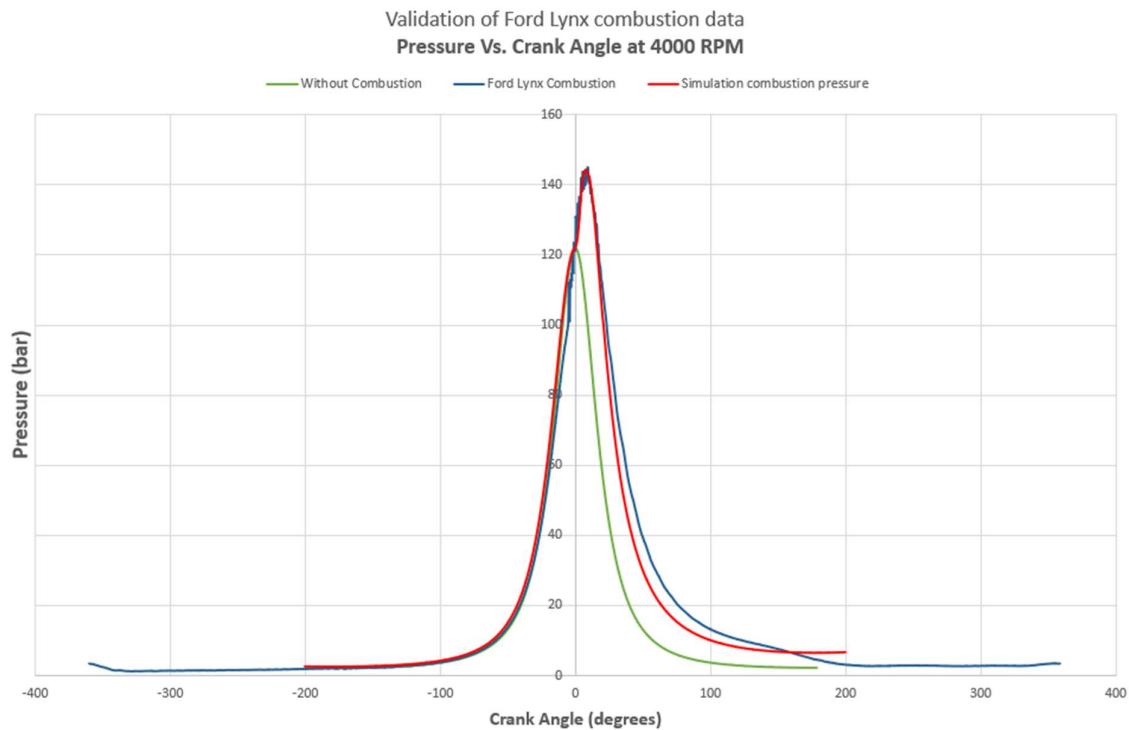
Graph 13 outlines the cylinder pressure of Ford Lynx at an engine operating speed of 4000 RPM and a non-combustion data calculated from the manufacturer data for a polytropic index of 1.3. The data concludes that, a fuel injection event influences the cylinder pressure when the piston is about to reach TDC. This drop in cylinder pressure is noted after CA -9 degrees. According to the Lynx manufacturer data, A peak cylinder pressure of 144 bar is noted after combustion of diesel at 8.4 degrees (CA) after TDC-The Combustion model pressure output is then validated with the manufacturer data for the lynx geometry specifications as shown in table 13.

Graph 14 discusses about the combustion model output of in cylinder pressure between CA -200 and +200 degrees. For the input geometry and simulation parameters as mentioned in table 13, a

complete combustion with a maximum pressure of 144 bar is obtained at a CA of 8.06 degrees after TDC-The merged data of the combustion model and manufacturer data is shown in Graph 15.



Graph 14: ANSYS Ford Lynx combustion model pressure data for stoichiometric AFR



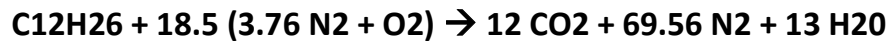
Graph 15 : Merged (Pressure vs CA) data of ANSYS Forte combustion model and Ford Lynx manufacturer pressure data.

Graph 15 concludes that the compression stroke, combustion event and expansion till 50 degrees after TDC have similar combustion performance followed by a sudden deterioration in combustion model pressure. A wide range of analysis is done in fuel mass, injection timing, injection duration,

cylinder, and piston geometry to fit the data throughout the expansion pressure as in manufacturer data. Since the combustion phenomena can be validated, further research is being carried out to understand the effects of EGR, AFR and emissions, developed in cylinder during combustion.

Forte R2 Ford Lynx model input	Ford Lynx
Research 1	Simulation Analysis of influence of AFR
Injected Fuel Mass(Stoichiometry)($\lambda=1$)	0.039 grams
Injected Fuel Mass 10% Rich ($\lambda=0.9$)	0.0429 grams
Injected Fuel Mass 10% Lean ($\lambda=1.1$)	0.351 grams
Research 2	Simulation Analysis of influence of EGR
EGR Temperature	140 degrees (413 K)
Mole Fraction CO ₂ (EGR 10 %)	0.0288
Mole fraction H ₂ O (EGR 10%)	0.01285
Mole fraction CO ₂ (EGR 20%)	0.0576
Mole fraction H ₂ O (EGR 20%)	0.0257

Table 13 : Input parameters for the study of influence of AFR and EGR on Ford Lynx engine performance and emissions.



Equation 1 : Chemical formula of combustion of n-dodecane for a stoichiometric AFR

The mole fraction of the carbon dioxide and dry H₂O for varying EGR are calculated from the chemical reaction of combustion of normal dodecane mentioned in equation 1. As 1 molecule of CO₂ is .0024 mole fraction and 1 molecule of H₂O is .00098 in mole fraction, the chemical reaction of combustion of diesel helps to calculate the total number of combustion products in terms of finding the CO₂ and H₂O by-products.

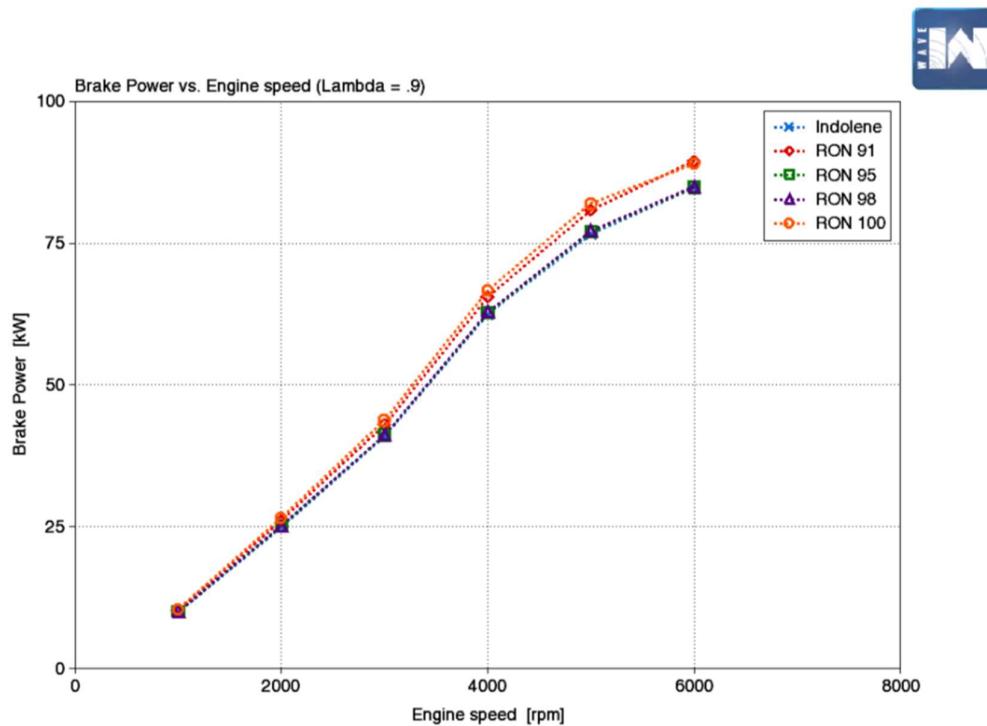
5.Simulation results and analysis

5.1 1-Dimensional combustion modelling analysis into Research Octane Number (RON) and Air Fuel Ratio of fuel mixtures influence on engine performance and emissions in a gasoline powertrain.

There was a need for research to determine the effects of varying gasoline fuel properties on powertrain performance and emissions worldwide. The combustion modelling analysis has been conducted in fuels with RON 91,95,98,100 and indolene, an 87-octane test fuel. Chapter 5.1 discuss about effects of RON in Ford Sigma engine performance in rich and lean air fuel mixture operation.

5.1.1 Simulation analysis of influence of Research Octane Number (RON) on Ford Sigma engine performance

5.1.1.1 Brake Power and Indicated Power



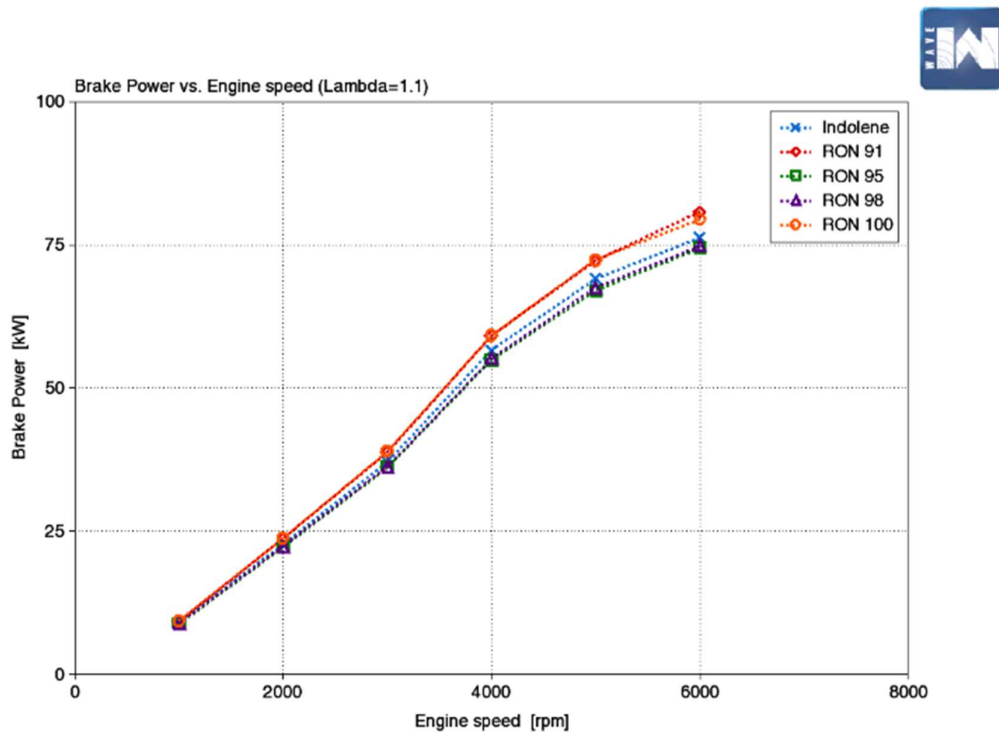
Graph 16: Impact of RON on engine Brake Power for rich air fuel mixture ($\lambda = 0.9$)

Graph 16 outlines the power output available at the crankshaft of the Ford Sigma combustion model for a rich air fuel mixture. RON 100 fuel is recorded to have highest fuel performance up to 5000 RPM, this is significantly closer to the performance output of RON 91 fuel which have a maximum brake power of 89.43 kW at 6000 RPM over RON 100. It has been demonstrated that RON 100's power output declines between 5000 and 6000 RPM while RON 91's power output is linearly increasing at all engine operating conditions.

Brake Power vs Engine speed ($\lambda=0.9$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	kW	kW	kW	kW	kW
1000	9.90	10.27	10.01	10.02	10.50
2000	24.97	26.15	25.15	25.20	26.60
3000	40.98	43.01	41.18	41.24	43.83
4000	62.45	65.60	62.75	62.87	66.71
5000	76.74	80.85	76.98	77.16	81.90
6000	84.78	89.44	84.81	84.94	88.99

Table 14 : Brake Power combustion model performance data for different fuels ($\lambda=0.9$)

Research fuels RON 95 and RON 98 are low-performance fuels from both data. RON 98, on the other hand, displays microscale times greater performance than both RON 95 and indolene. Based on the results from the rich fuel mixture simulation, indolene, RON 95, and RON 98 are all average performing fuels. However, RON 91 offers a performance advantage of 4.62 kW at 6000 RPM over RON 95.



Graph 17: Impact of RON on engine Brake Power for lean air fuel mixture ($\lambda=1.1$)

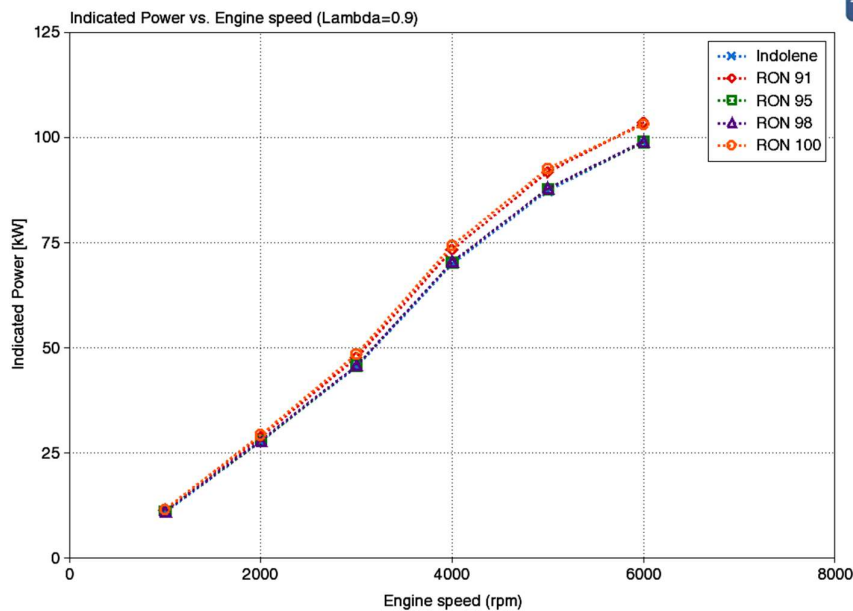
Graph 17 outlines the simulated power output available at the dynamometer for the lean air-fuel mixture. It has been noted that there has been a downswing in brake power on all fuels when the mixture is lean. RON 91 has a performance loss of 8.73 kW at 6000 RPM which concludes that a rich mixture has a significant impact on the combustion performance of gasoline blends. Indolene offers better performance compared to RON 98 and RON 95 when the mixture is lean.

Brake Power vs Engine speed ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	kW	kW	kW	kW	kW
1000	9.07	9.36	8.86	8.89	9.42
2000	22.81	23.73	22.29	22.33	23.82
3000	37.21	38.90	36.21	36.35	38.97
4000	56.57	59.11	54.93	55.20	59.22
5000	69.07	72.26	67.00	67.48	72.34
6000	76.93	80.71	74.45	74.78	79.47

Table 15 : Brake Power combustion model performance data for different fuels ($\lambda=1.1$)

The indicated power of different fuels is studied from graphs 15 and 16 as a function of engine speed for varying fuel mixtures. Indicated power is the actual power generated in the cylinders due to the combustion of fuel which outlines the key difference in fuel performance. Considering that indicated power is a function of brake power and friction power (also referred to as losses in an engine), it has the same characteristics as the data shown in graphs 13 and 14.

Characteristics of RON in graph 18 conclude that for lambda 0.9, RON 100 and RON 91 are the high-performance fuels followed by RON 98, indolene, and RON 95, where RON 91 has a power output of 103.56 kW at 6000RPM, 4.75kW more than RON 95. By comparing the data, RON 98, indolene, and RON 95 are low-performance fuels and have similar power outputs in all engine operating conditions.

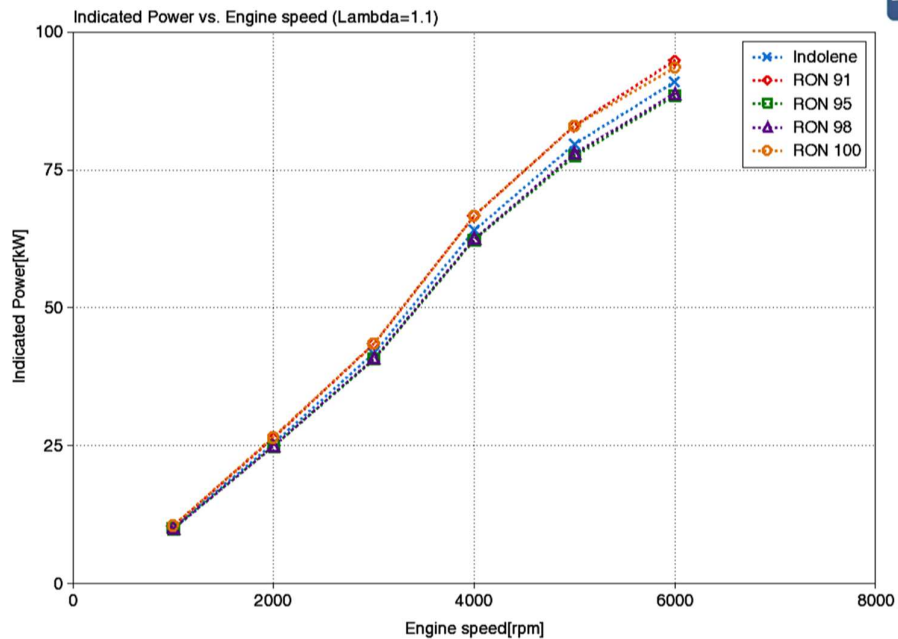


Graph 18: Impact of RON on engine Indicated Power for rich air fuel mixture ($\lambda = 0.9$)

Indicated Power vs Engine speed ($\lambda=0.9$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	kW	kW	kW	kW	kW
1000	11.08	11.47	11.19	11.20	11.68
2000	27.69	28.92	27.86	27.91	29.34
3000	45.62	47.76	45.82	45.89	48.52
4000	70.02	73.33	70.31	70.45	74.35
5000	87.32	91.74	87.68	87.86	92.66
6000	98.82	103.57	98.88	99.01	103.11

Table 16 : Indicated Power combustion model performance data for different fuels ($\lambda=0.9$)

Graph 19 depicts the indicated power for lean air fuel mixture for a λ of 1.1. It has been noted that there has been a significant decrease in indicated power on all fuels when the mixture is lean. RON 91 has a performance loss of 8.73 kW at 6000 RPM which concludes that rich mixture has a significant impact in the combustion performance of gasoline blends. Indolene offers better performance compared to RON 98 and RON 95 when the mixture is lean.



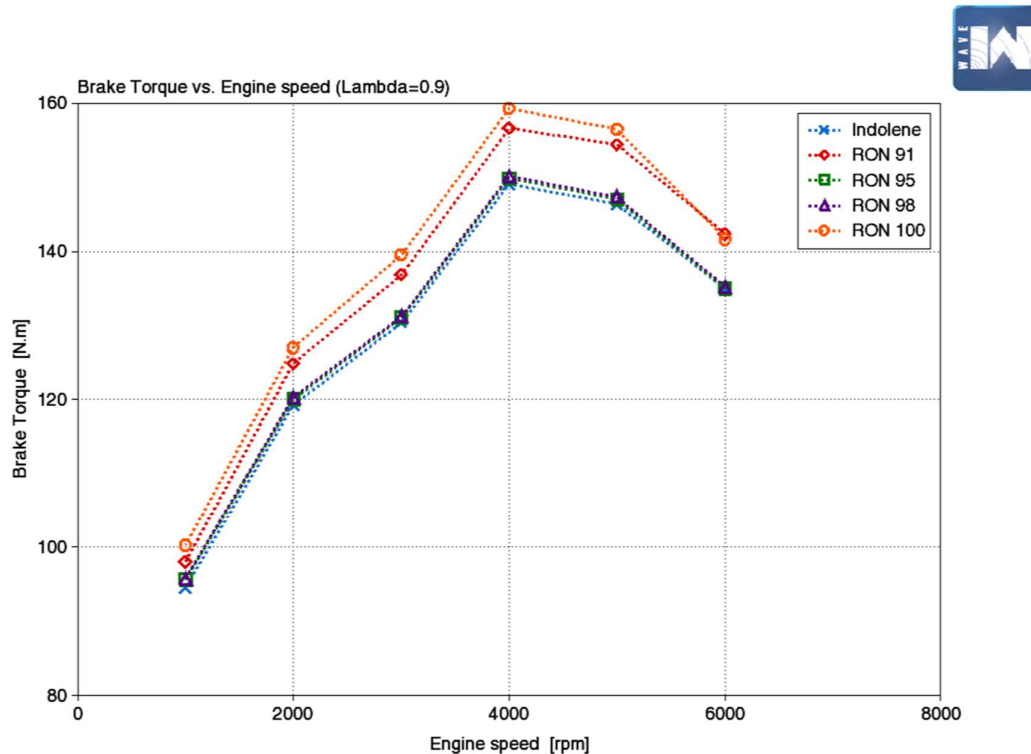
Graph 19 : Impact of RON on engine Indicated Power for lean air fuel mixture ($\lambda=1.1$)

Indicated Power vs Engine speed ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	kW	kW	kW	kW	kW
1000	10.22	10.53	9.99	10.02	10.57
2000	25.47	26.43	24.93	24.98	26.50
3000	41.78	43.55	40.76	40.91	43.58
4000	64.03	66.69	62.30	62.59	66.70
5000	79.63	82.98	77.48	77.97	82.93
6000	90.98	94.83	88.46	88.81	93.57

Table 17 : Indicated Power combustion model performance data for different fuels ($\lambda=1.1$)

5.1.1.2 Brake Torque and Indicated Torque

Torque is a function used to indicate the performance of an engine, produced on the crankshaft due to combustion of fuel that push piston from TDC to BDC during a power stroke. It is directly proportional to the power and inversely proportional to the engine speed. Brake torque is the output torque available after frictional and pumping losses whereas indicated torque is the torque available at crankshaft (Ribbens, 2013).



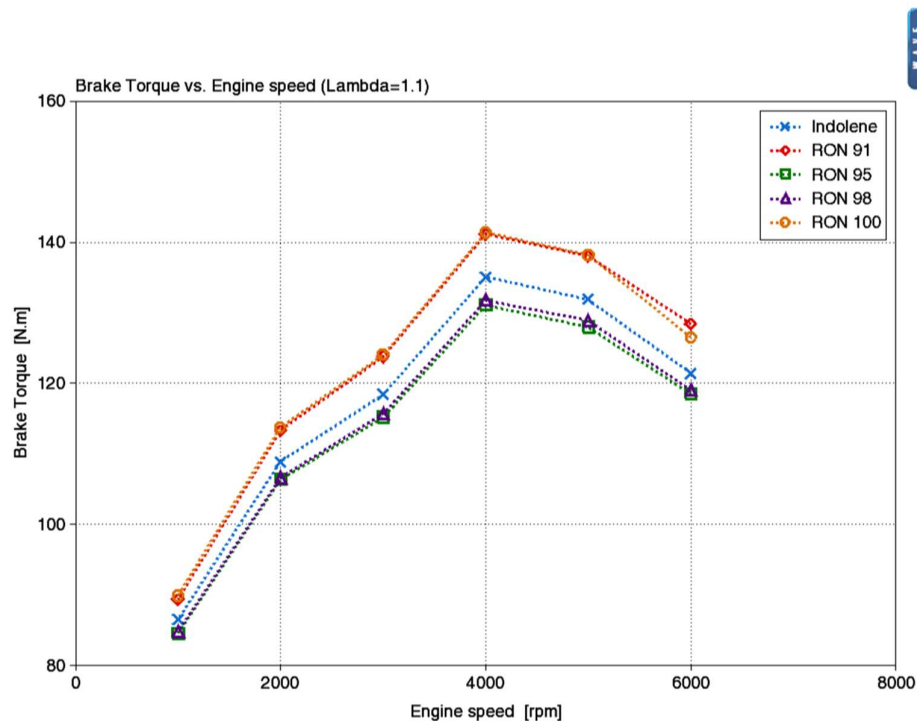
Graph 20: impact of RON on engine Brake Torque for rich air fuel mixture ($\lambda=0.9$)

Graph 20 and graph 21 depicts the engine brake torque characteristics for rich and lean air fuel mixtures a Ford Sigma geometry. Torque increases gradually up to 4000 RPM where maximum

torque is obtained. RON 100 is the high-performance fuel with 159.26 Nm at 4000 RPM which have a 1.66% increase in brake torque over RON 91 with 156.61 Nm. RON 98 produces a torque of 150.09 Nm and have slight increase in brake torque at 4000 RPM than RON 95 and indolene with 149.81 Nm and 149.07 Nm.

Brake Torque vs Engine speed ($\lambda=0.9$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	Nm	Nm	Nm	Nm	Nm
1000	94.57	98.07	95.62	95.72	100.28
2000	119.24	124.87	120.07	120.31	127.00
3000	130.44	136.91	131.07	131.26	139.50
4000	149.07	156.61	149.81	150.09	159.26
5000	146.34	154.42	147.03	147.37	156.42
6000	134.91	142.35	134.99	135.19	141.64

Table 18 : Brake Torque combustion model performance data for different fuels ($\lambda=0.9$)



Graph 21: Impact of RON on engine Brake Torque for lean air fuel mixture ($\lambda=1.1$)

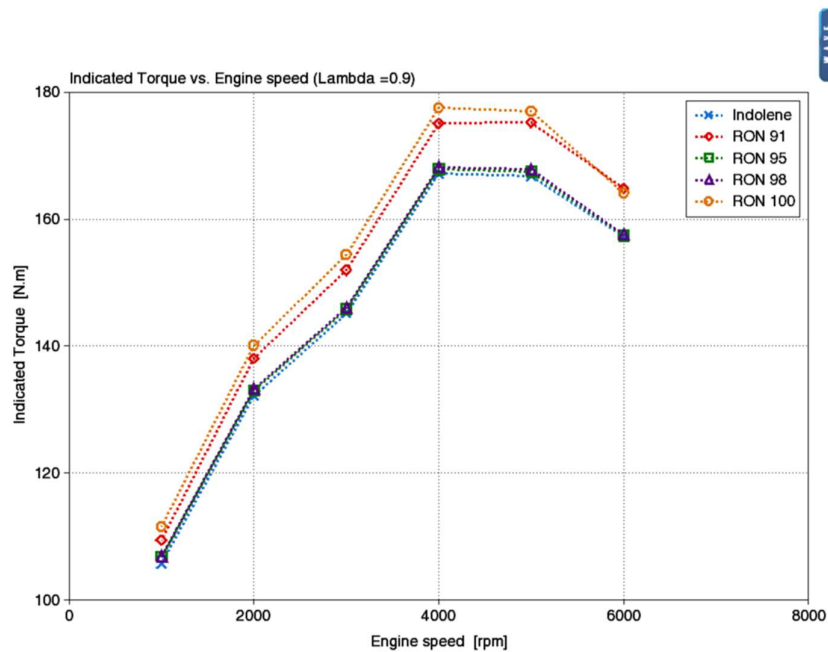
Graph 21 concludes that, for a lean air fuel mixture, RON 100 and RON 91 have literally the same performance at 4000 RPM where there was a contrasting difference during combustion of rich mixture. Indolene is the third best performance fuel over RON 98 and RON 95 when the mixture is lean. It is noted that RON 100 has a torque loss of 17.89 Nm when lambda changes from 0.9 to 1.1, which is an 11.23% reduction in brake torque.

Graph 22 and 23 describes about the influence of RON on the Indicated torque of the Ford Sigma WAVE combustion model. As previously mentioned about the behaviour of RON on the rich and lean mixture of the different air fuel mixtures, RON 100 is the high-performance fuel with respect to

indicated torque with a peak value of 177.49 Nm over 175.06 Nm of RON 91. Indicated torque decreases slowly and literally remains the same between 4000 and 5000 RPM. In terms of lean air fuel mixture when lambda is 1.1, RON 100 delivers a maximum indicated torque of 159.24 Nm, 18.25 Nm less than the indicated torque when lambda is .9.

Brake Torque vs Engine speed ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	Nm	Nm	Nm	Nm	Nm
1000	86.58	89.41	84.57	84.85	89.94
2000	108.90	113.29	106.41	106.63	113.74
3000	118.44	123.83	115.25	115.72	124.06
4000	135.05	141.12	131.12	131.77	141.37
5000	131.91	138.01	127.97	128.88	138.16
6000	121.41	128.46	118.49	119.02	126.49

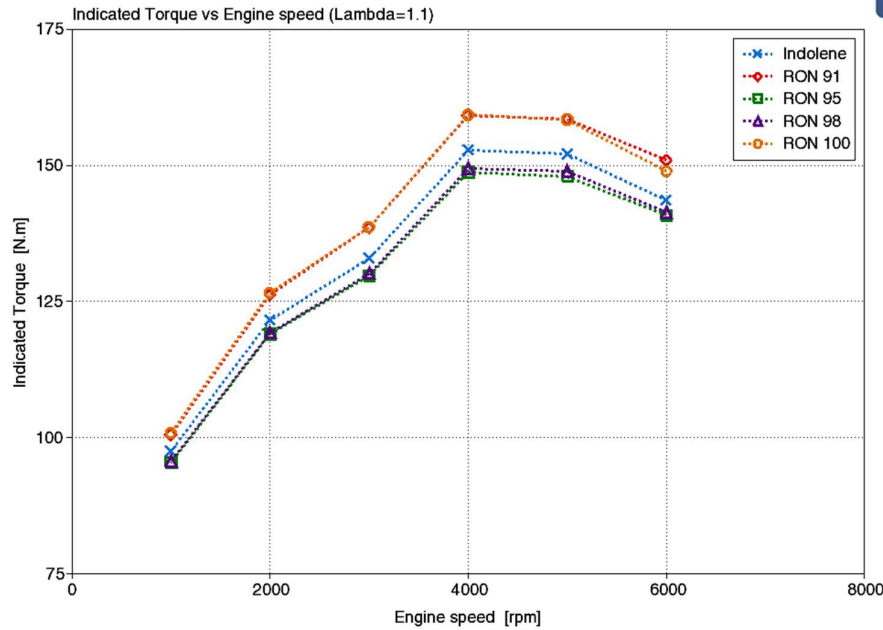
Table 19 : Brake Torque combustion model performance data for different fuels ($\lambda=1.1$)



Graph 22 : Impact of RON on engine Indicated Torque for rich air fuel mixture ($\lambda=0.9$)

Indicated Torque vs Engine speed ($\lambda=0.9$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	Nm	Nm	Nm	Nm	Nm
1000	105.80	109.50	106.82	106.93	111.57
2000	132.19	138.09	133.02	133.27	140.11
3000	145.23	152.02	145.84	146.07	154.45
4000	167.16	175.07	167.86	168.18	177.50
5000	166.77	175.21	167.46	167.80	176.98
6000	157.27	164.84	157.37	157.58	164.11

Table 20 : Indicated Torque data from combustion model performance analysis of different fuels ($\lambda=0.9$)



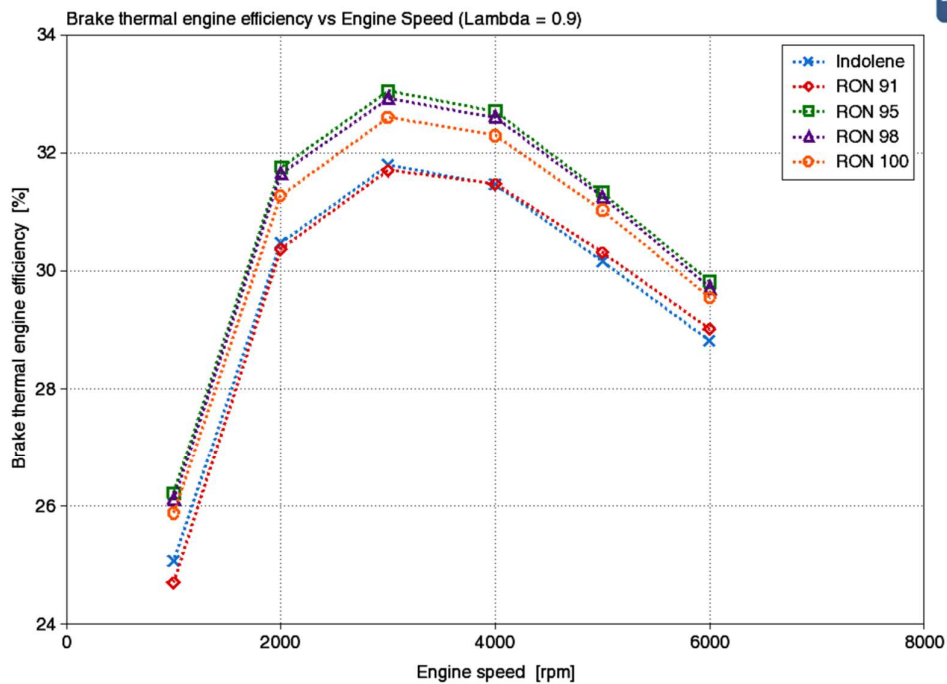
Graph 23 :Impact of RON on engine Indicated Torque for lean air fuel mixture ($\lambda = 1.1$)

Due to the presence of frictional and pumping losses in brake torque, the graph has a significant decrease in brake torque from 4000 to 5000 RPM. Indolene is observed to have higher indicated torque than RON 98 and RON 95 during combustion of lean mixture. This characteristics of indolene concludes that due to low RON in indolene the performance characteristics are better during combustion of lean mixture. The higher-octane number in RON 100 also shows a behaviour in peak performance when the mixture is lean.

Indicated Torque vs Engine speed ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	Nm	Nm	Nm	Nm	Nm
1000	144.80	150.93	140.79	141.34	148.92
2000	152.08	158.48	147.97	148.90	158.38
3000	152.85	159.22	148.74	149.42	159.24
4000	132.99	138.63	129.73	130.21	138.71
5000	121.63	126.22	119.03	119.26	126.55
6000	97.55	100.56	95.42	95.71	100.94

Table 21 : Indicated Torque data from combustion model performance analysis of different fuels ($\lambda=1.1$)

5.1.1.3 Brake Thermal Engine Efficiency

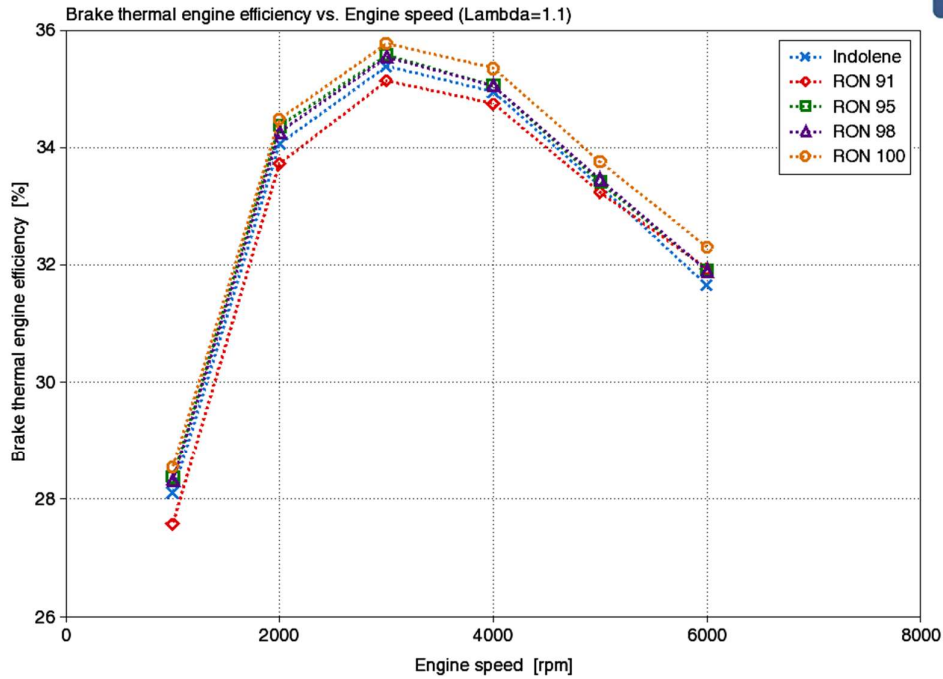


Graph 24 : Impact of RON on Brake thermal engine efficiency for rich air fuel mixture ($\lambda=0.9$)

Graph 24 and 25 illustrate the brake thermal engine efficiency for lambda 0.9 and 1.1, which is a measure of fuel consumption that can be used to evaluate the efficiency of the system and its influence on greenhouse emissions. As fuel energy released in the cylinder is an inverse function of Brake thermal engine efficiency, it is important to compare the influence of RON during combustion to understand the possible effects of RON.

Brake Thermal Engine Efficiency ($\lambda=0.9$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	%	%	%	%	%
1000	25.08	24.71	24.71	26.11	25.88
2000	30.47	30.37	30.37	31.65	31.27
3000	31.79	31.71	31.71	32.92	32.60
4000	31.46	31.47	31.47	32.60	32.29
5000	30.17	30.31	30.31	31.25	31.03
6000	28.81	29.01	29.01	29.71	29.55

Table 22 : Brake Thermal Engine Efficiency data from combustion model performance analysis of different fuels ($\lambda=0.9$)



Graph 25: Impact of RON on Brake thermal engine efficiency for lean fuel mixture ($\lambda=1.1$)

During the combustion of rich mixture from graph 24, The maximum brake thermal efficiency in ford sigma powertrain is noted at 3000 RPM, recorded highest for RON 95 with 33.04% at 3000 RPM and from idling conditions to higher RPM. RON 95 and RON 98 have consistent BTEE throughout the engine speed where a peak efficiency is noted for RON 95 at 2000 RPM with .1 % compared to RON 98. (Vedat Karaman, 2016) states that BTEE is directly proportional to brake power and inversely proportional to fuel energy which is a function of fuel flow and calorific value.

Brake Thermal Engine Efficiency ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	%	%	%	%	%
1000	28.11	27.58	28.36	28.31	28.54
2000	34.06	33.73	34.36	34.26	34.48
3000	35.38	35.14	35.58	35.54	35.77
4000	34.95	34.75	35.06	35.06	35.35
5000	33.33	33.24	33.41	33.46	33.76
6000	31.91	31.93	31.90	31.90	32.30

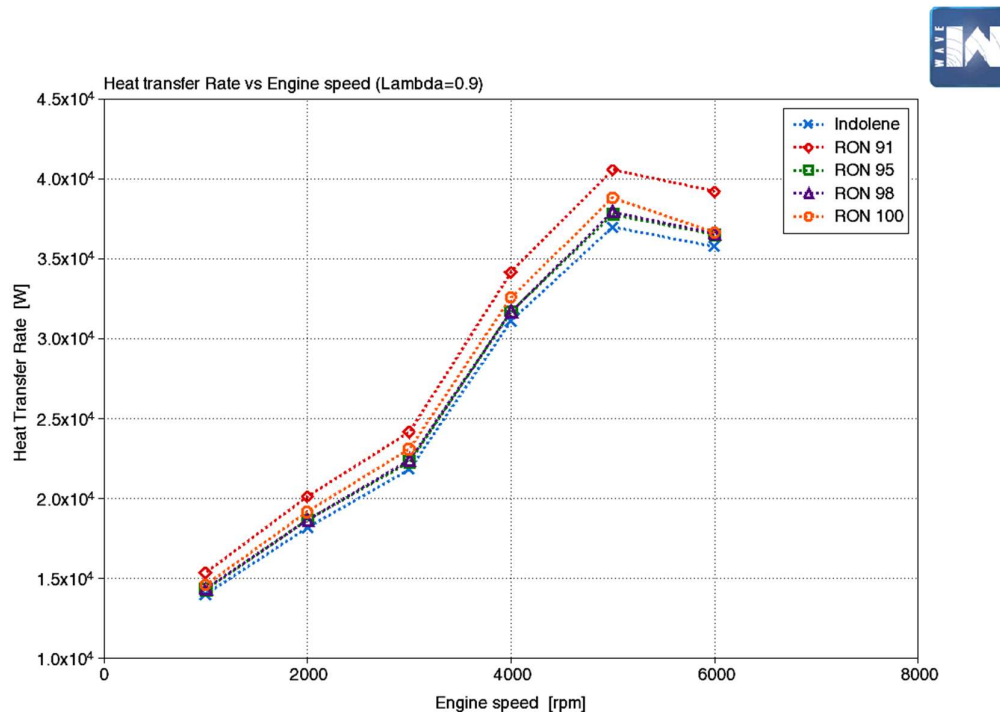
Table 23 : Brake Thermal Engine Efficiency data from combustion model performance analysis of different fuels ($\lambda=0.9$)

This concludes that, apart from being a high-performance fuel, RON 91 has the least BTTE than all the fuel used for study. At 3000 RPM RON 100 have .62 % more BTEE than RON 91. This is due to the highest calorific value or heating value of RON 91 as an ideal gas compared to other fuels which are discussed in chapter 5.1.2. As RON 95 is highly efficient, it is 4.05% more efficient than RON 91 at 3000 RPM.

Graph 25 depicts the brake thermal engine efficiency of the combustion of lean mixture. A different behaviour is noted in the data, which concludes that RON 100 has highest brake thermal engine efficiency over RON 95 and 98 with 35.76% at 4000 RPM. RON 91 has the least efficiency compared to all fuels taken for analysis when the mixture is lean and rich, 1.75% less efficient than RON 100 at 3000 RPM.

5.1.1.4 Heat Transfer

The heat transfer characteristics of different fuels during combustion of rich and lean air fuel mixtures are shown in graph 26 and 27. Heat is often transferred from combustion zone to cylinder head, liner, and piston.



Graph 26: Impact of RON on Heat Transfer Rate for rich air fuel mixture ($\lambda = 0.9$)

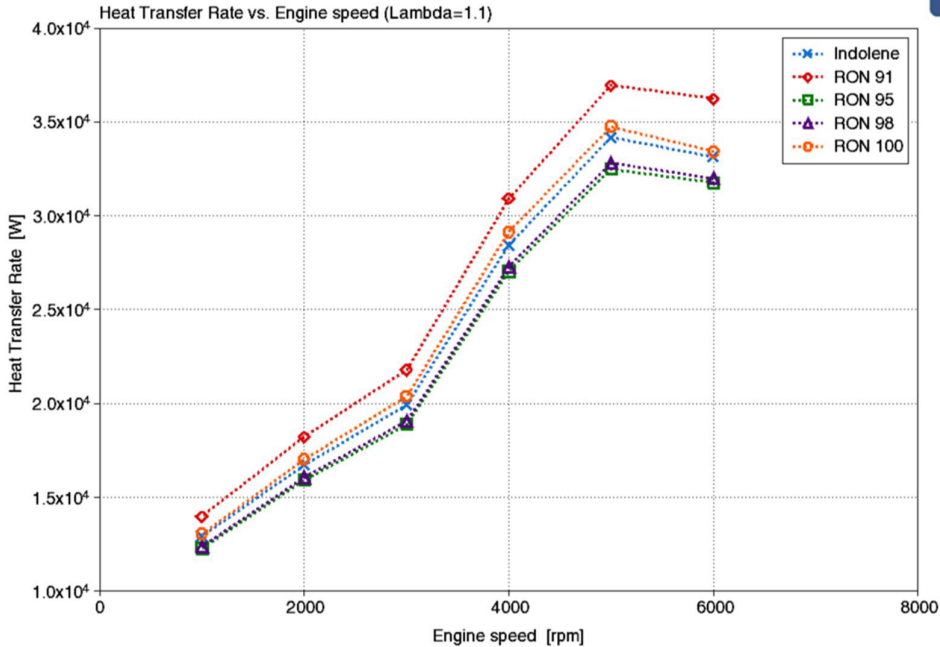
Graph 26 concludes that for lambda 0.9, heat transfer increases from 0 to 5000 RPM and decreases gradually above 5000 RPM in a Ford Sigma powertrain. RON 91 as a performance fuel have highest heat transfer rate at 5000 RPM with 40.53 kW. RON 100 as a high-performance fuel over RON 91 in

operating conditions between 1000 and 5000 RPM have less heat transfer rate than RON 91 is a contrasting behaviour. This can result in higher emissions in RON 91 which will be discussed in later chapters. At 5000 RPM, RON 91 has a heat transfer of over 1.72 kW than RON 100 with a maximum heat transfer rate of 38.81 kW. RON 95 and RON 98 have similar heat transfer ate throughout the engine speed with indolene with least heat transfer.

Heat Transfer Rate vs Engine speed ($\lambda=0.9$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	kJ/s	kJ/s	kJ/s	kJ/s	kJ/s
1000	14.02	15.36	14.31	14.36	14.60
2000	18.19	20.15	18.65	18.66	19.17
3000	21.85	24.18	22.30	22.42	23.11
4000	31.11	34.15	31.68	31.71	32.59
5000	36.98	40.54	37.77	37.94	38.81
6000	35.77	39.20	36.50	36.56	36.63

Table 24 : Heat Transfer Rate data from combustion model performance analysis of different fuels ($\lambda=1.1$)

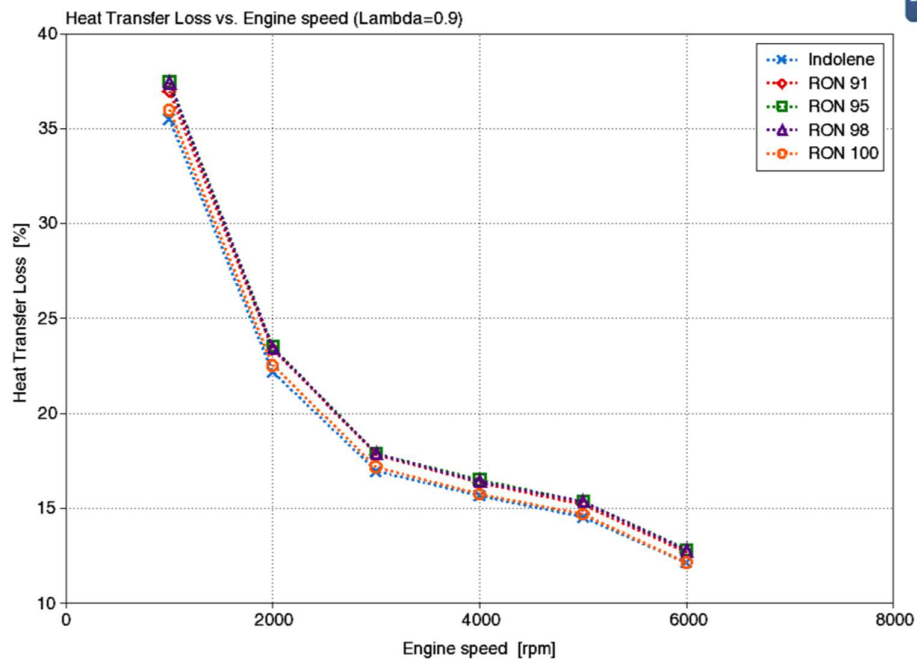
Graph 27 concludes the combustion of lean air fuel mixture. As in previous discussions lean mixtures have a negative impact on delivering fuel energy into power output, heat transfer rate decreases gradually at all operating conditions compared to a combustion of a rich fuel mixture. RON 91 has a highest heat transfer rate of 3.69 kW over RON 100 followed by indolene. As discussed in previous performance analysis, Indolene as in lean air fuel mixture combustion have better fuel characteristics than RON 95 and 98, the heat transfer rate concludes that indolene have more heat transfer than RON 95 and RON 98. Comparing the heat transfer loss of RON 91 at 5000 RPM for a lambda of 0.9 and 1.1, lean mixture has a heat transfer loss of 3.59 kW in a single cycle.



Graph 27 : Impact of RON on Heat Transfer Rate for lean air fuel mixture ($\lambda=1.1$)

Heat Transfer Rate vs Engine speed ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	kJ/s	kJ/s	kJ/s	kJ/s	kJ/s
1000	12.94	13.98	12.28	12.38	13.07
2000	16.74	18.21	15.92	16.06	17.04
3000	19.93	21.80	18.90	19.08	20.40
4000	28.43	30.93	27.04	27.28	29.13
5000	34.16	36.94	32.49	32.81	34.76
6000	33.47	36.24	31.77	31.99	33.42

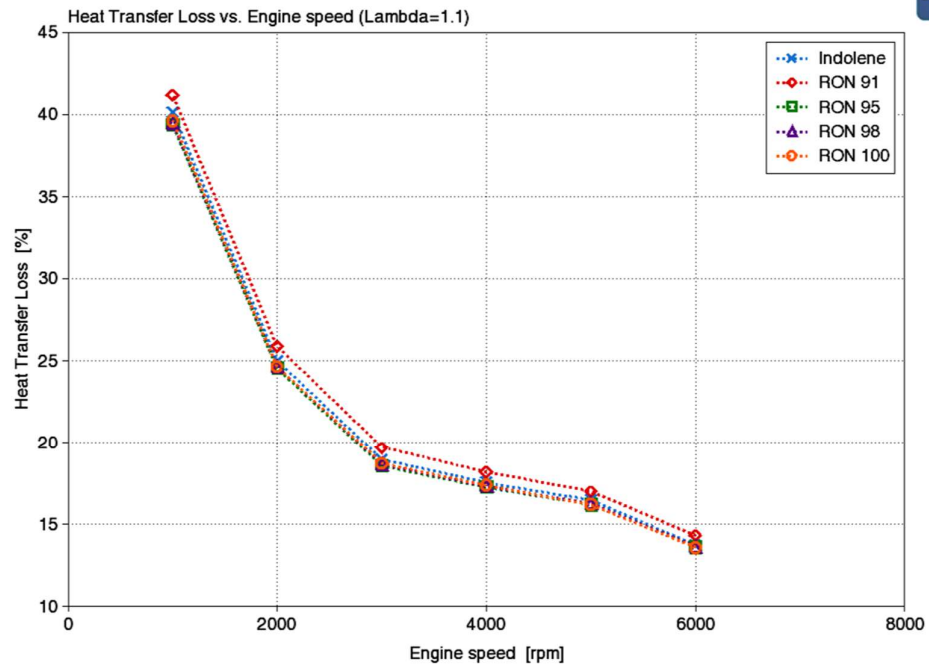
Table 25 : Heat Transfer Rate data from combustion model performance analysis of different fuels ($\lambda=1.1$)



Graph 28: Impact of RON on Heat Transfer Loss for rich air fuel mixture ($\lambda=0.9$)

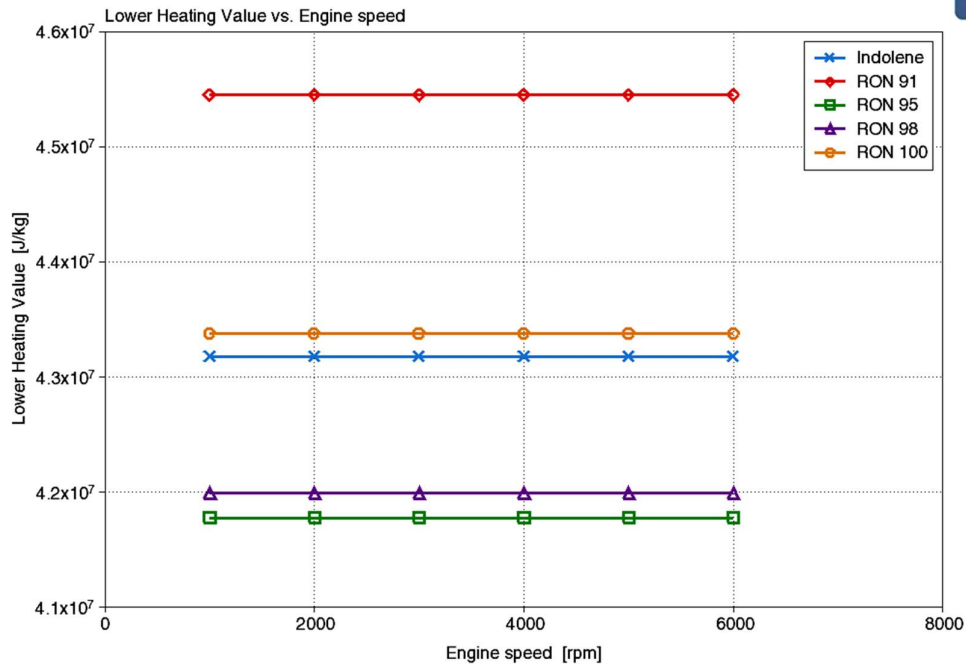
Graph 28 and 29 outlines the heat transfer loss at the varying engine speed for different fuel mixtures. Heat transfer loss is recorded highest in idling conditions and decreases when engine speed increases. Graph 25 concludes that during the combustion of rich air fuel mixture, RON 95 and RON 98 fuel have highest heat transfer loss over RON 91, RON 100 and Indolene.

Graph 28 concludes that during the combustion of lean air fuel mixture, RON 91 is recorded with highest heat transfer loss of 41.18% at 1000 RPM which is significantly higher. RON 100,95,98 and indolene fuel have close characteristics in terms of heat transfer loss.



Graph 29: Impact of RON on Heat Transfer Loss for lean air fuel mixture ($\lambda = 1.1$)

5.1.2 Simulation analysis of influence of Research Octane Number (RON) on Fuelling, Breathing and Thermodynamic properties of Ford sigma engine



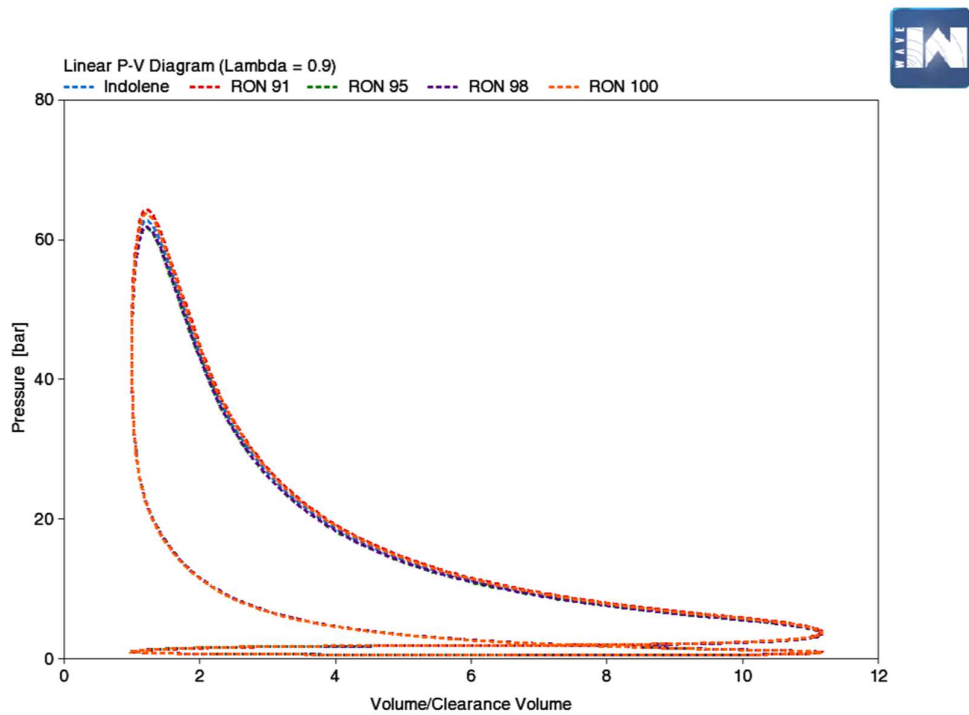
Graph 30 : Lower Heating Value of different Fuels

Graph 30 shows the lower heating value of the fuels used for simulation. RON 91 as a high-performance fuel have the highest LHV compared to other fuels with 45.45 MJ/kg followed by RON 100 with 43.38 MJ/kg, Indolene with 43.18 MJ/kg, RON 98 with 41.99 and RON 95 with 41.78 MJ/kg least LHV with 41.78 MJ/kg. RON 91 release 3.67 more MJ/kg of fuel burned energy compared to that of RON 95 as least performance fuel, which should reflect on engine performance, emissions, and thermodynamics.

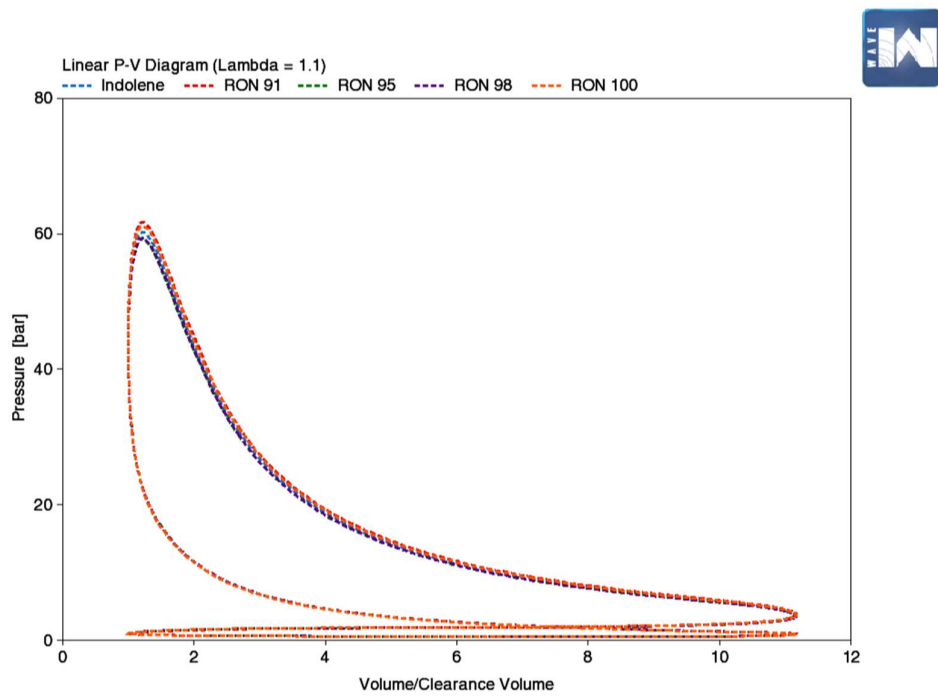
Fuel	Indolene	RON 91	RON 95	RON 98	RON 100
Lower Heating Value (MJ/kg)	43.18	45.45	41.78	41.99	43.38
Maximum Pressure (bar) ($\lambda=0.9$)	62.72	64.30	61.68	61.82	63.68
Maximum Pressure (bar) ($\lambda=1.1$)	60.22	61.68	59.20	59.33	61.11

Table 26 : Lower Heating Value, maximum combustion pressure data from Ford sigma WAVE combustion model analysis

Previously in chapter 4.1, the validation of Ford Sigma combustion model and manufacturer data is done on the linear pressure-volume(P-V) diagram where the test fuel was assumed as indolene, which is a test fuel used in dynamometers. Graph 31 and 32 gives an analysis into the combustion characteristics of other fuels.



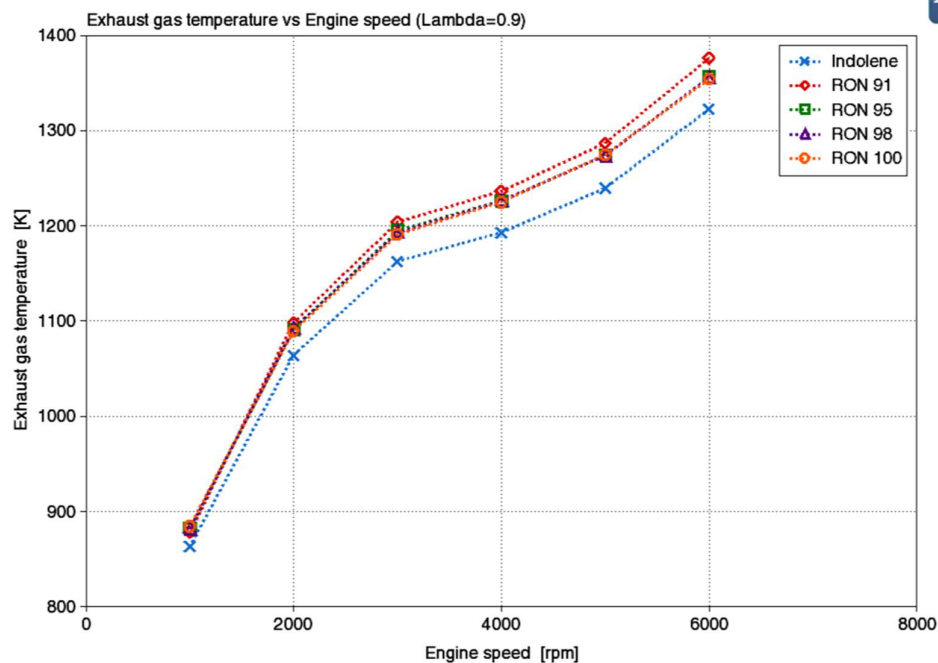
Graph 31 : Impact of RON on Linear P-V diagram for rich air fuel mixture ($\lambda = 0.9$)



Graph 32: Impact of RON on Linear P-V diagram for lean air fuel mixture ($\lambda = 1.1$)

Graph 31 discusses the influence of RON on the combustion performance of a rich air fuel mixture. The study is based on $\lambda = 0.9$. RON 91 has a peak pressure of 64.29 bar over RON 100 with 63.67 bar. Indolene has the third highest combustion pressure of 62.72 bar whereas RON 98 has 61.81 bar and RON 95 has 61.68 bar. This peak pressure analysis is obtained at a clearance volume of 48.14 cubic capacity when the piston retracts from TDC to BDC due to the combustion process.

Graph 32 concludes that the lean mixture has a deteriorating influence on the cylinder combustion pressure. RON 91 has a peak pressure of 61.68 bar over RON 100 with 61.11 bar. Indolene has the third highest combustion pressure at 60.21 bar, while RON 98 has 59.32 bar and RON 95 has 59.20 bar.

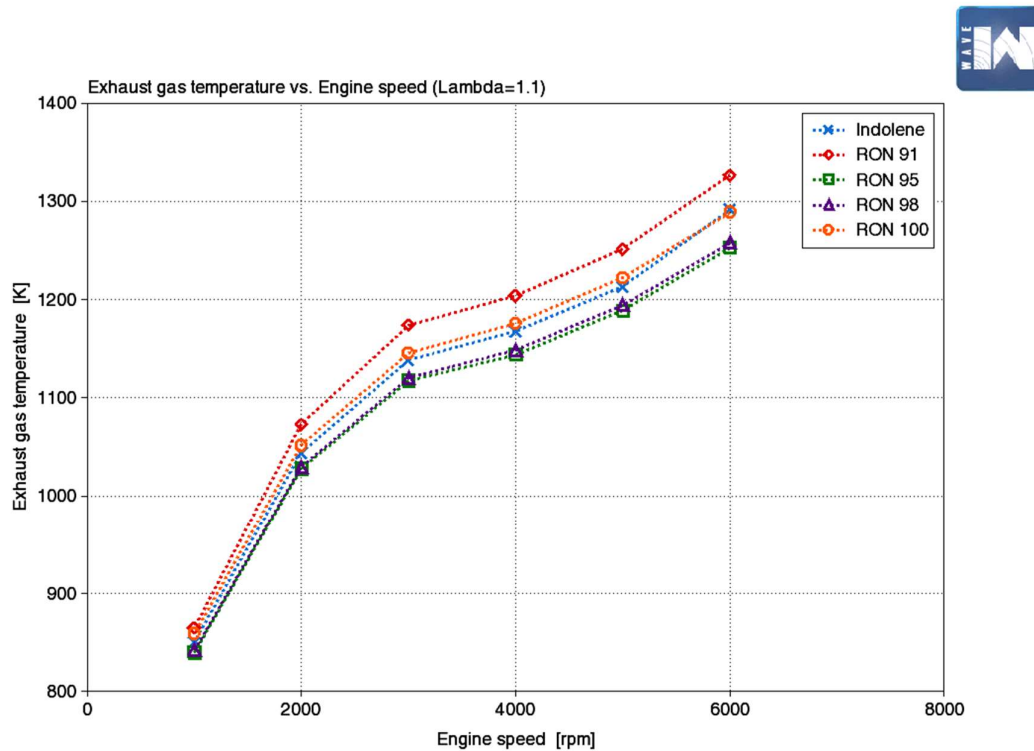


Graph 33: Impact of RON on exhaust gas temperature for rich air fuel mixture ($\lambda = 0.9$)

Graph 33 discusses the influence of RON in gasoline blends on the exhaust gas temperature at varying operating conditions. Graph outlines that regardless of fuel, exhaust gas temperature is directly proportional to engine speed and is greatest for RON 91. The exhaust gas temperatures of RON 98, RON 95, and RON 100 are similar, with indolene having the lowest temperature. An analysis of the air-fuel ratio is done by comparing the study of the rich air-fuel mixture with the lean mixture, addressed in graph 34.

Exhaust gas temperature vs Engine speed ($\lambda=0.9$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	K	K	K	K	K
1000	863.57	878.42	882.17	881.67	884.59
2000	1063.79	1098.37	1091.28	1092.95	1089.69
3000	1162.66	1204.13	1194.98	1193.28	1191.06
4000	1192.93	1236.79	1225.89	1226.62	1225.07
5000	1239.57	1287.22	1273.58	1273.74	1274.72
6000	1322.57	1376.30	1356.11	1356.07	1354.55

Table 27 : Exhaust gas temperature from WAVE combustion model performance analysis of different fuels ($\lambda=0.9$)



Graph 34: Impact of RON on exhaust gas temperature for lean air fuel mixture ($\lambda = 1.1$)

Exhaust gas temperature vs Engine speed ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	K	K	K	K	K
1000	850.45	865.24	839.70	842.14	859.20
2000	1043.87	1072.80	1028.13	1029.55	1051.73
3000	1138.51	1174.34	1117.63	1120.40	1145.66
4000	1167.80	1204.15	1144.16	1148.39	1176.06
5000	1213.12	1251.45	1188.80	1194.39	1222.10
6000	1281.19	1326.79	1252.96	1257.72	1289.23

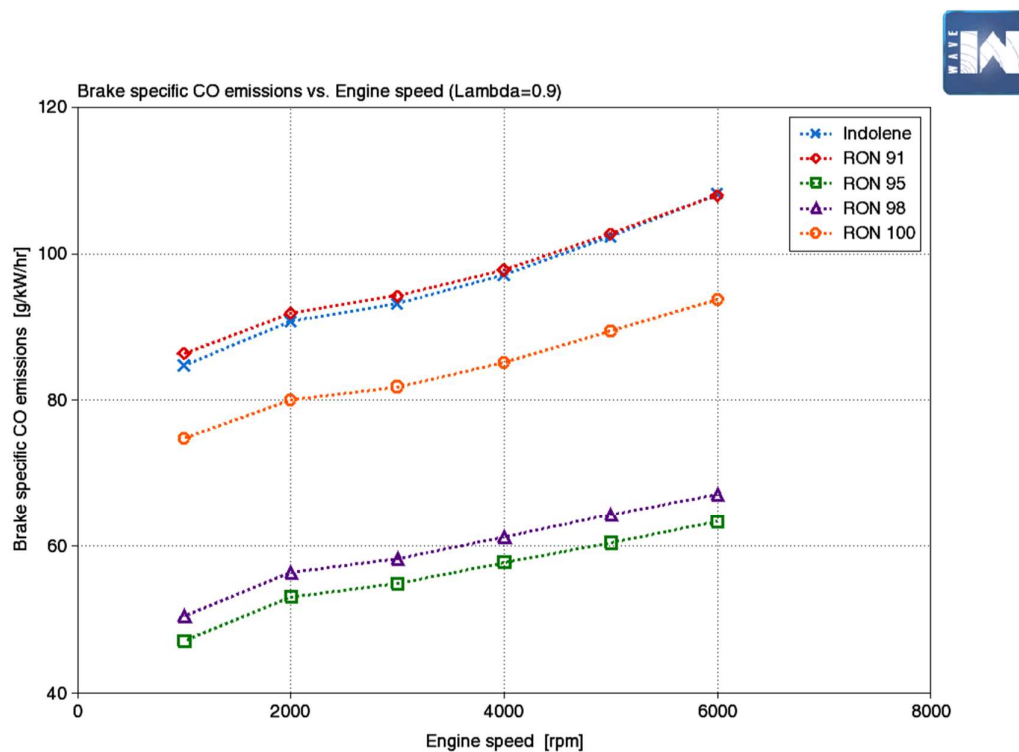
Table 28 : Exhaust gas temperature data from WAVE combustion model performance analysis of different fuels ($\lambda = 1.1$)

Graph 34 concludes that fuel characteristics change when the air-fuel mixture is controlled. The exhaust gas temperature of all fuels is decreased at all engine speeds for the lean mixture. For a lean mixture, RON 91 still generates the highest exhaust temperature. This is due to the highest heating value of RON 91 over other fuels which is discussed in the same chapter.

The lowest exhaust gas temperatures are found in RON 98 and RON 95. Despite the difference in the octane number between RON 100 and indolene, both fuel molecules have similar exhaust gas temperatures. According to the results, RON 91 has an exhaust gas temperature of 1376 kelvin for λ 0.9 compared to 1326 kelvin for λ 1.1.

5.1.3 Simulation analysis of influence of Research Octane Number (RON) on ford sigma engine emissions

5.1.3.1 CO Emissions

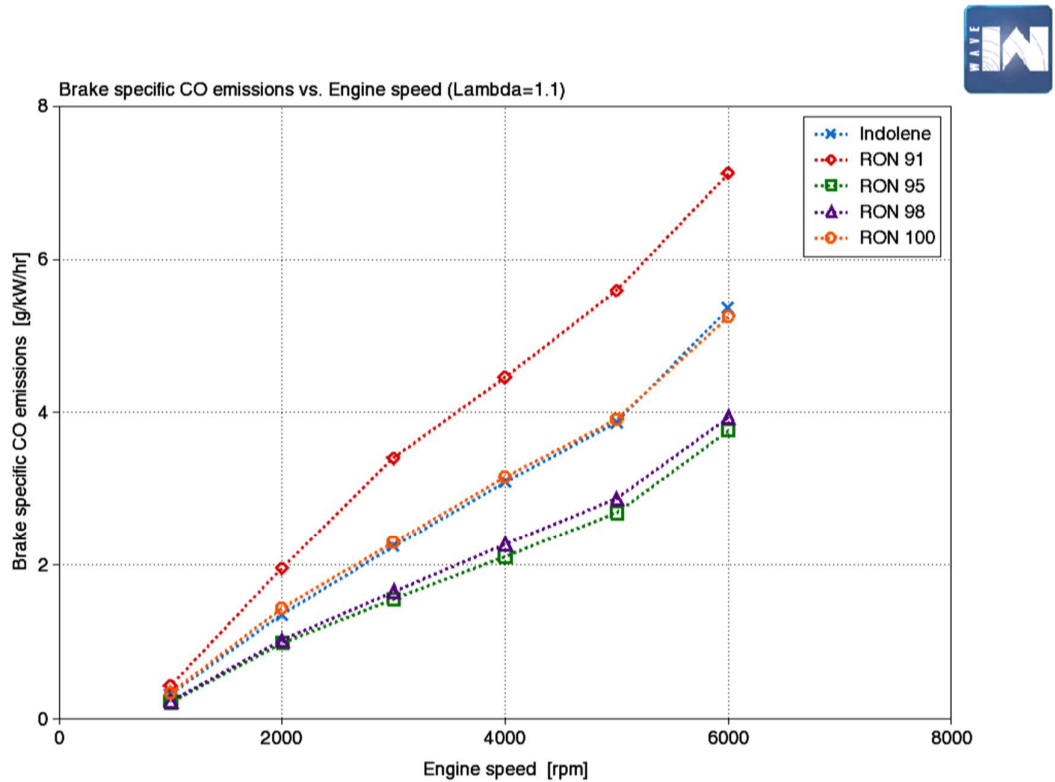


Graph 35: Impact of RON on Brake specific carbon monoxide for rich air fuel mixture ($\lambda = 0.9$)

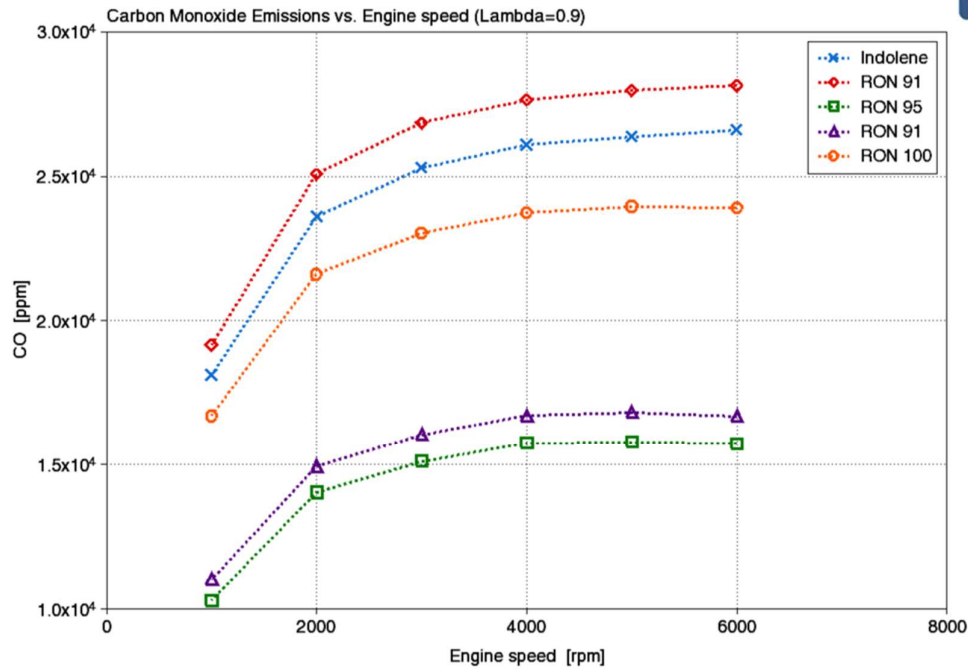
Graph 38 discusses the brake-specific carbon monoxide emissions for λ 0.9 and it records highest for RON 91 and indolene. CO emissions are directly proportional to the amount of fuel burned and they increase with respect to engine speed. RON 91 has an emission of 107.95 g/kW/hr at 6000 RPM 14.17 g/kW/hr more than RON 100 in the category of high-performance fuel.

RON 98 and RON 95 have the least BSCO emissions as RON 98 generates 67.15 g/kW/hr and RON 95 produces 63.39 g/kW/hr in a 720-degree crank cycle. It is also found that at almost all engine operating conditions, RON 91 produces 44.55 g/kW/hr, 41.27 % increased BSCO than RON 95.

Graph 36 discussed the BSCO for lean air fuel mixture ($\lambda=1.1$) and it concludes that lean mixture has very less influence on CO emissions. RON 91 has 100.85 g/kW/hr less BSCO when the air fuel mixture turns from rich($\lambda=0.9$) to lean ($\lambda=1.1$). RON 100 and indolene has similar contribution to BSCO where RON 98 and RON 95 have the least emission of all fuels taken for analysis.



Graph 36 : Impact of RON on Brake specific carbon monoxide for lean air fuel mixture ($\lambda=1.1$)

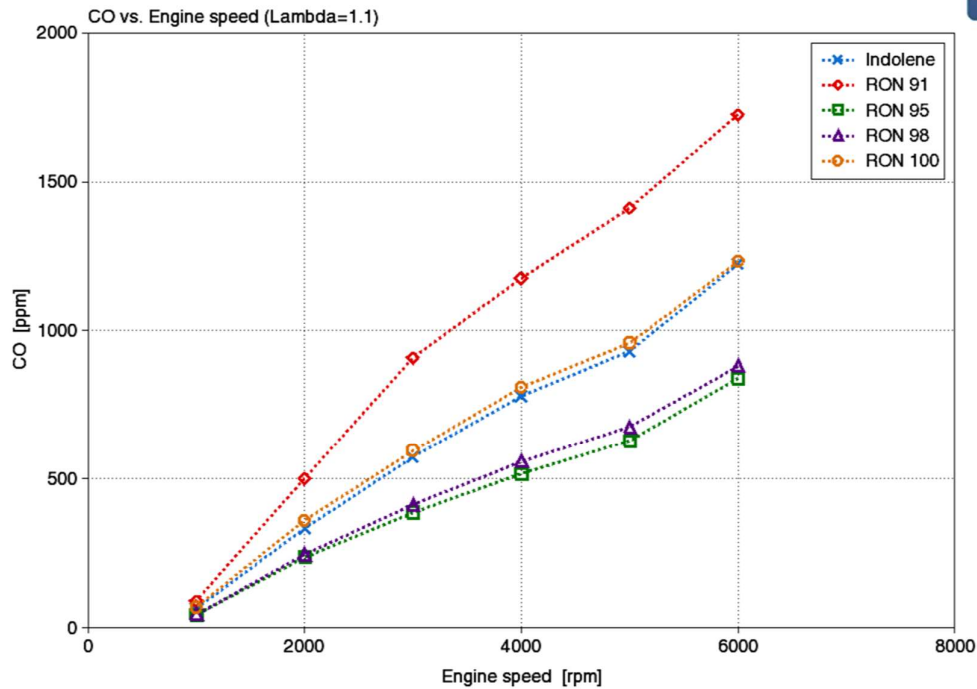


Graph 37 : Impact of RON on carbon monoxide emissions for rich air fuel mixture (Lambda=0.9)

Graph 37 depicts that CO emissions increases gradually and supports the data from graph 32 concluding that when the mixture is rich (lambda=0.9), RON 91 produces 28133 ppm of CO at 6000 RPM whereas when the mixture is lean as in graph 35, RON 91 produces 1724 ppm. At 6000 RPM RON 100 produces 1232 ppm of CO slightly more than 1188 ppm of indolene, followed by RON 98 and 95 with least emissions during combustion of rich and lean mixture.

CO Emissions vs Engine speed ($\lambda=0.9$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	ppm	ppm	ppm	ppm	ppm
1000	18141	19183	10304	11035	16711
2000	23608	25077	14048	14949	21612
3000	25291	26858	15124	16065	23033
4000	26087	27647	15760	16716	23734
5000	26368	27969	15794	16839	23962
6000	26602	28133	15750	16709	23908

Table 29 : CO data from WAVE combustion model emission analysis of different fuels ($\lambda=0.9$)

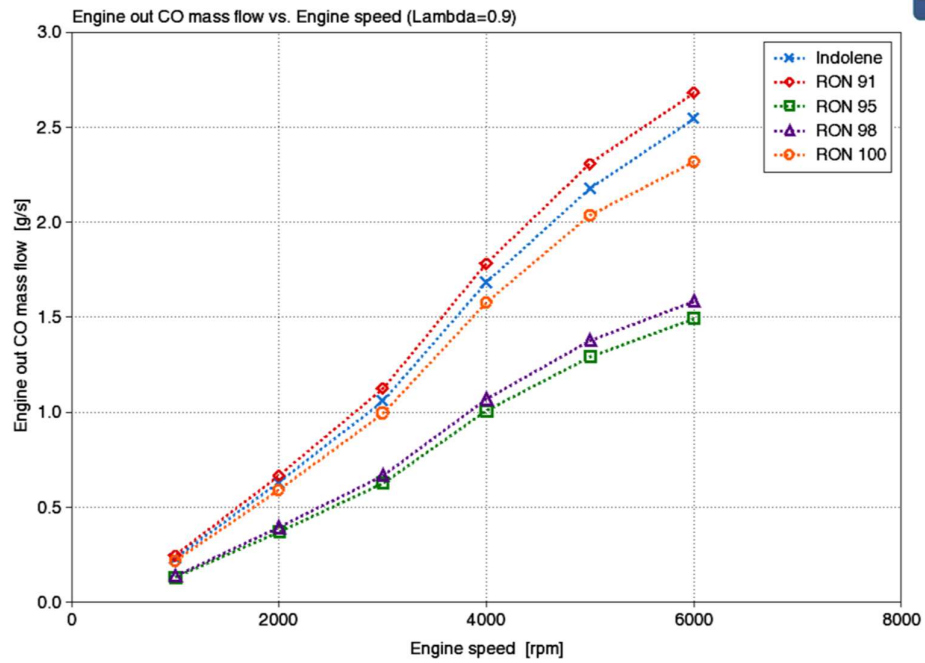


Graph 38: Impact of RON on carbon monoxide emissions for lean air fuel mixture ($\lambda=0.9$)

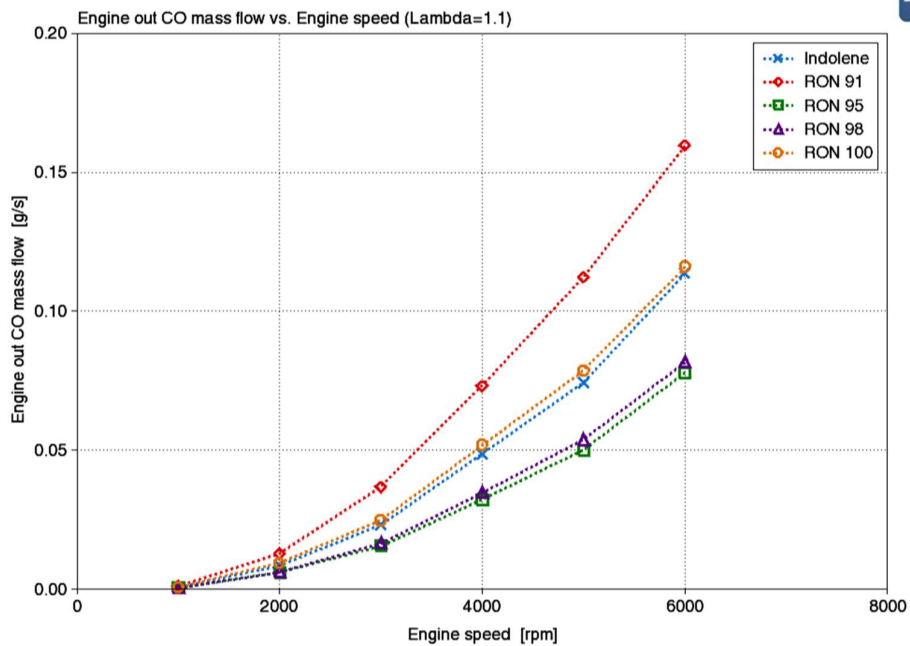
Graph 39 and 40 discusses about the mass of the outflow of CO at different engine operating conditions. As previously discussed from Graph 32 to 35 the amount of CO released to atmosphere is high for combustion of rich mixture, whereas lean mixture contributes a small proportion of CO compared to that of rich mixture. Since RON 95 and 98 are recorded to have least composition of CO emissions for both mixture composition, Indolene produces more CO emissions than RON 100 during combustion of rich mixture and lesser during combustion of lean mixture. The data from brake specific emissions to engine out mass flow of CO concludes that lean mixture controls the CO emissions, whereas there might be a considerable compensation in engine performance.

CO Emissions vs Engine speed ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	ppm	ppm	ppm	ppm	ppm
1000	66	92	44	46	71
2000	334	502	237	246	361
3000	575	910	388	413	597
4000	780	1176	518	560	811
5000	930	1411	628	675	959
6000	1189	1725	839	882	1233

Table 30 : CO data from WAVE combustion model emission analysis of different fuels ($\lambda=1.1$)

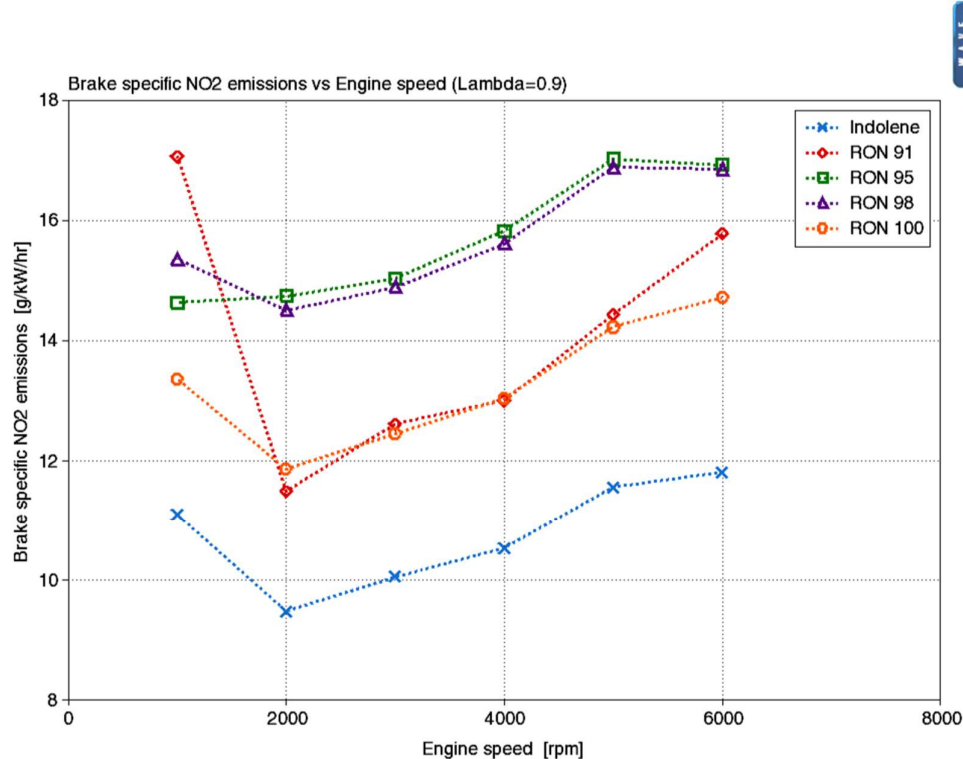


Graph 39: Impact of RON on engine out carbon monoxide mass flow rate for rich air fuel mixture ($\lambda = 0.9$)



Graph 40: Impact of RON on engine out carbon monoxide mass flow rate for lean air fuel mixture ($\lambda = 1.1$)

5.1.3.2 NOx Emissions

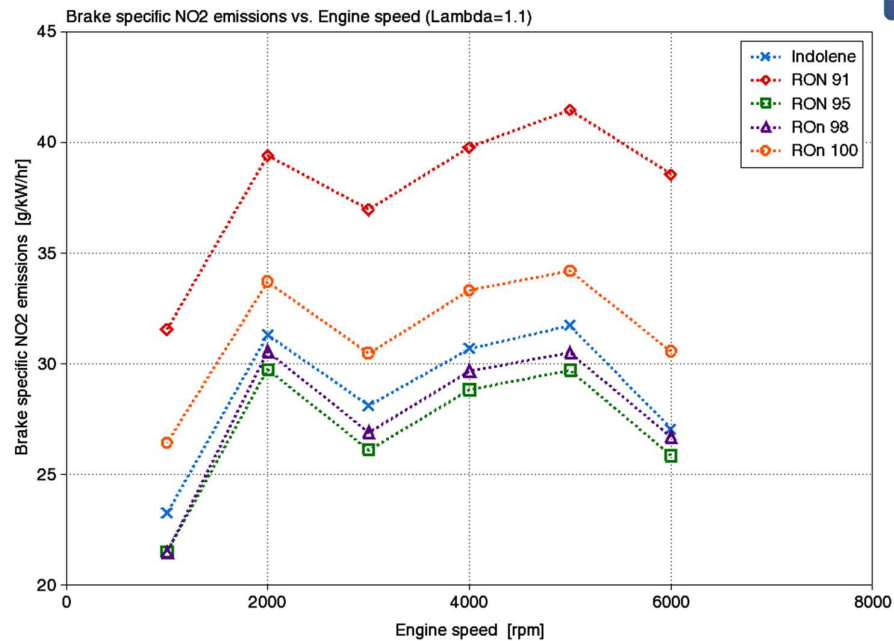


Graph 41: Impact of RON on brake specific NO2 emissions for rich air fuel mixture ($\lambda=0.9$)

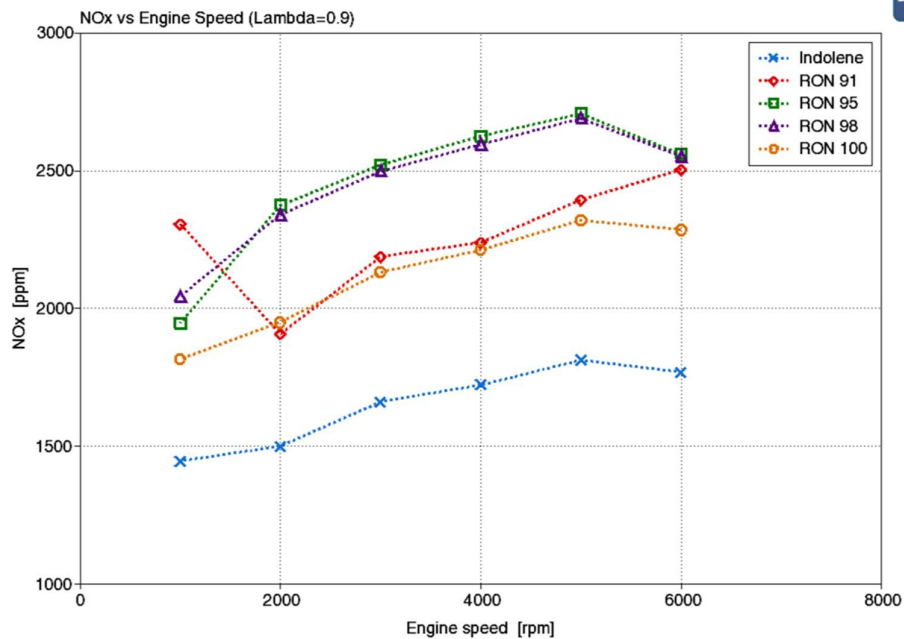
Graph 41 and 42 discuss about the combustion of rich and lean AFR of different fuels. During the combustion of rich AFR, study shows that RON 95 and 98 records the highest contribution towards the brake specific NO2 emissions with 16.92 and 15.78 g/kW/hr. RON 91 and RON 100 have similar characteristics with respect to brake specific nitrogen dioxide emissions, highest recorded for RON 91 at 1000, 3000, 5000 and 6000 RPM. Indolene has the least contribution with 5.10 g/kW/hr emissions lesser than RON 95 at 6000 RPM.

During the combustion of lean AFR, brake specific NO2 are recorded high for RON 91 followed by RON 100, indolene, RON 98 and RON 95. Highest emissions are recorded at 5000 RPM where RON 91 contributes 38.54 g/kW/hr, 12.69 g/kW/hr higher than RON 95 emissions. This concludes that RON 95 and 98 mixtures have higher chance of contribution towards brake specific NO2 emissions during the combustion of rich AFR.

Graph 40 and 41 gives a detailed analysis on the amount of composition of oxides of nitrogen, NOx is released during wide engine operating conditions for the rich and lean AFR.



Graph 42: Impact of RON on brake specific NO₂ emissions for lean air fuel mixture ($\lambda = 1.1$)



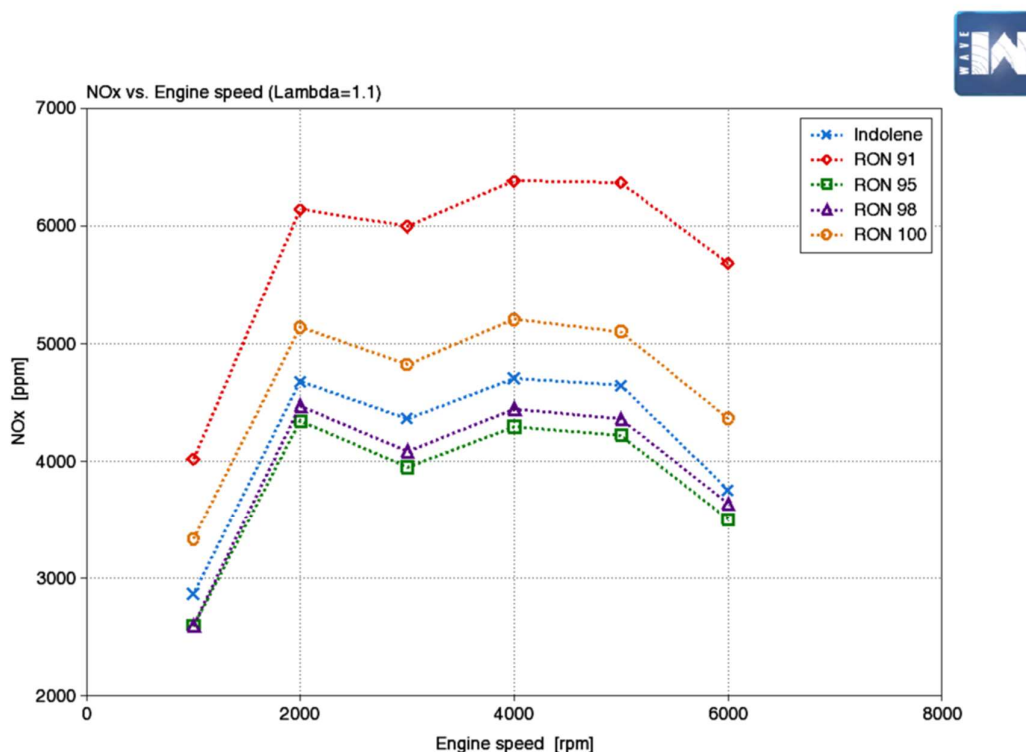
Graph 43: Impact of RON on NO_x emissions for rich air fuel mixture ($\lambda = 0.9$)

NOx Emissions vs Engine speed ($\lambda=0.9$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	ppm	ppm	ppm	ppm	ppm
1000	1447	2306	1946	2044	1817
2000	1501	1909	2375	2341	1950
3000	1662	2189	2521	2499	2134
4000	1724	2239	2625	2595	2212
5000	1814	2394	2708	2691	2320
6000	1769	2504	2560	2552	2285

Table 31 : NOx data from WAVE combustion model emission analysis of different fuels ($\lambda=0.9$)

As discussed in graph 43 for the analysis of combustion of rich AFR, NOx has been recorded to have highest for RON 95 at 5000 RPM with a value of 2707 ppm followed by 2691 ppm for RON 98. RON 91 is expected to have increase in NOx while increasing engine speed and is recorded to have 2504 ppm at 6000 RPM, 55 ppm lesser than RON 95.

Graph 44 depicts the analysis of combustion of rich AFR and its influence on NOx emissions, RON 91 is recorded to have highest emissions at 4000 and 5000 RPM with 6379 and 6367 pp. Since the influence of rich mixture has increase in emissions for RON 95, for lean conditions RON 95 has the least NOx emissions with a difference of 2084 ppm at 4000 RPM.

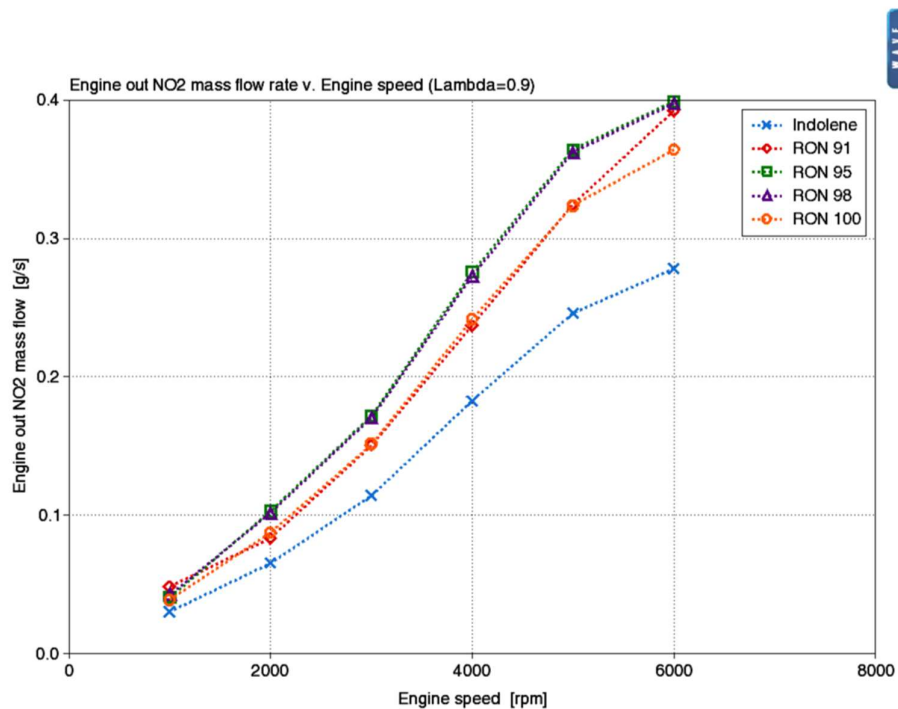


Graph 44: Impact of RON on brake specific NO2 emissions for rich air fuel mixture ($\lambda=1.1$)

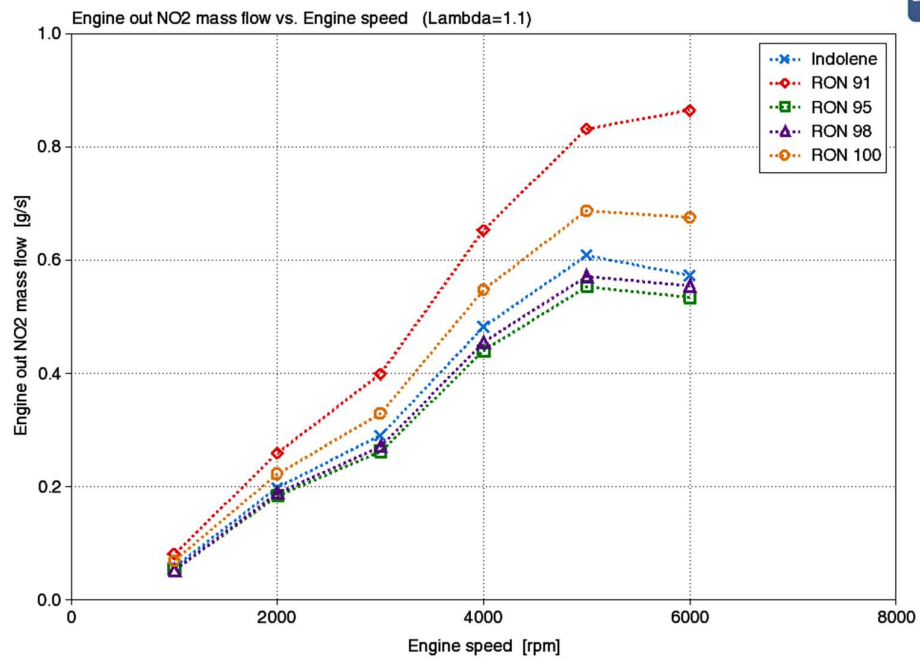
NOx Emissions vs Engine speed ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	ppm	ppm	ppm	ppm	ppm
1000	2868	4019	2593	2597	3335
2000	4679	6136	4342	4471	5139
3000	4366	5998	3950	4085	4821
4000	4707	6380	4295	4444	5205
5000	4642	6368	4222	4359	5103
6000	3949	5682	3505	3637	4364

Table 32 : NOx data from WAVE combustion model emission analysis of different fuels ($\lambda=1.1$)

Graph 45 and 46 concludes that RON 95 releases a mass flow rate of .39 g/s of NO₂ for a rich AFR whereas RON 91 releases .86 g/s of NO₂ for a lean combustion. Indolene release more NO_x than RON 95 and 98 when the mixture is lean and the least production of all other fuels when the mixture is rich.

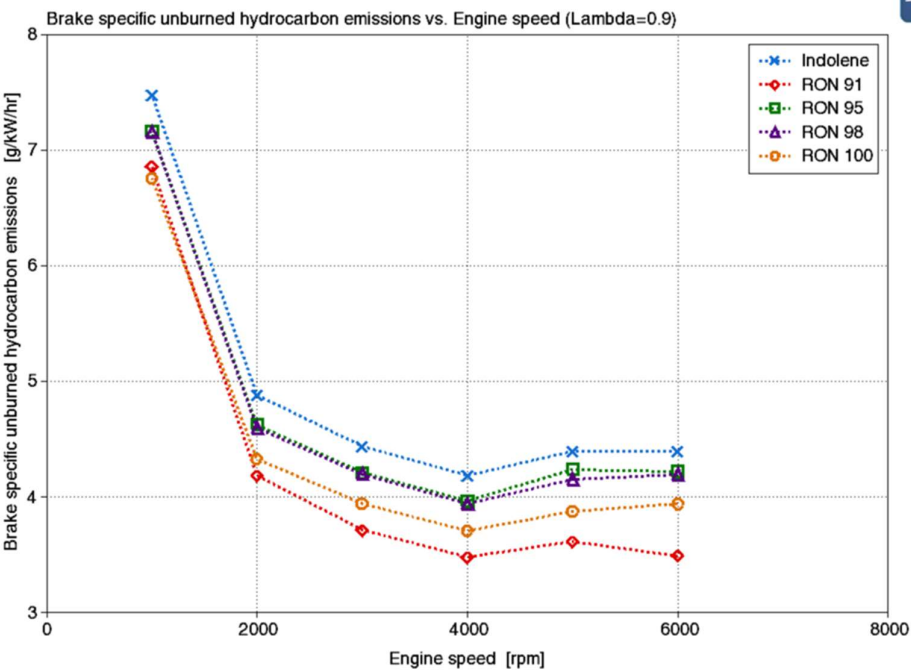


Graph 45 : Impact of RON on engine out NO₂ mass flow rate for rich air fuel mixture ($\lambda=0.9$)

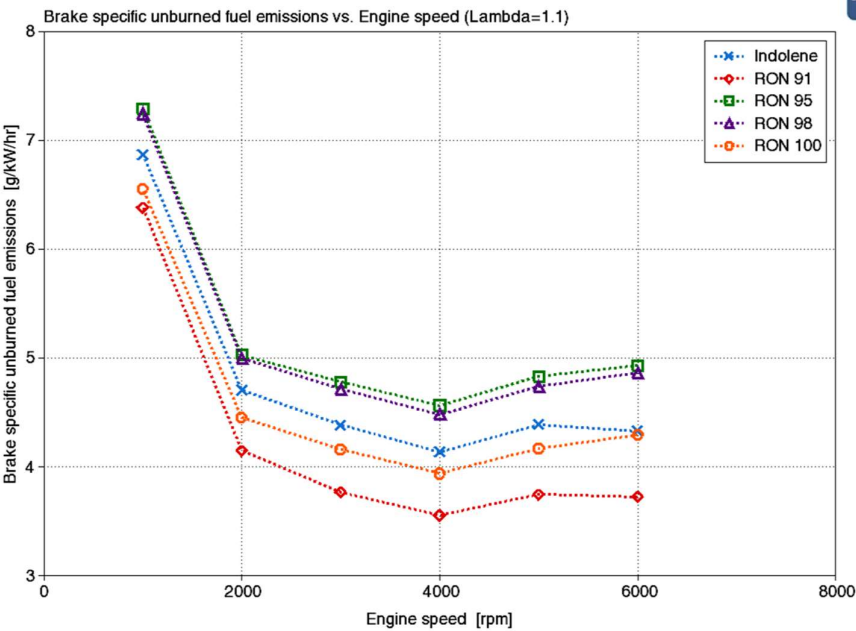


Graph 46: Impact of RON on engine out NO₂ mass flow rate for lean air fuel mixture ($\lambda = 1.1$)

5.1.3.3 Hydrocarbon Emissions



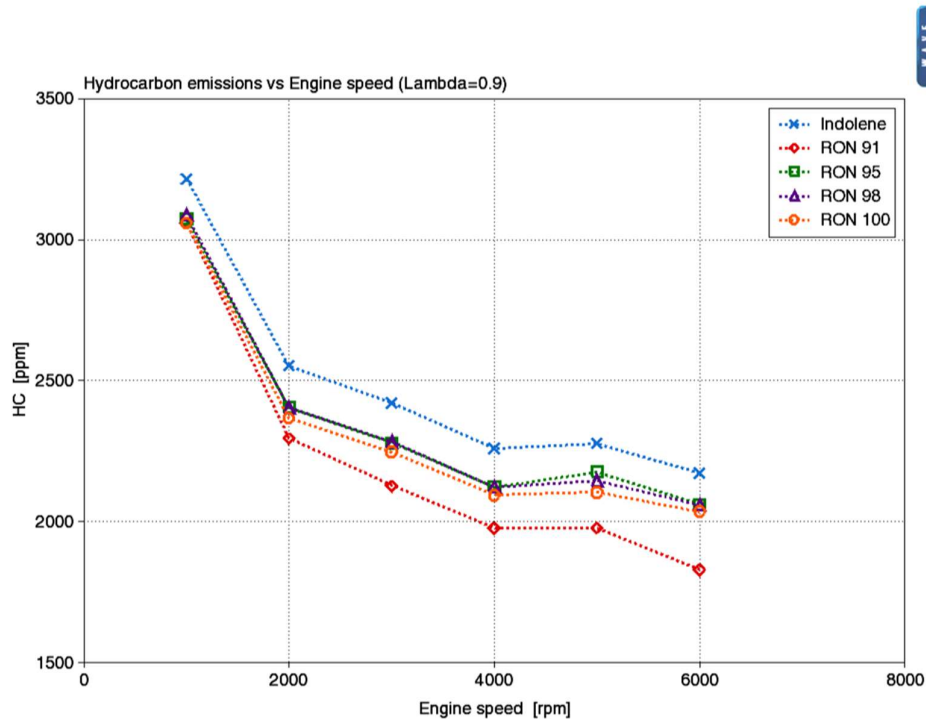
Graph 47 : Impact of RON on brake specific hydrocarbon emissions for rich air fuel mixture ($\lambda = 0.9$)



Graph 48: Impact of RON on brake specific hydrocarbon emissions for lean air fuel mixture ($\lambda = 1.1$)

Graph 47 and 48 describes about the brake specific hydrocarbon emissions for rich and lean AFR. It has been noted that the amount of hydrocarbon emissions in both AFR are literally similar in consumption. Lowest the engine speed, highest the hydrocarbon emissions. Indolene is recorded to have highest brake specific emissions when AFR is rich, followed by RON 95 and RON 98 having similar consumption with 7.16 and 7.15 g/kW/hr respectively at 1000 RPM. RON 100 has more BSHE compared to RON 91 which is the least contributor towards BSHE.

Graph 48 concludes that RON 95 and 98 have highest significance on BSHE when AFR is lean followed by indolene, RON 100 and RON 91. This characteristics in BSHE concludes that the heating value of fuel has a fundamental influence on the unburned fuel. Higher the heating value, fewer the hydrocarbon emissions with more combustion of fuel mixture.

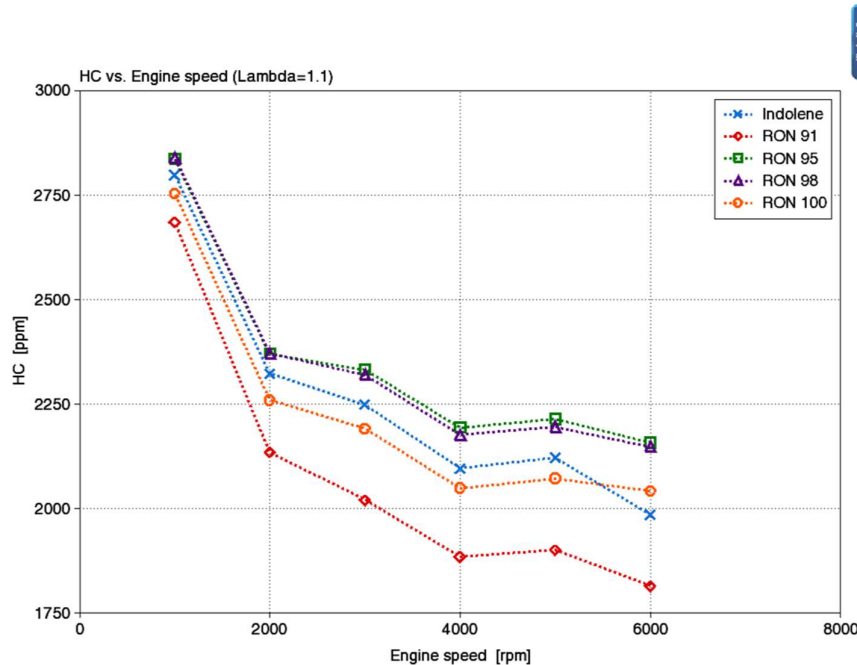


Graph 49 : Impact of RON Hydrocarbon emissions for rich air fuel mixture ($\lambda=0.9$)

HC Emissions vs Engine speed ($\lambda=0.9$)		Indolene	RON 91	RON 95	RON 98	RON 100
RPM	ppm	ppm	ppm	ppm	ppm	ppm
1000	3216	3060	3074	3086	3058	
2000	2553	2295	2406	2404	2368	
3000	2421	2128	2278	2284	2248	
4000	2278	1979	2177	2146	2104	
5000	2260	1978	2124	2123	2094	
6000	2173	1831	2060	2059	2037	

Table 33 : Hydrocarbon data from WAVE combustion model emission analysis of different fuels ($\lambda=0.9$)

Graph 49 concludes the behaviour of RON in fuel for production of HC emissions during combustion of rich AFR. Indolene has the highest hydrocarbon emissions at 1000 RPM with 3216 ppm followed by close emissions for RON 91,95,98, and 100. When the engine speed increases, RON 91 has the least contribution towards HC, followed by RON 100, RON 98, and RON 95. This wide spread characteristics of HC in the increasing operating condition is strictly related to the burn characteristics of the fuel. Typically, RON 95 and RON 98 have more contribution towards HC emissions. This is widely addressed during the combustion of lean air fuel mixture in graph 47 as RON 95 produces 2836 ppm at 1000 RPM 342 ppm more than RON 91 at 1000 RPM.

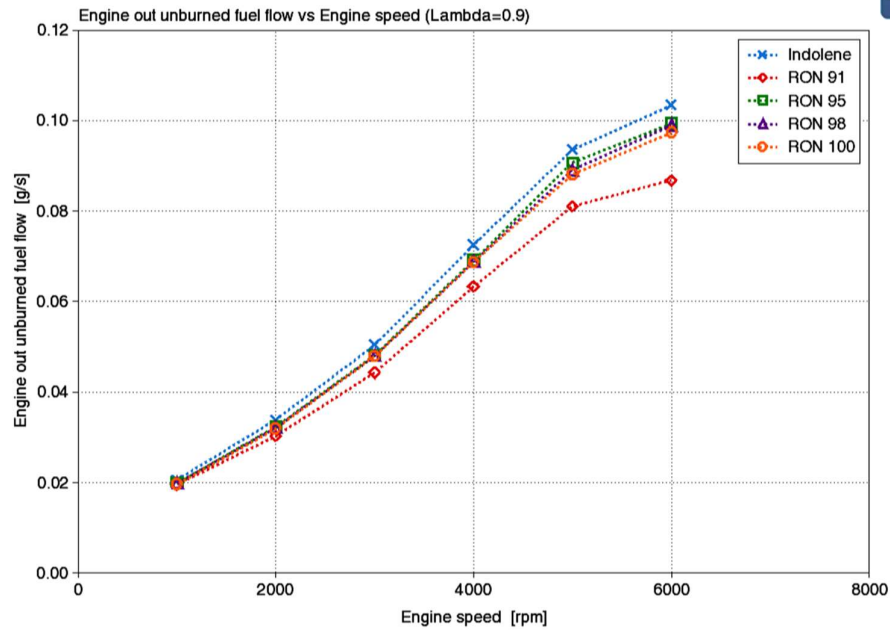


Graph 50: Impact of RON on Hydrocarbon emissions for lean air fuel mixture ($\lambda=1.1$)

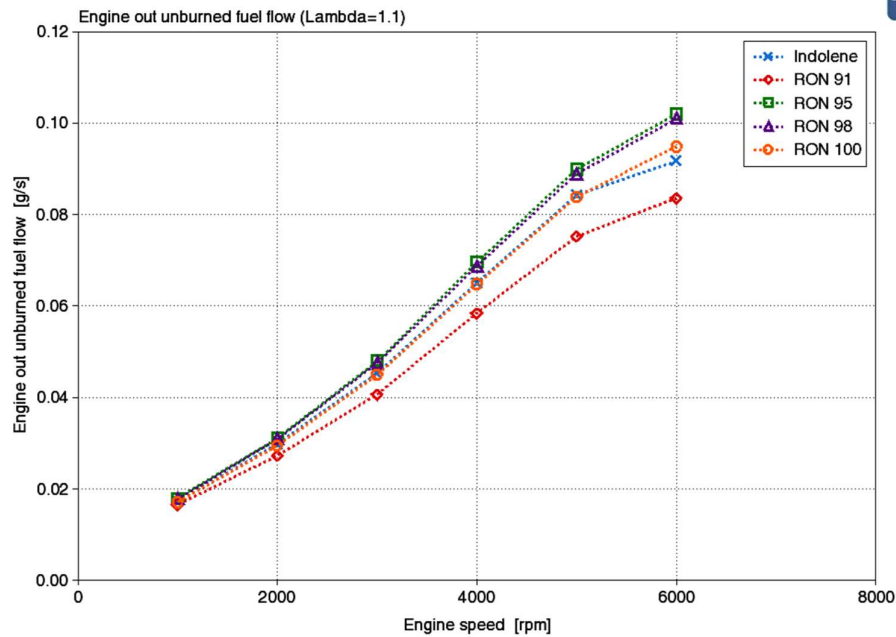
HC Emissions vs Engine speed ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	ppm	ppm	ppm	ppm	ppm
1000	2798	2685	2836	2839	2753
2000	2323	2136	2370	2371	2260
3000	2248	2021	2333	2320	2191
4000	2096	1885	2194	2176	2050
5000	2123	1902	2214	2196	2073
6000	2060	1816	2158	2148	2042

Table 34 : Hydrocarbon data from WAVE combustion model emission analysis of different fuels ($\lambda=1.1$)

Graph 51 and 52 discusses about the engine out unburned hydrocarbon mass flow rate during rich and lean combustion. Indolene records the highest unburned fuel flow with .1 g/s at 6000 RPM over RON 91, which is the least contributor of the fuel taken for study with .086 g/s at 6000 RPM. In lean combustion, RON 95 and RON 98 have the highest contribution towards engine out unburnt fuel flow with.1 g/s over indolene RON 991 and RON 100.

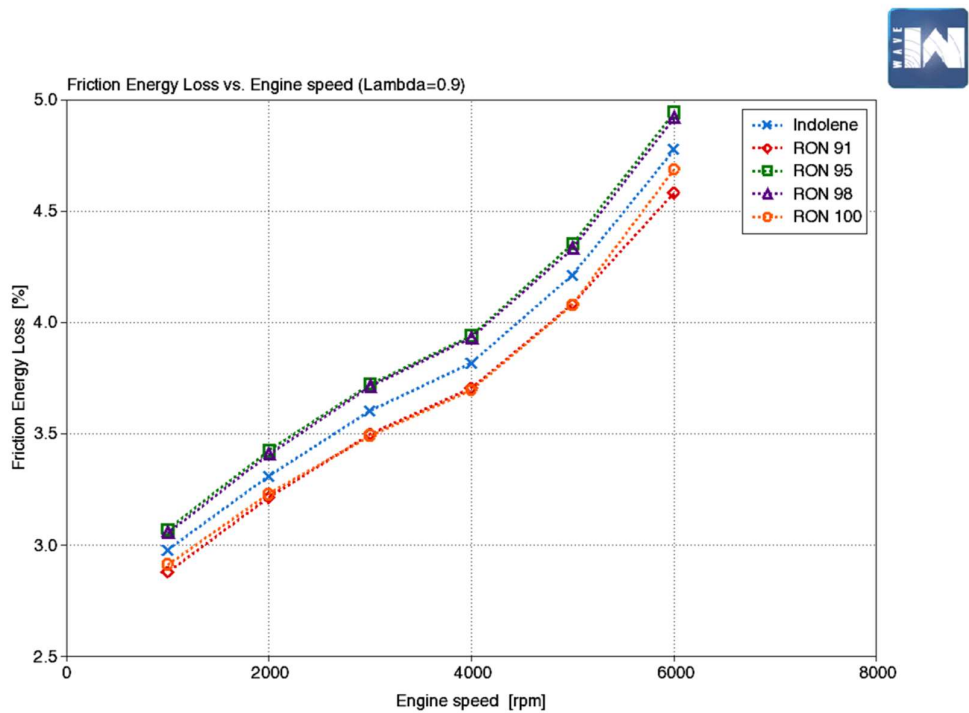


Graph 51: Impact of RON on engine out unburned fuel flow rate for rich air fuel mixture ($\lambda = 0.9$)

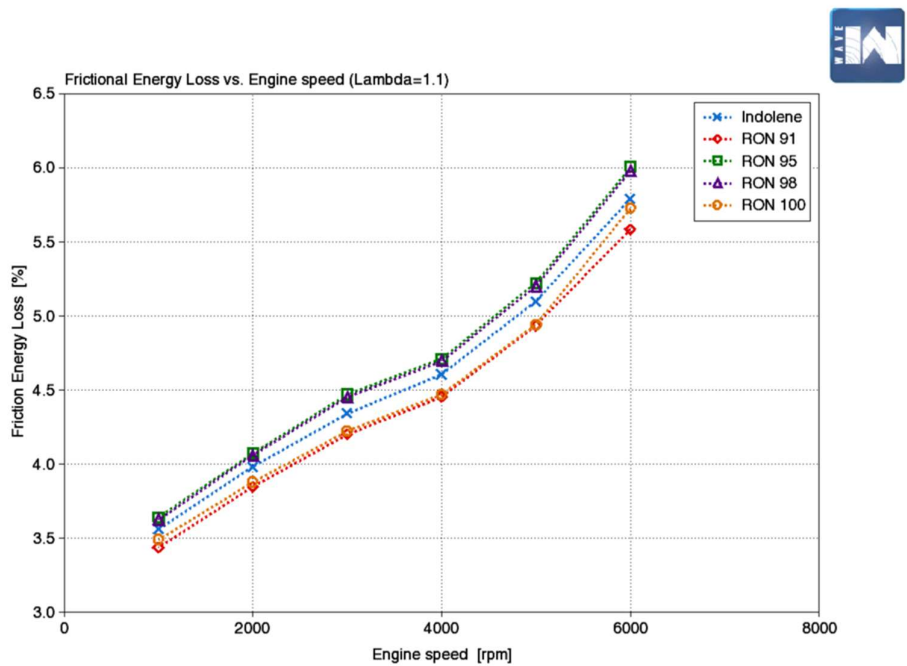


Graph 52 : Impact of RON on engine out unburned fuel flow rate for lean air fuel mixture ($\lambda = 1.1$)

5.1.4 Frictional performance and losses



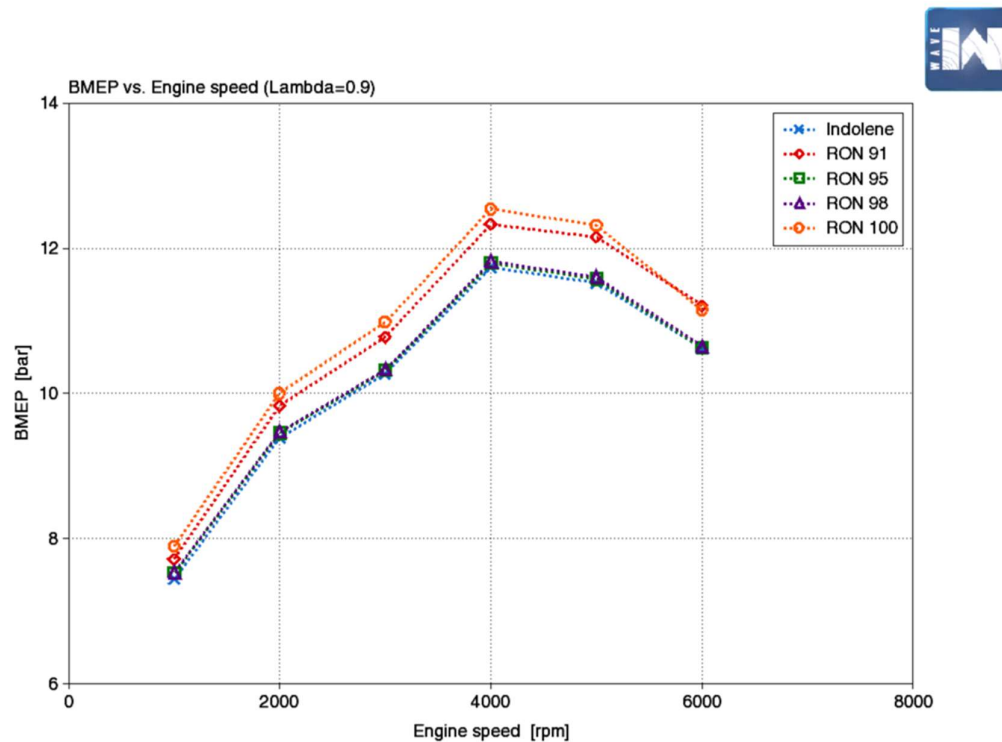
Graph 53: Impact of RON on Friction Energy Loss for rich air fuel mixture ($\lambda = 0.9$)



Graph 54: Impact of RON on Friction Energy Loss for lean air fuel mixture ($\lambda = 1.1$)

From a performance perspective, graph 53 outlines about the influence of RON on the frictional energy loss during combustion of rich fuel mixture. RON 95 and RON 98 have the highest energy loss due to friction followed by indolene, RON 100 and RON 91. At an engine speed of 6000 RPM, RON 95 has a loss of 4.94% where RON 91 has 4.58 %, this varying characteristic of fuel concludes that RON have a contrasting behaviour in frictional losses during combustion. Graph 54 outlines about the combustion characteristics of lean air fuel mixture, where the frictional losses are high in all engine operating conditions.

5.1.5 Pumping Performance



Graph 55: Impact of RON on Brake Mean Effective Pressure for rich air fuel mixture ($\lambda=0.9$)

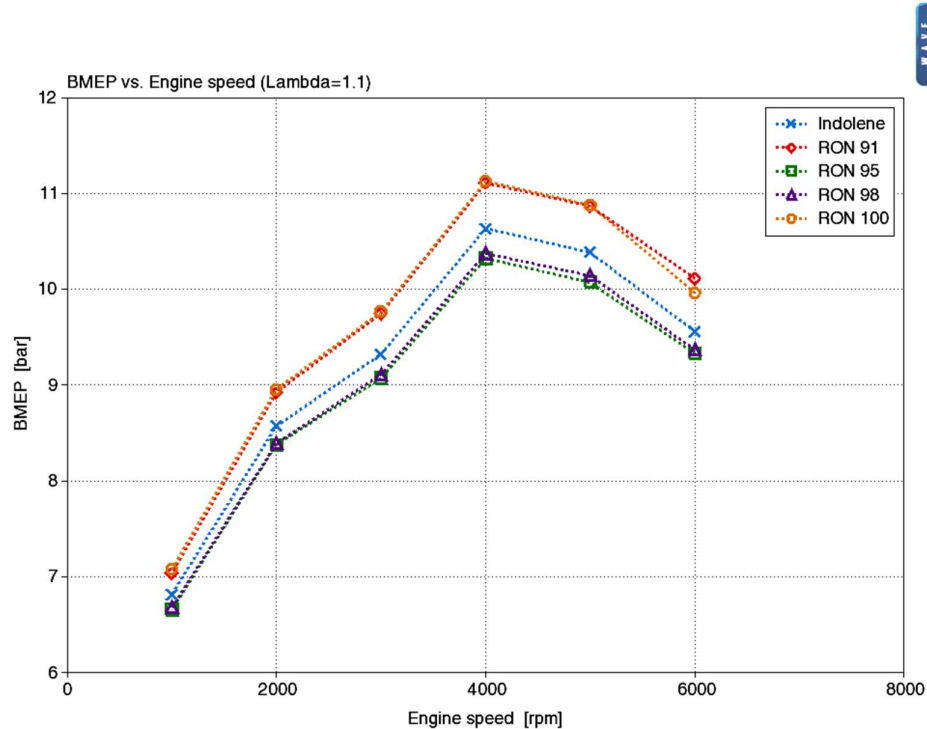
In chapter 5.1.2, the effect of RON on the linear P-V diagram for rich and lean mixture were discussed and the data concludes that RON have an impact on the net pressure developed in the combustion zone. In order to produce the measured (brake) power output, the pistons would be subjected to a uniform mean pressure throughout each power stroke defines as Brake Mean Effective Pressure (BMEP) (Inc., 2019). It simply measures how efficiently a given engine produces torque based on its displacement.

Graph 55 discusses about the influence of RON in BMEP. RON 100 has highest conversion of energy output followed by RON 91 in all engine speed. Since BMEP is a function of torque, it is recorded highest at 4000 RPM where the torque is highest. For lambda 0.9, RON 100 has a BMEP of 12.53 bar followed by 12.33 bar of RON 91. RON 98 has third most effective BMEP, followed by RON 95, where indolene has least BMEP of 11.73 bar.

BMEP vs Engine speed ($\lambda=0.9$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	bar	bar	bar	bar	bar
1000	7.45	7.72	7.53	7.54	7.90
2000	9.39	9.83	9.45	9.47	10.00
3000	10.27	10.78	10.32	10.34	10.98
4000	11.74	12.33	11.80	11.82	12.54
5000	11.52	12.16	11.58	11.60	12.32
6000	10.62	11.21	10.63	10.64	11.15

Table 35 : BMEP data from WAVE combustion model performance analysis of different fuels ($\lambda=0.9$)

Graph 56 outlines the BMEP for fuels when lambda is lean. RON 100 and RON 91 has same performance throughout the operating conditions followed by indolene RON 98 and RON 95. Comparing the performance characteristic of RON 100 as high-performance fuel, when the mixture is 10% lean at 4000 RPM, it has a loss of 1.39 bar than that of 10 % rich mixture.

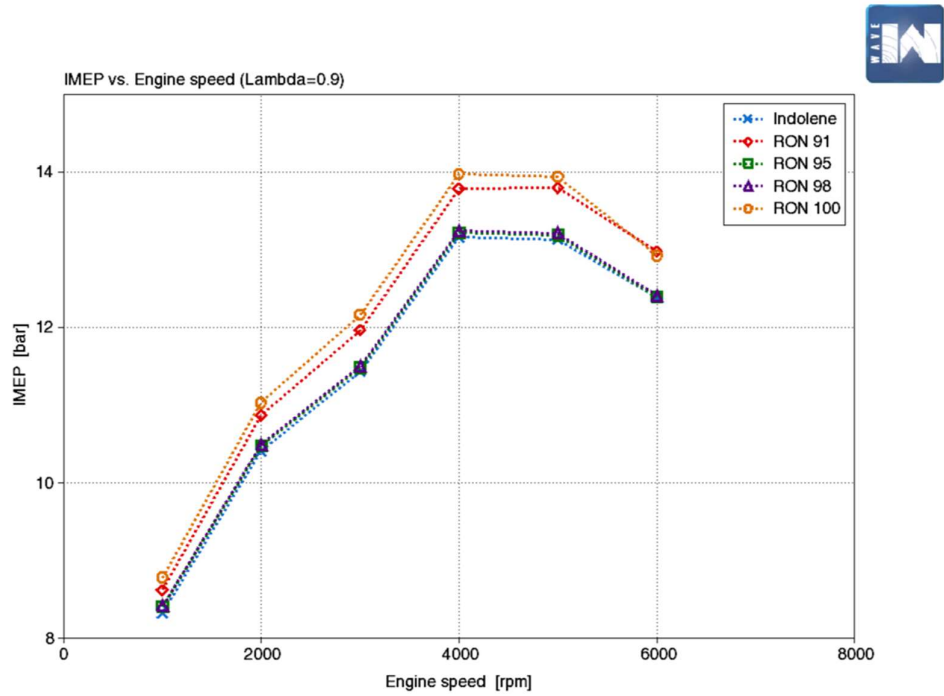


Graph 56 : Impact of RON on Brake Mean Effective Pressure for lean air fuel mixture ($\lambda=1.1$)

Like BMEP, Indicated Mean Effective Pressure (IMEP) is the mean effective pressure calculated with indicated power or work done. According to the indicated power discussed in chapter 5.1.1.1, IMEP is directly proportional to the fuel mixture and is highest when the mixture is rich over lean mixture with less indicated powerline is constant for all fuel mixtures between 4000 and 5000 RPM. Graph 57 discusses about the IMEP of rich fuel mixtures. RON 100 has the highest IMEP at 4000 RPM with 13.97 bar whereas indolene has least IMEP with 13.16 bar. When the mixture is air fuel mixture is lean, the study shows that IMEP and BMEP have similar characteristics as they are the function of performance, Indolene with least IMEP in rich conditions have more IMEP than RON 95 and RON 98 in lean mixture which are mentioned on graph 58.

BMEP vs Engine speed ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	bar	bar	bar	bar	bar
1000	6.82	7.04	6.66	6.68	7.08
2000	8.57	8.92	8.38	8.40	8.96
3000	9.33	9.75	9.07	9.11	9.77
4000	10.63	11.11	10.32	10.38	11.13
5000	10.39	10.87	10.08	10.15	10.88
6000	9.64	10.11	9.33	9.37	9.96

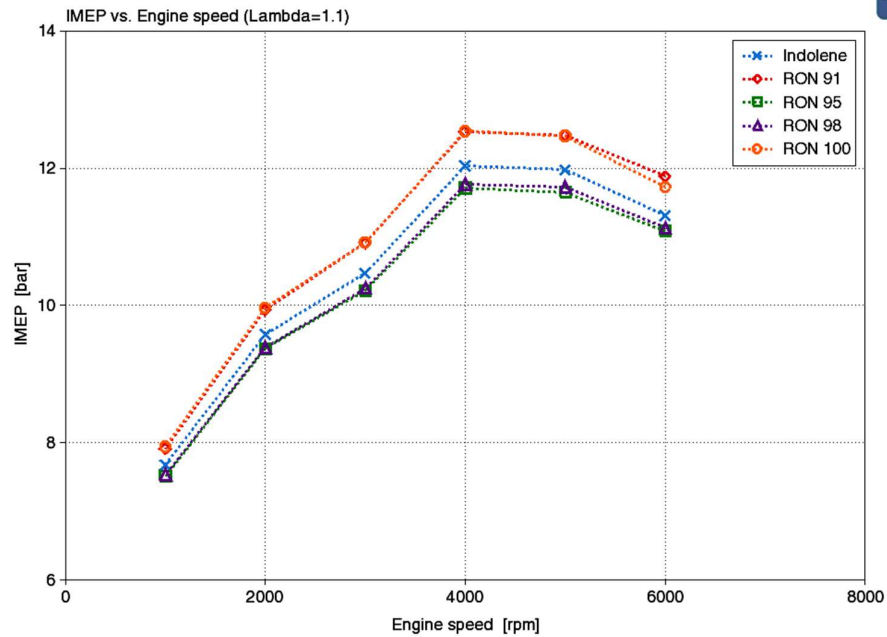
Table 36 : BMEP data from WAVE combustion model performance analysis of different fuels ($\lambda=1.1$)



Graph 57 : Impact of RON on Indicated Mean Effective Pressure for rich air fuel mixture ($\lambda=0.9$)

IMEP vs Engine speed ($\lambda=0.9$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	bar	bar	bar	bar	bar
1000	8.33	8.62	8.41	8.42	8.79
2000	10.41	10.87	10.47	10.49	11.03
3000	11.43	11.97	11.48	11.50	12.16
4000	13.16	13.78	13.22	13.24	13.98
5000	13.13	13.80	13.19	13.21	13.93
6000	12.38	12.98	12.39	12.41	12.92

Table 37 : IMEP data from WAVE combustion model performance analysis of different fuels ($\lambda=0.9$)



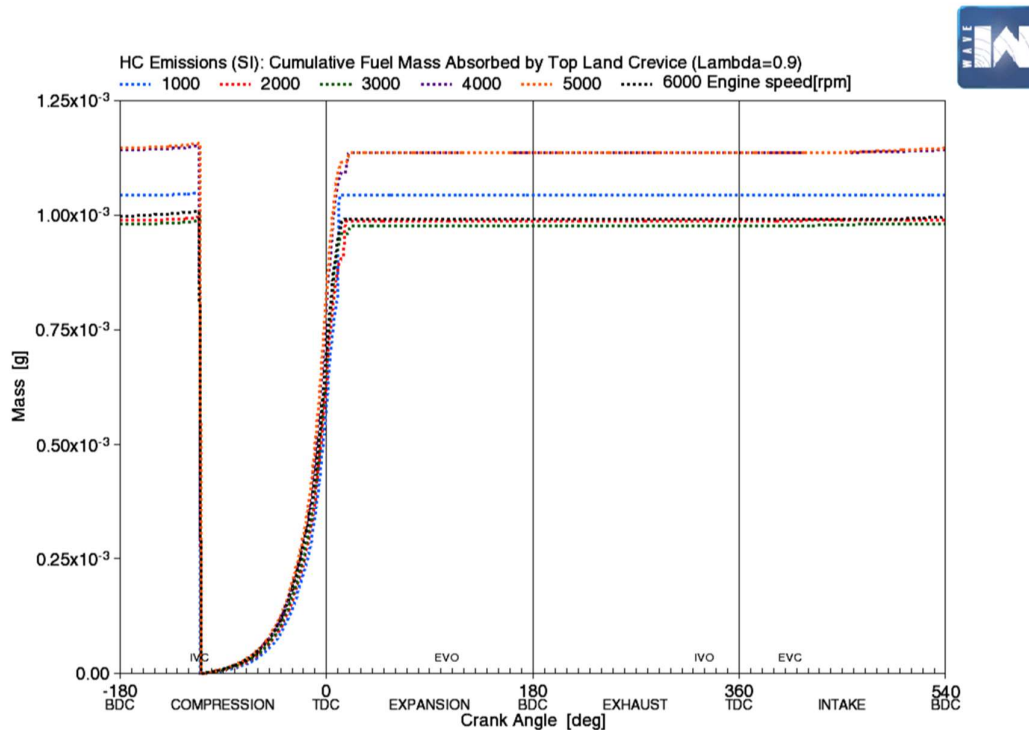
Graph 58 : Impact of RON on Indicated Mean Effective Pressure for lean air fuel mixture ($\lambda=1.1$)

IMEP vs Engine speed ($\lambda=1.1$)	Indolene	RON 91	RON 95	RON 98	RON 100
RPM	bar	bar	bar	bar	bar
1000	7.68	7.92	7.51	7.54	7.95
2000	9.58	9.94	9.37	9.39	9.96
3000	10.47	10.92	10.21	10.25	10.92
4000	12.04	12.54	11.71	11.76	12.54
5000	11.97	12.48	11.65	11.72	12.47
6000	11.40	11.88	11.09	11.13	11.73

Table 38 : IMEP data from WAVE combustion model performance analysis of different fuels ($\lambda=1.1$)

5.2 1-Dimensional combustion modelling analysis into molecular diffusion in piston top land crevice volume and hydrocarbon emissions

5.2.1 Cumulative fuel mass absorbed by Top land crevice and Oil layer

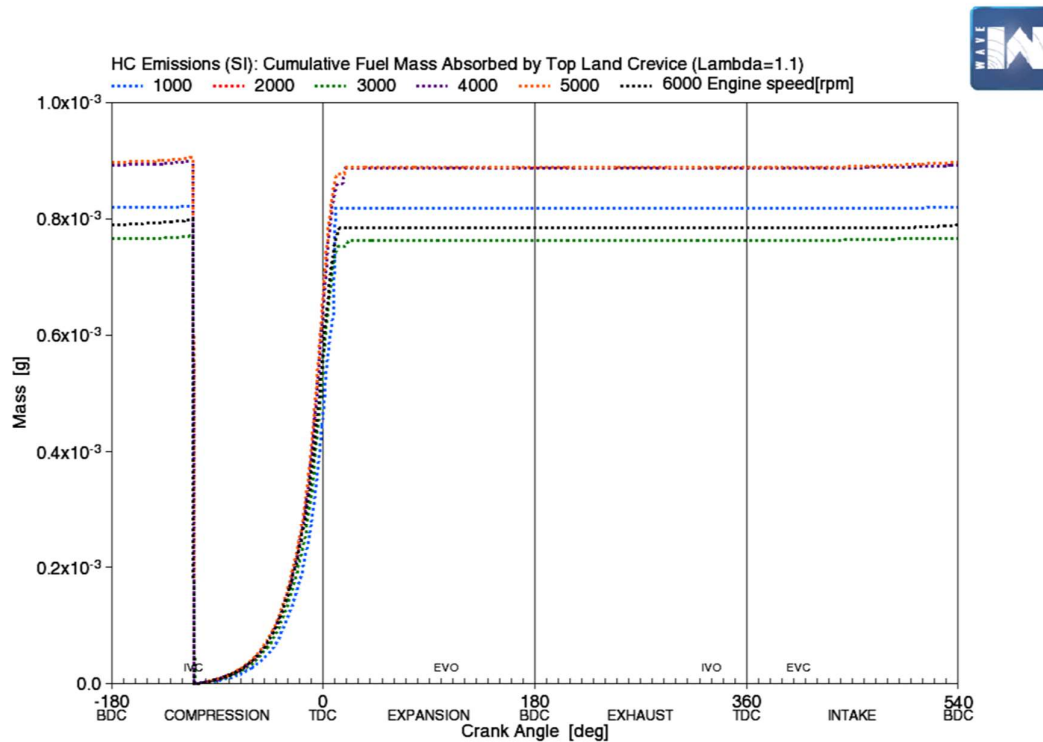


Graph 59: Cumulative fuel mass absorbed by top land crevice during combustion of rich air-fuel mixture($\lambda=0.9$)

Graph 59 discusses about the amount of unburned hydrocarbon emissions in the piston top land crevice volume at wide operating conditions for a 10 % rich air fuel mixture. The data outlines that the absorption rate is higher at 4000 and 5000 RPM whereas the absorption rate is lower while accelerating between 1000 to 3000 RPM and at peak 6000 RPM. Data outlines about the mass of fuel absorbed as a function of crank angle.

The graph shows that the amount of unburned fuel in crevice are trapped between beginning of expansion stroke and IVC phenomena in compression stroke. This indicates that the crevice volume has a significant impact on crevice accommodation of unburned fuel mixtures. When the inlet valve is closed the unburned fuel is released into cylinder combustion zone due to turbulence and increase in pressure inside the cylinder. This gathered fuel mixture by oil layer is mixed with the injected air fuel mixture which fluctuates the AFR in every cycle. As the piston travels from BDC to TDC, fuel accumulation increases and a specific amount of unburned fuel mixture is noted at the beginning of expansion stroke gathered by top land crevice volume after IVC.

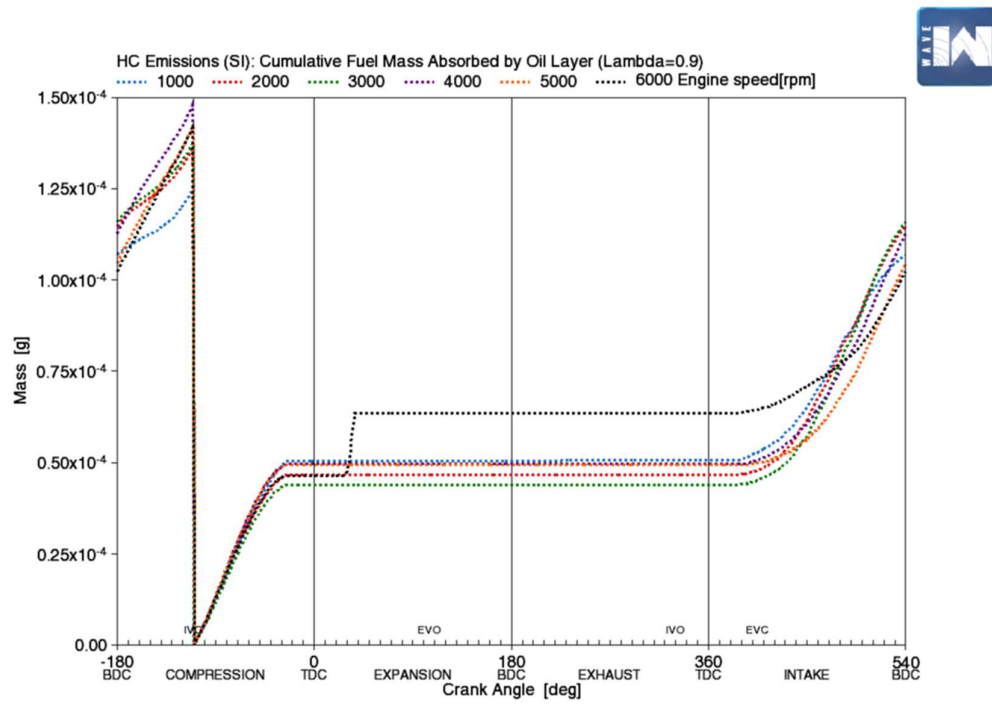
The data from graph 59 states that .001145 grams of fuel is trapped at 5000 RPM, .001142 grams of fuel is trapped at 4000 RPM, and the least is recorded at 3000 RPM .000981 grams.



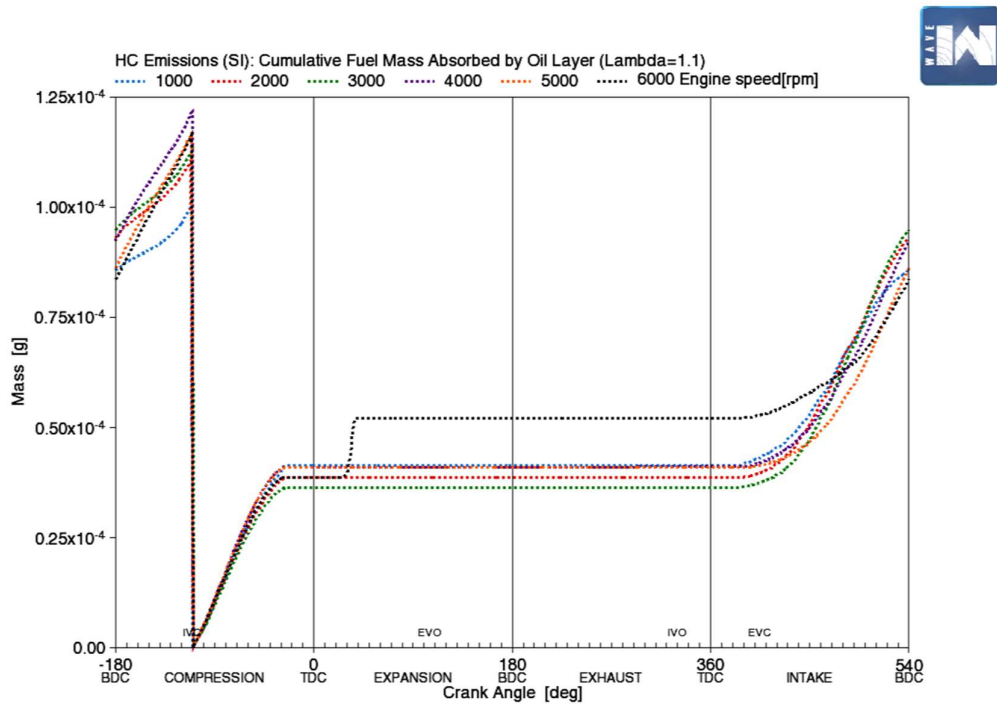
Graph 60 : Cumulative fuel mass absorbed by top land crevice during combustion of lean air-fuel mixture($\lambda=1.1$)

Graph 60 discusses about the amount of fuel absorption during the combustion of lean air-fuel mixture. The rate of fuel mixture trapped inside crevice volume are very less in quantity compared to that of rich air fuel mixture combustion. The least absorption is noted at 3000 RPM with .000766 grams of fuel, .00192 grams less mixture accumulated during combustion of rich air fuel mixture. Engine speed have a strong influence on the accumulation of unburned HC in crevice volume and is proportion even in rich and lean conditions. From graph 57, at 5000 RPM .000897 grams of fuel are absorbed, .000245 grams lesser than rich mixture.

The influence of lubricant layer on cylinder walls and the absorption of unburned fuel are studied using graph 61 and 62. The oil layer interaction with unburned fuel mixture during engine operating conditions shows that, absorption increases gradually during the intake stroke and compression stroke until the IVC. The pressure and turbulence again swept the unburned fuel to the combustion zone and as the piston compress from BDC to TDC more fuel is being absorbed until the piston reaches before TDC. From graph 61, during the compression of rich mixture the highest absorption in oil layer is noted at 4000 RPM which accumulates .000148 grams of fuel followed by 6000, 5000, 3000, 2000 and 1000 RPM. These characteristics conclude that the absorption of unburned HC increases with engine speed. At 6000 RPM from the beginning of expansion stroke till end of exhaust stroke, high rate of absorption is noted compared to other engine speed.



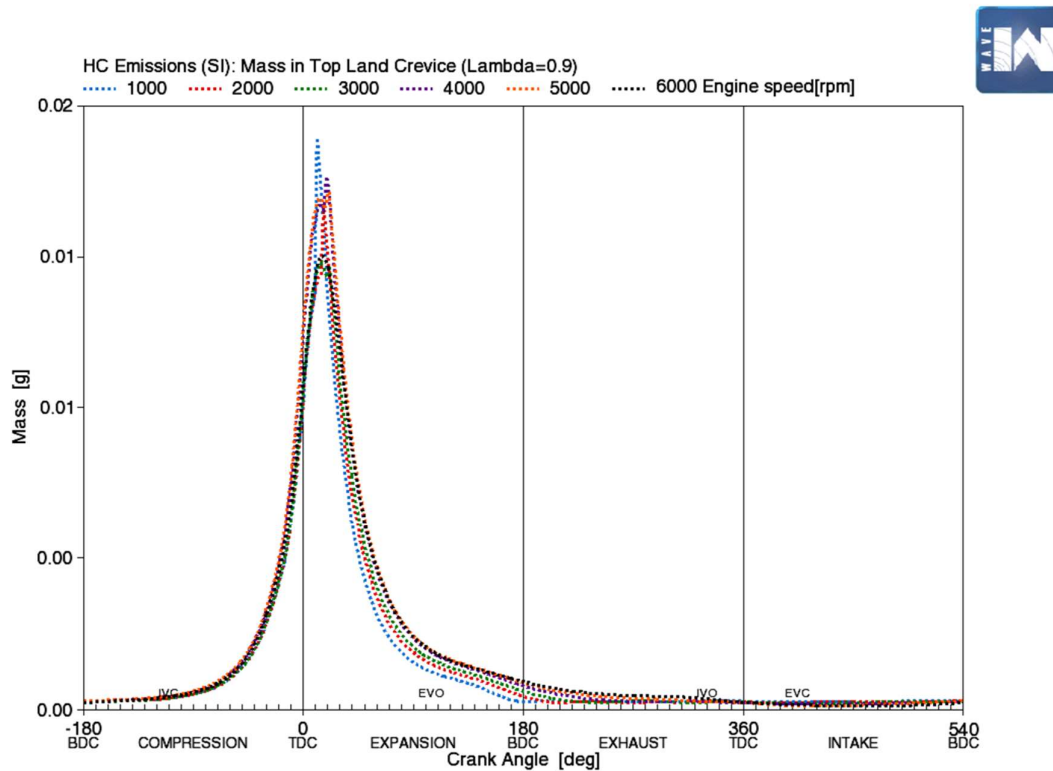
Graph 61 : Cumulative fuel mass absorbed by oil layer during combustion of rich air-fuel mixture($\lambda=0.9$)



Graph 62: Cumulative fuel mass absorbed by oil layer during combustion of lean air-fuel mixture($\lambda=1.1$)

Graph 62 discusses about the fuel mass absorbed by lubricant layer for a lean mixture combustion. The highest absorption is noted at 4000 RPM with .000125 grams of fuel, 15.54% fewer consumption when lambda changes from 0.9(rich) to 1.1(lean).

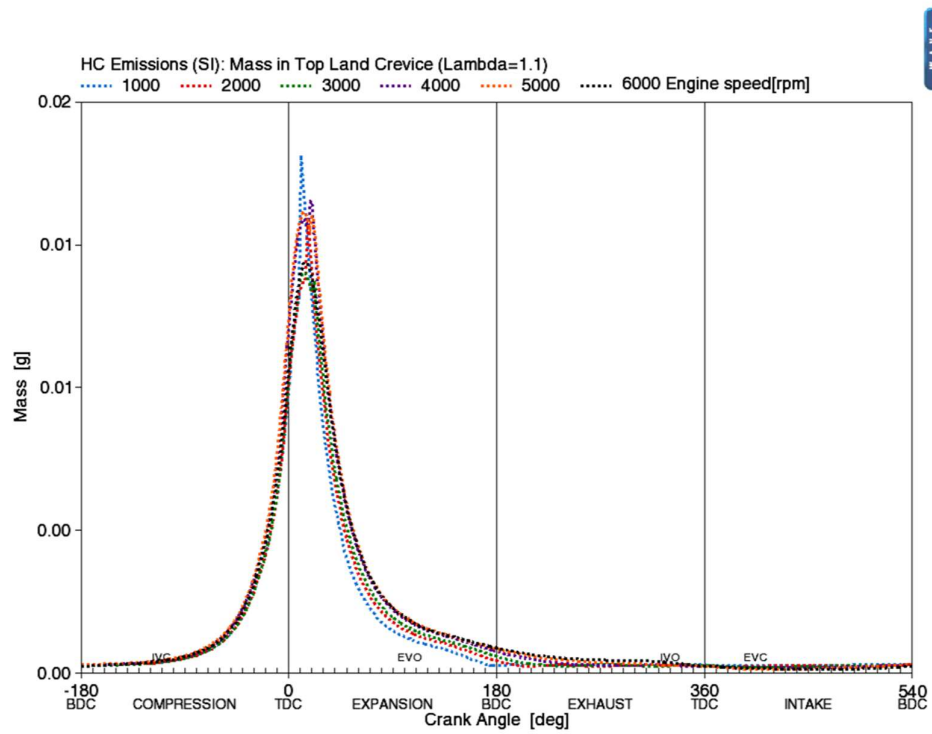
5.2.2 Mass fraction of unburned fuel mixtures in piston top land crevice



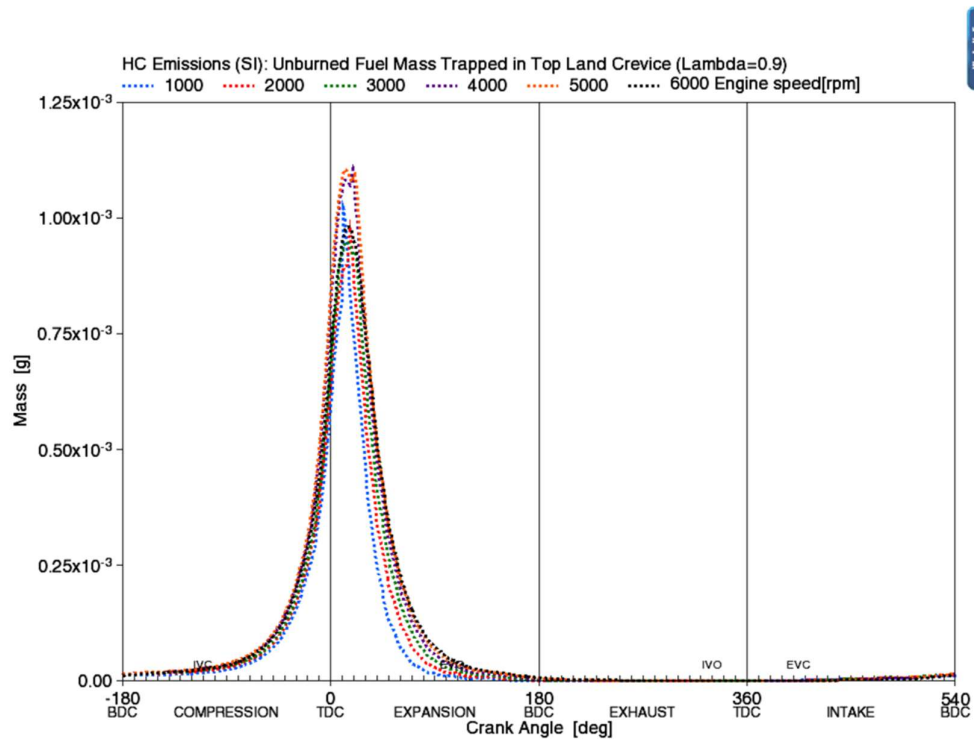
Graph 63 : Mass in top land crevice during combustion of rich air-fuel mixture($\lambda=0.9$)

Graph 63 and 64 discusses about the characteristics of the fuel mass accumulation in the top land crevice over a 4-stroke cycle of a Ford Sigma combustion model. The mass fraction of fuel increases gradually after intake stroke, when the piston retracts from BDC to TDC. The increase in the fuel accumulation in top land crevice volume increases when inlet valve closes during compression stroke. The highest composition of fuel trapped in crevice volume is noted at 10 to 16 degrees after TDC during expansion stroke and it varies according to the engine speed.

At 1000 RPM, .0189 gram of fuel is recorded as highest at a CA of 11.37 degrees. Least accumulation is noted at 6000 RPM with .0159 grams at CA 15.4 degrees. Meanwhile graph 64 concludes that .0178 grams of fuel is the highest accumulation at 1000 RPM, 5.82% less in fuel mass at a CA of 11.37 degrees when the air fuel mixture is lean.



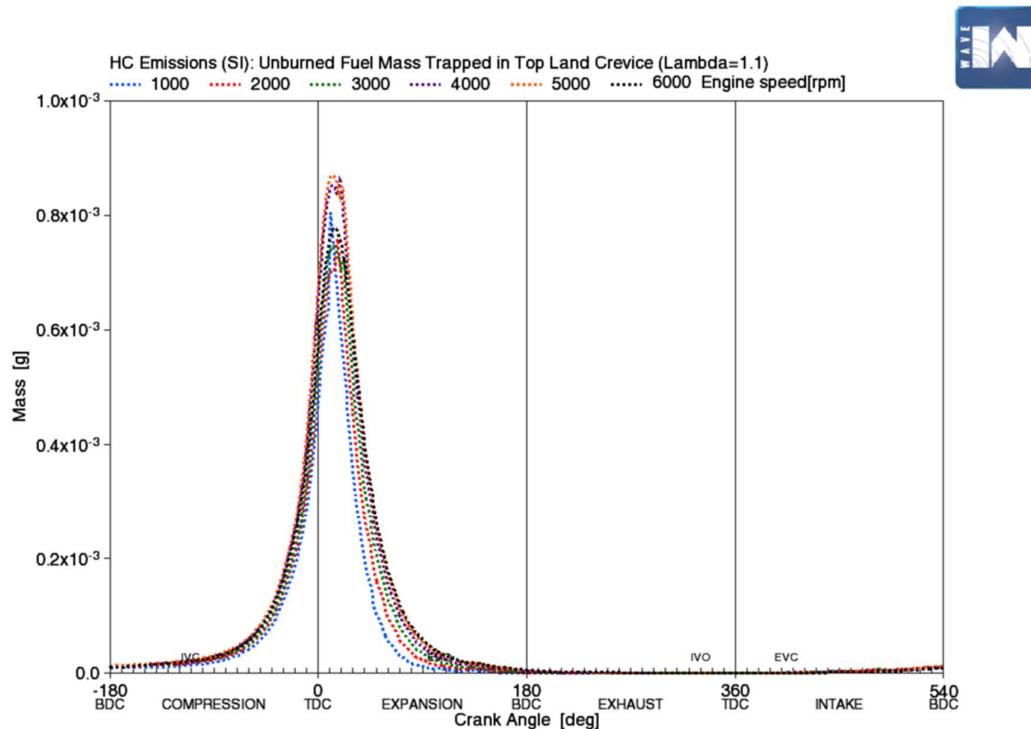
Graph 64: Mass in top land crevice during combustion of lean air-fuel mixture ($\lambda = 1.1$)



Graph 65: Unburned fuel mass in top land crevice during combustion of rich air-fuel mixture ($\lambda = 0.9$)

Graph 65 and 66 address about the unburned fuel composition in top land crevice volume for rich and lean fuel combustion on a 4-stroke cycle of a Ford Sigma combustion model. As discussed in previous section about mass fraction of fuel in top land crevice volume, During the piston's retraction from BDC to TDC, the unburned fuel composition increases gradually. When the inlet valve closes during compression stroke, fuel accumulates in top land crevices.

The highest composition of fuel trapped in crevice volume is noted at 10 to 16 degrees after TDC during expansion stroke and it varies according to the engine speed. For lambda 0.9, the mass of unburned fuel is recorded highest at 4000 RPM as .0011 grams at a CA of 19 degrees whereas for lambda 1.1, more unburned fuel mass is trapped at 5000 RPM,.00087g at CA 11 to 13 degrees. This concludes that for lean mixtures the unburned hydrocarbon emissions.



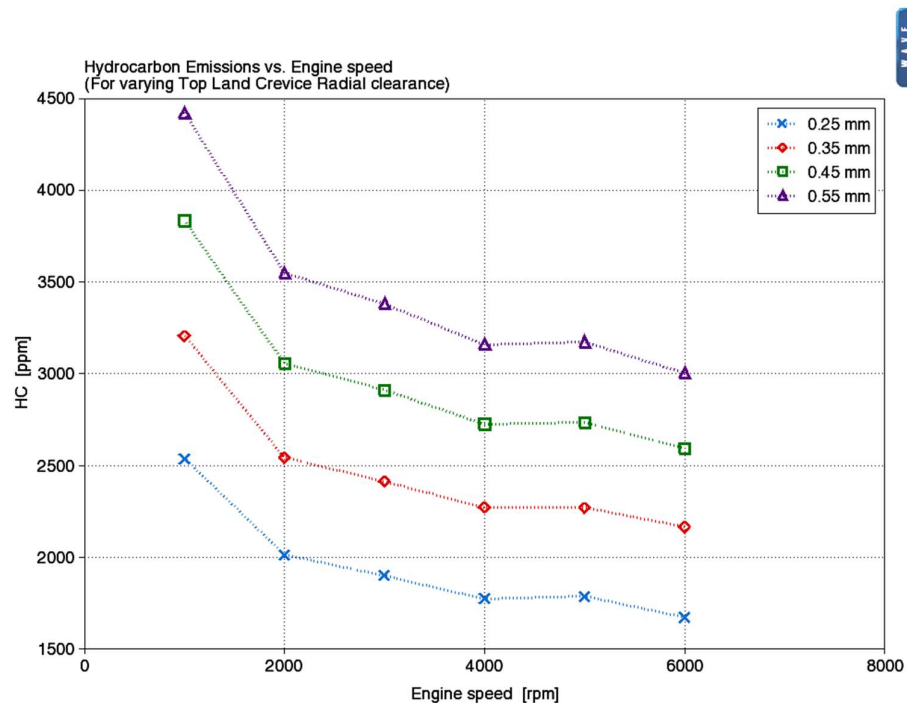
Graph 66: Unburned fuel mass trapped during in top land crevice during combustion of lean air-fuel mixture($\lambda=0.9$)

5.3 Investigation into hydrocarbon emissions due to top land crevice radial clearance

Even though the air fuel mixture has its influence on the hydrocarbon emissions in top land crevice, the discussions from section 5.2 concludes that the unburned fuel mass trapped inside the crevices are directly proportional to the volume or radial, clearance between the piston and the cylinder liner. This section discusses about the influence of radial thickness in hydrocarbon emissions. Radial thickness of a crevice is likely to increase in small proportions when it comes to longevity of a powertrain dependant on friction.

This study is considered on validation air fuel mixture ($\lambda=0.9$). The study is done for varying radial thickness 0.25,0.35,0.45,0.55 mm respectively. 0.35 mm radial clearance was assumed as the standard radial clearance during combustion modelling in the previous chapters.

5.3.1 Hydrocarbon emissions



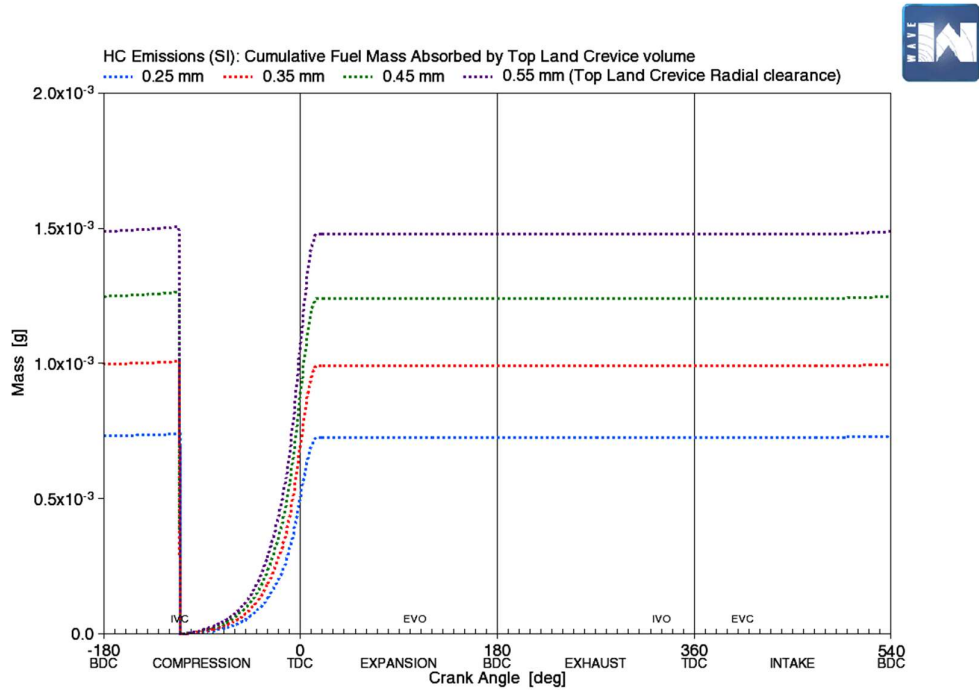
Graph 67: Hydrocarbon emissions on a Ford Sigma combustion model as a function of top land crevice radial clearance

Graph 67 discusses about the hydrocarbon emissions for a measurement of 0.25,0.35,0.45, and 0.55 mm radial clearance values. The graph depicts that hydrocarbon emissions are higher at lower RPM and decreases with respect to increase in engine speed. Meanwhile the key conclusion is that as radial clearance at top land crevice increases the hydrocarbon emissions increases due to more crevice volume.

At 1000 RPM, for a radial clearance of 0.35mm, 3207 ppm of HC emissions is recorded. An increase in 0.1 mm can result in 3833 ppm of HC emissions at 1000 RPM. This is an increase in 19.53% of hydrocarbon emissions with change in the radial clearance of 0.1mm. A 0.55 mm radial clearance is

recorded to have 4420 ppm at 1000 RPM. Compared to the data of 0.35 mm, an increase in 37.84% of HC emissions are a serious consideration that needs to be addressed with regards to controlling the emissions due to internal combustion engines.

5.3.2 Influence of radial thickness on top land crevice and oil layer hydrocarbon emissions

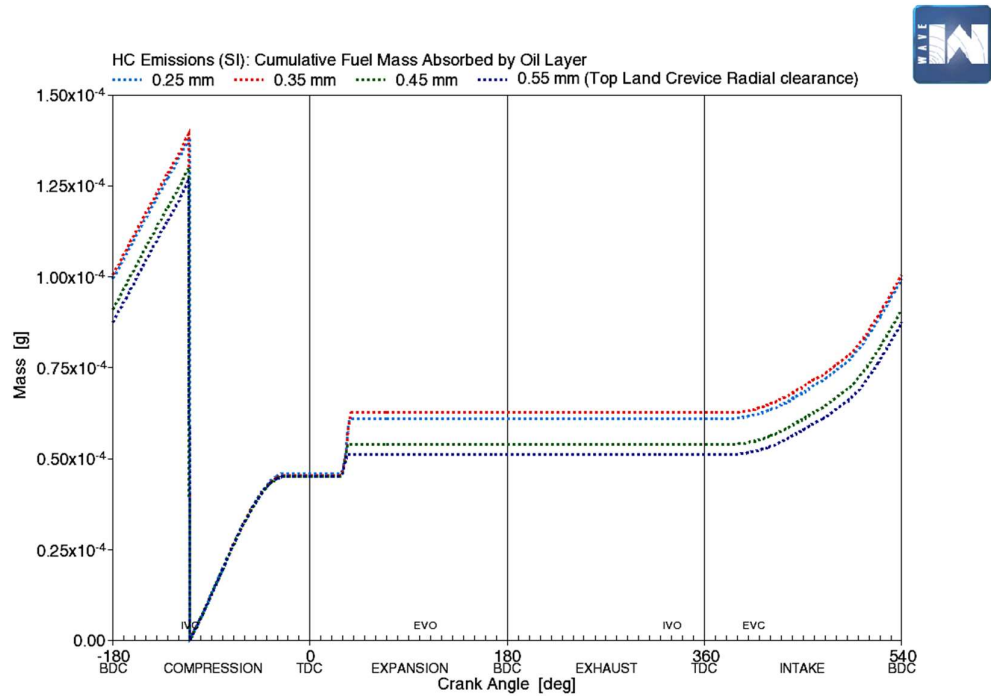


Graph 68 : Fuel mass absorbed by top land crevice volume due to varying radial thickness

In chapter 5.2.1 the data has discussed about the fuel mass absorption in both crevice volume and oil layers at varying engine speeds. Graph 68 discusses about the amount of fuel mass absorbed in top land crevice due to the varying radial thickness. As in the previous section, results conclude that HC emissions increases with radial thickness, higher fuel mass absorption into top land crevice volume is noted due to the increase in radial clearance from the beginning of expansion stroke to the IVC phenomena in compression stroke.

For a radial clearance of 0.35 mm, 0.001009 gram of fuel mass is absorbed into the crevice volume, whereas increase in 0.1 mm (0.45 mm net clearance) leads to 0.001262 grams of fuel mass absorption ,25.02 % increase in fuel mass composition to that of 0.35 mm radial clearance.

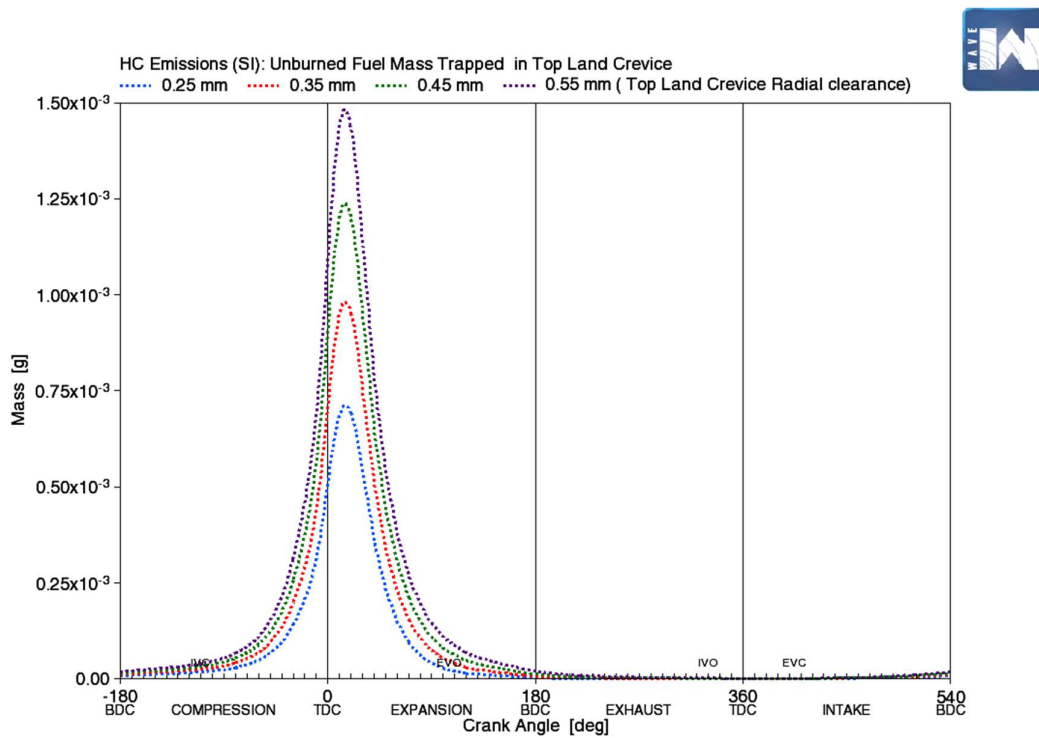
For a radial clearance of 0.55 mm, 0.20 mm increase compared to that of standard geometry,0.001499 grams of fuel have been absorbed between the top land piston and cylinder liners, 48.54% higher emissions in crevice volume compared to that of standard geometry top land crevice radial clearance.



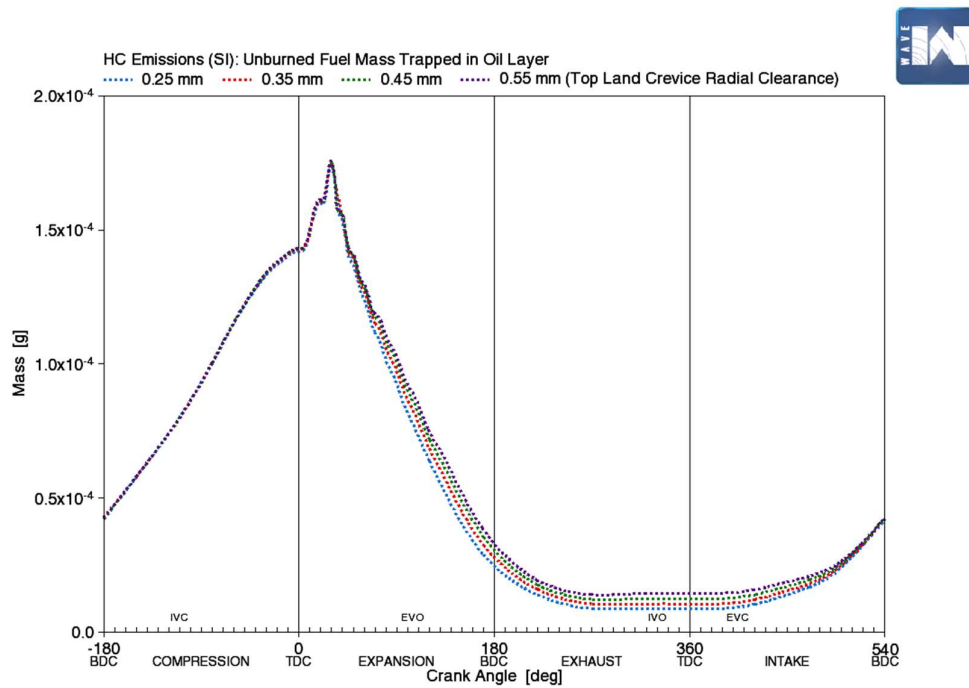
Graph 69 : Fuel mass absorbed by oil layer due to varying radial thickness

Graph 69 gives a detailed analysis into the influence of radial thickness on fuel mass absorbed by the lubricant layer within the cylinder line. The rate of fuel mass absorbed in the oil layer is a proportion of absorbed fuel in piston top land crevice volume. The absorption of fuel increases gradually from the IVC of compression stroke till beginning of expansion stroke. Once the fuel injection event is not happening in the cylinder between expansion and exhaust stroke, the absorbed fuel in oil layer has same concentrations of mass of fuel. This increases gradually in intake stroke after the EVC event, where the injected fuel traps in crevice volume. Highest rate of absorption in oil layer is recorded at a CA just before IVC event where increase in cylinder pressure during compression stroke leads to quick release of absorbed fuel into combustion zone and next cycle of fuel absorption starts at IVC.

From graph 69, it is noted that the fuel mass absorbed is frequently of higher concentration when the radial clearance is low, whereas the absorption rate decreases with increase in crevice volume. According to the data the highest absorbed fuel mass over a 720-degree cycle is .000139 grams for radial clearance of 0.35 mm, at a CA of -111 degrees before IVC event. The absorption rate is recorded second highest for a radial clearance of 0.25 mm with .000138 grams, followed by 0.45mm with .000130 and the least recorded fuel mass absorption for a highest radial clearance of 0.55 mm with .000125 grams.



Graph 70: Unburned fuel mass trapped in top land crevice due to varying radial clearance



Graph 71: Unburned fuel mass trapped in oil layer as function of varying radial clearance

Graph 70 and 71 discusses about the unburned fuel mass trapped in piston top land crevice volume and oil layer and how the characteristics are throughout the 4 strokes for a varying radial thickness. Graph 70 concludes that the unburned fuel mass trapped in crevice volume increases with radial clearance. The unburned fuel mass trapped inside the crevice increases during the compression stroke during the IVC event followed by highest accumulation at a CA of 14.65 degrees, in the beginning of expansion stroke as the piston combust the fuel and travels from TDC to BDC. The unburned hydrocarbon in top land crevice is recorded lowest for radial clearance of 0.25 mm with .000712 grams of fuel, 0.000981 ,0.001237 ,0.001482 grams for 0.35 mm,0.45mm and 0.55 mm radial clearance respectively.

Graph 71 discuss about the amount of unburned fuel mass trapped in oil layer. The quantity of unburned fuel mass stored is literally the same even throughout the end of intake, compression, and medieval expansion stroke. there is a rate of increase of unburned fuel mass trapped in oil layer with respect to radial clearance between CA 70 and 500 degrees. The higher rate is recorded at a CA of 29 degrees with a composition of 0.000171 grams.

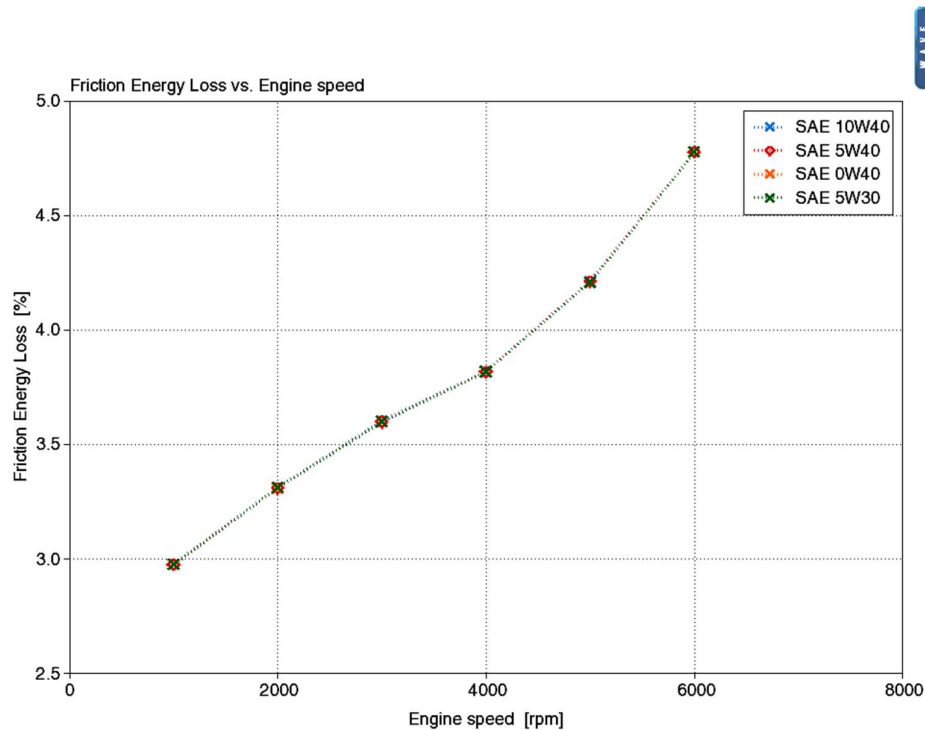
5.4 1-Dimensional combustion modelling analysis into frictional performance and lubricant properties

In order to analyse the possible influence of lubricant on performance characteristics like friction, the combustion model output of WAVE is to study the possible influence of lubricant specifications on frictional energy loss and the friction torque in the computational model of Ford Sigma.

The results are collected from simulation inputs where cylinder sub model lubricant parameters are input according to multigrade characteristics as shown in table 39. The research is done on SAE 10W-40, SAE 05W-40, SAE 0W-40 and 05-W30 grades.

Brand	ADDINOL			
Lubricant Specifications	Semi synthetic 1040	Super light 0540	Superior 040	Giga Light MV 0530 LL
Grade	10W-40	05W-40	0W-40	05W-30
Density	863	856	844	851
Dynamic Viscosity @100 °C (mm ² /s)	13.3	14.42	13.6	12.1
Dynamic Viscosity @100 °C (cP)	10.64	11.54	10.88	9.8
TBN (mg KOH / g)	11	11.25	10.3	6.5
Flash Point (°C)	235	235	230	230

Table 39 : Lubricant specifications and typical parameters used for combustion modelling (ADDINOL Lube Oil GmbH - High-Performance Lubricants, 2020) (ADDINOL Lube Oil GmbH - High-Performance Lubricants, 2018) (Anon., 2016) (ADDINOL Lube Oil GmbH - High-Performance Lubricants, 2020)



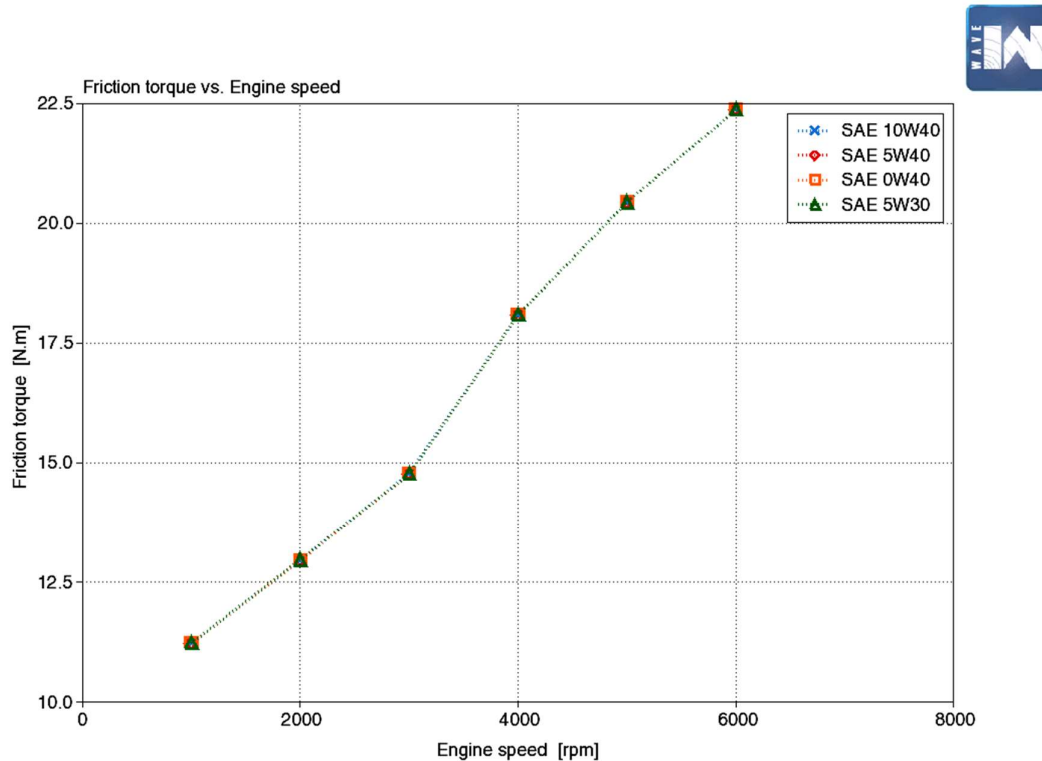
Graph 72 : Frictional energy loss characteristics of multigrade lubricants

Frictional Energy Loss vs Engine speed	SAE 10W-40	SAE 5W-40	SAE 0W-40	SAE 5W-30
RPM	%	%	%	%
1000	2.977580	2.977584	2.977531	2.978156
2000	3.311462	3.311023	3.309689	3.309706
3000	3.600920	3.601383	3.600959	3.603876
4000	3.819629	3.819638	3.819345	3.818437
5000	4.210627	4.210752	4.208052	4.212529
6000	4.777565	4.777552	4.777568	4.777568

Table 40 : Frictional energy loss WAVE data for wide range of multigrade lubricants

Graph 72,73 and table 40,41 discusses about the computational output of variable lubricant parameters. Graph 72 and 73 concludes that there is no relative effect on FEL and friction torque. According to the data collected from table 40 and 41, a fractional influence is noted for the varying properties of lubricant like density and dynamic viscosity.

Table 40 concludes that at 1000 RPM, SAE 5W-30 have more FEL with a negligible difference of .000625% than SAE 0W40 with least of four multigrade lubricants, at 2000 RPM SAE 10W-40 have the highest FEL, and at 6000 RPM, SAE 0W-40 and SAE 5W-30 have same FEL.



Graph 73: Frictional energy loss characteristics of multigrade lubricants

Frictional Torque vs Engine speed	SAE 10W-40	SAE 5W-40	SAE 0W-40	SAE 5W-30
RPM	Nm	Nm	Nm	Nm
1000	11.2310	11.2290	11.2295	11.2317
2000	12.9508	12.9576	12.9516	12.9516
3000	14.7864	14.7755	14.7745	14.7864
4000	18.0906	18.0952	18.0946	18.0905
5000	20.4336	20.4375	20.4332	20.4337
6000	22.3696	22.3695	22.3694	22.3696

Table 41 : Frictional Torque WAVE data for wide range of multigrade lubricants

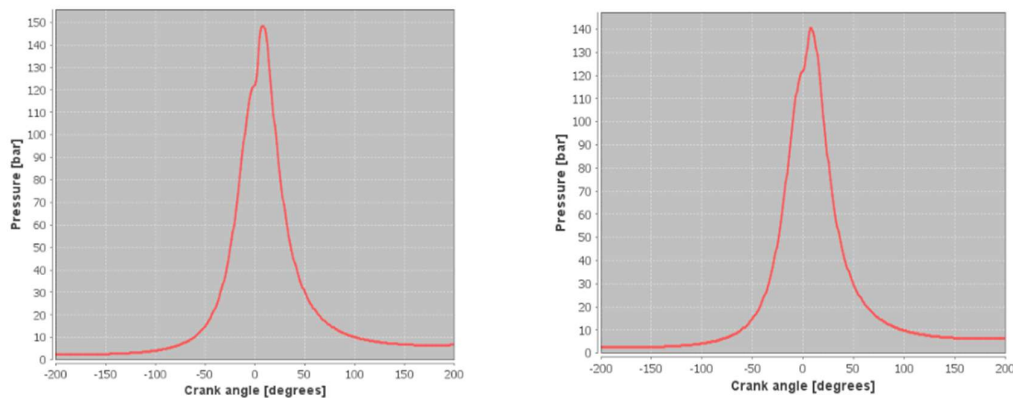
Table 41 outlines the simulation output of variation in the powertrain friction torque at varying engine speed. Friction torque varies and have insignificant impact according to the change in lubricant grade whereas the results are dependent on the kinematic viscosity, density, and molecular weight of the lubricant. This concludes that WAVE have evaluation on lubricant parameters, whereas the results concludes that variation in lubricant properties have no significant impact on friction and performance of a powertrain.

5.5 3-Dimensional combustion modelling analysis into AFR, EGR and its influence on thermodynamic properties and emissions in Ford Lynx diesel powertrain.

5.5.1 Simulation analysis of influence of AFR on Ford Lynx engine performance and emissions

This chapter is focussed on computational modelling analysis on combustion thermodynamics and emissions on a diesel engine. The research is done on stoichiometric, lean, and rich mixtures to study the effects of dodecane in a 3-dimensional computational fluid dynamics software.

5.5.1.1 Combustion thermodynamic analysis

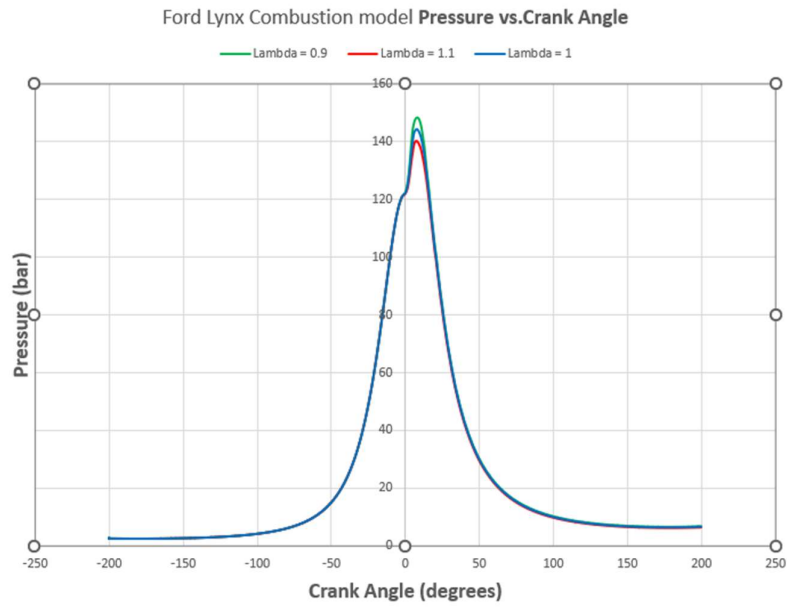


Graph 74 :Ford Lynx combustion analysis cylinder pressure (A) $\lambda=0.9$ (B) $\lambda=1.1$

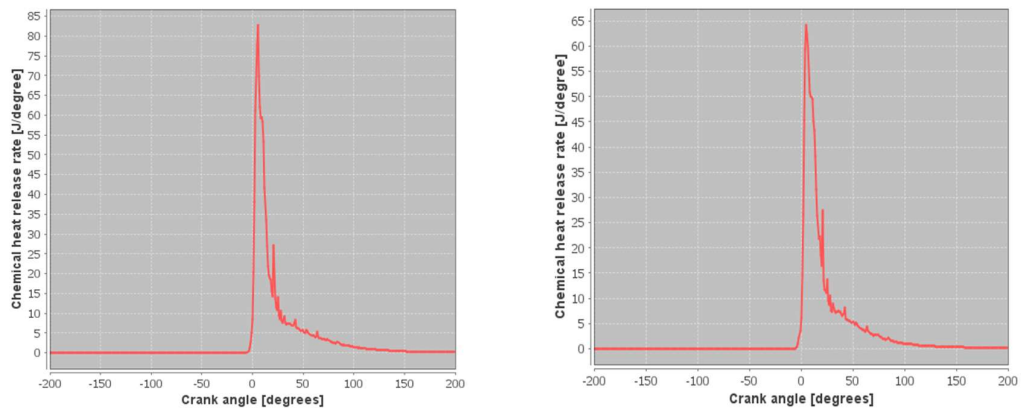
Graph 74 outlines the cylinder pressure for $\lambda=0.9$ and $\lambda=1.1$ for a combustion simulation between CA -200 and 200. The Injection event starts at -8 CA as in validated simulation setup for a total duration of 25 degrees. The combustion of dodecane begins when the piston reaches -2 before TDC where the chemical heat release begins. The inclination in pressure in combustion zone at CA 1 degrees leads to a maximum pressure of 148.32 bar for $\lambda=0.9$, 140.31 bar for $\lambda=1.1$, when the piston is 8 degrees after TDC followed by expansion stroke.

The validation of the Ford Lynx geometry was done when $\lambda=1$. Graph 75 discusses about the combined combustion model output for 3 different mixtures ($\lambda=0.9, 1, 1.1$). Cylinder pressure have strong influence in the amount of injected fuel mass which outlines that 10% rich mixture have 8.01 more-cylinder pressure than 10 % lean mixture. As in previous chapter where influence of RON on varying fuel mixtures are studied, it is found that different octane rated gasoline and dodecane have great influence in cylinder performance output as the indicated power is directly proportional to the cylinder mean effective pressure, higher for rich air fuel mixture.

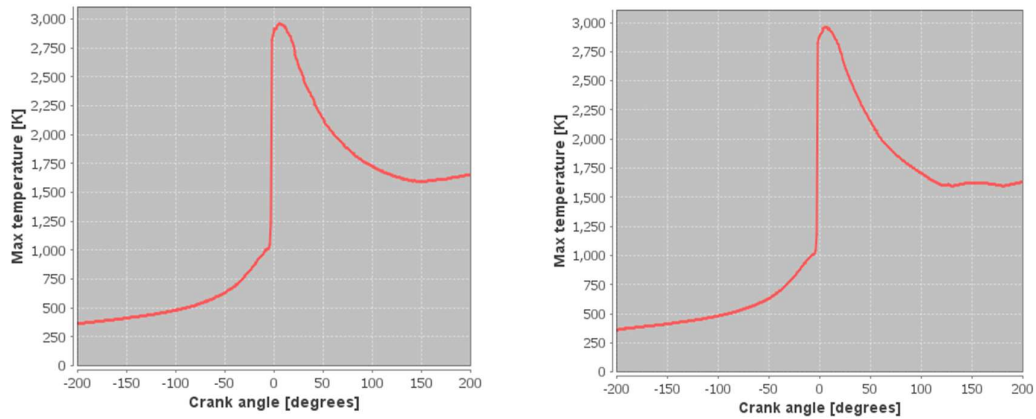
Graph 76 discusses about the chemical heat release in the combustion zone for $\lambda=0.9$ and $\lambda=1.1$. For a 10 % rich mixture, a maximum of 82.59 joules of energy is released per degree whereas for a 10 % lean mixture a maximum of 64.23 joules is released when piston is 5 degrees after TDC. More heat release is noted at a CA of 21 degrees, which can lead to disturbed combustion pressure phenomena within the combustion chamber. It is noted that the simulation has similar characteristics in cylinder pressure, heat release during compression and expansion strokes in wide range of fuel mixtures as the combustion is directly related with injected fuel mass.



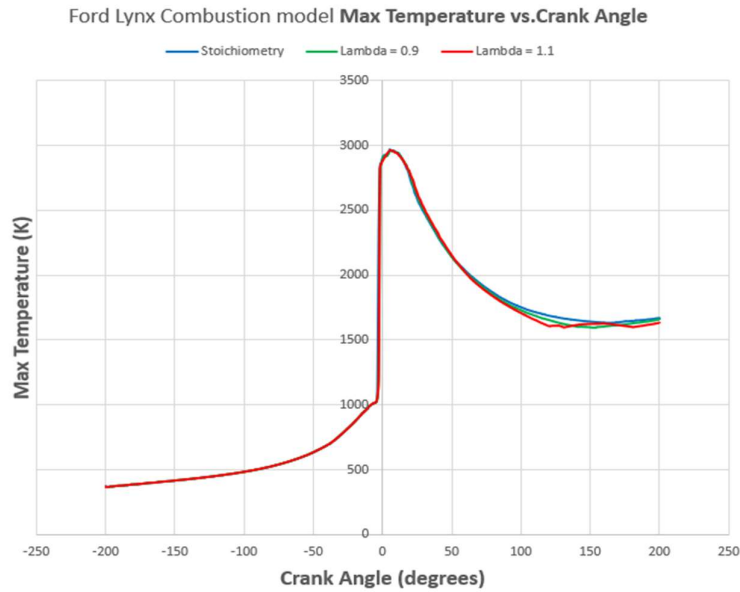
Graph 75 : Merged pressure data for different AFR of n-dodecane(diesel) combustion



Graph 76 : Influence of chemical heat release rate of fuel mixtures during combustion (A) $\lambda=0.9$ (B) $\lambda=1.1$



Graph 77 : Maximum combustion zone temperature analysis (A) $\lambda=0.9$ (B) $\lambda=1.1$

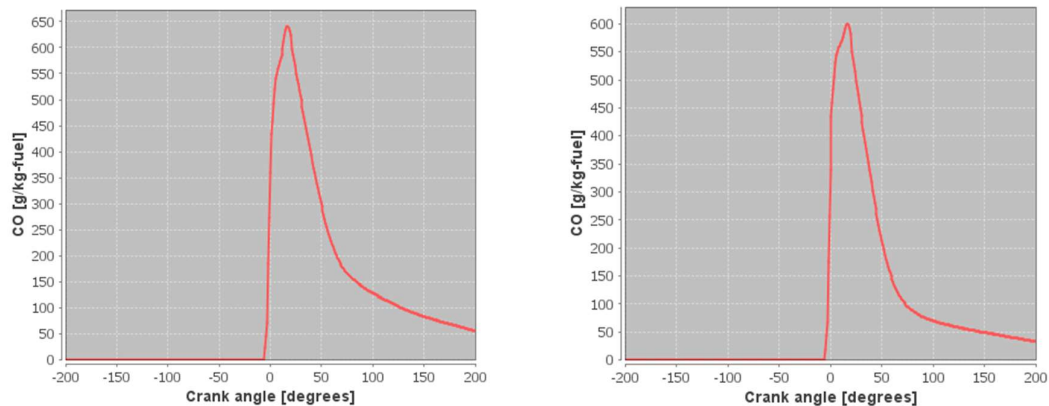


Graph 78 : Merged Temperature data for different AFR

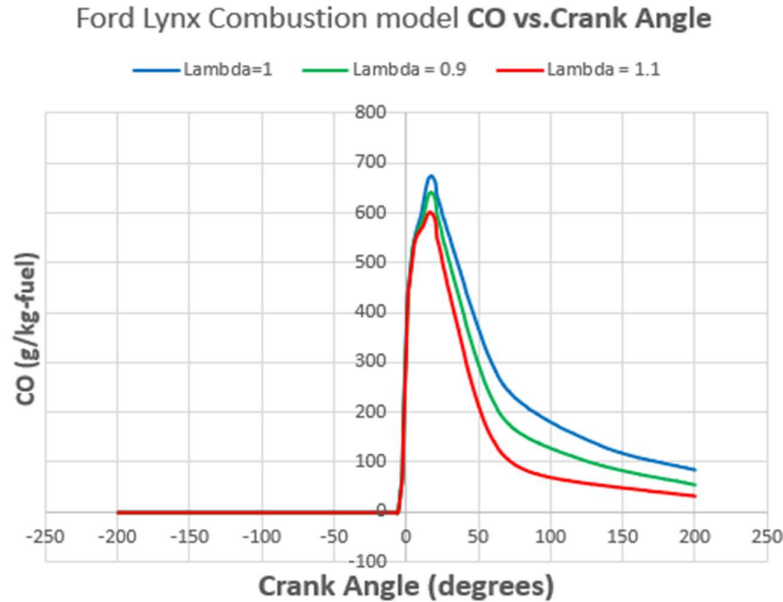
Graph 77 and 78 discusses about the maximum cylinder combustion zone temperature during the combustion of diesel for $\lambda=0.9$, 1 and 1.1. For $\lambda=0.9$, cylinder pressure increases from 1233 K to 2816 K at -1.99 CA with a maximum temperature of 2959 K recorded at 6 degrees after TDC whereas for $\lambda=1.1$, cylinder pressure increases from 1185 K to 2820 K at -1.99 CA with a maximum temperature of 2962 K recorded at 6 degrees after TDC. It is noted at -1.99 CA, the temperature rise in lean combustion is recorded higher than that of rich mixture. Meanwhile For $\lambda=1$, cylinder temperature during iteration increases from 1060 K to 2340 K at -2.96 CA, followed by an increase from 2340 K to 2846 K at -1.99 with a maximum temperature of 2969 K recorded at 5 degrees after TDC. As the temperature drops throughout the expansion stroke, lean mixtures have fluctuation in cylinder temperature between 120 and 180 degrees which exceed over $\lambda=0.9$ and 1.1.

5.5.1.2 Combustion emission analysis

Graph 79 and 80 outlines the carbon monoxide emission analysis from a Ford Sigma combustion model. The research is done on $\lambda=0.9$, 1 and 1.1 respectively to study the influence of AFR during the combustion dodecane($C_{12}H_{26}$).

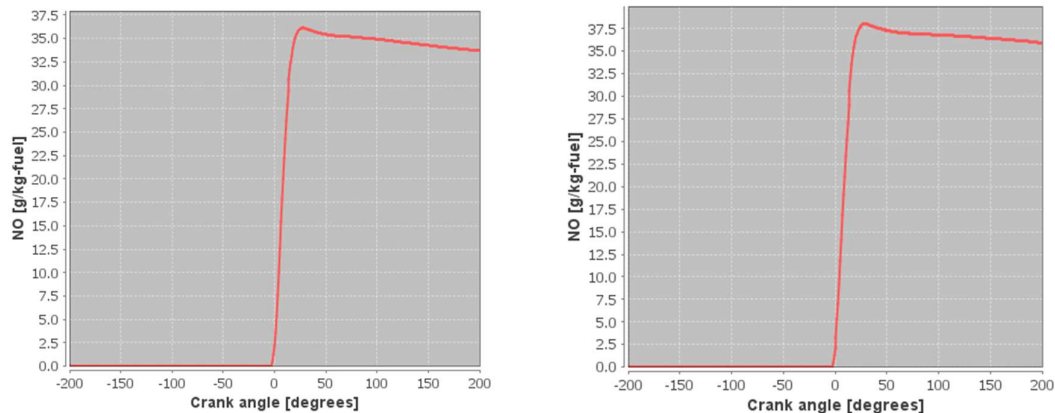


Graph 79 : Carbon monoxide emissions for varying fuel mixtures (A) $\lambda=0.9$ (B) $\lambda=1.1$



Graph 80: Merged CO data for varying fuel mixtures

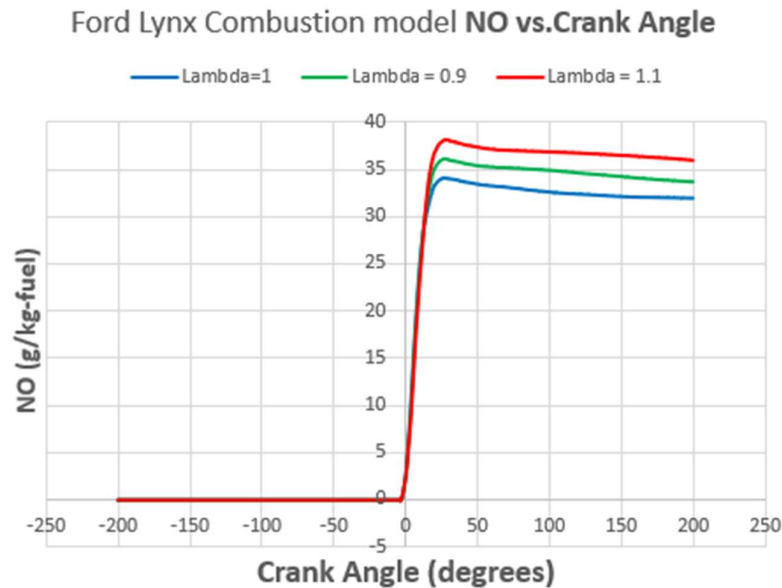
For $\lambda=0.9$, CO is recorded to have an amount of 640 grams per kilogram of fuel burned at 17 degrees after TDC. For $\lambda=1$, CO is recorded to have the highest contribution amount of 673 grams which decreases throughout the expansion stroke. For $\lambda=1.1$, CO is recorded to have an amount of 600 grams at 16 degrees after TDC. The simulation data concludes that CO is higher for $\lambda=1$, whereas rich mixture ($\lambda=1.1$) has least CO emissions throughout the combustion and expansion stroke over $\lambda=0.9$.



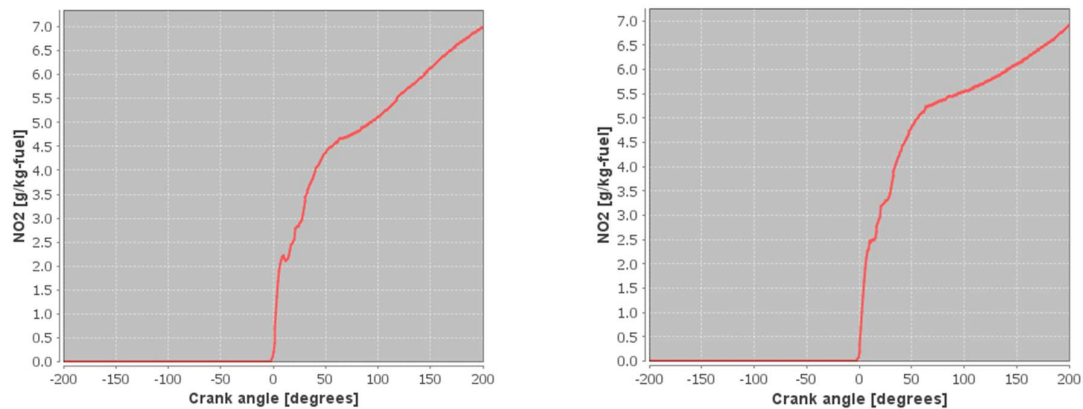
Graph 81 : Nitrogen monoxide emissions for varying fuel mixtures (A) $\lambda=0.9$ (B) $\lambda=1.1$

Graph 81 and 82 discusses about the influence of AFR on nitrogen monoxide emissions .In chapter 5.2.1.1 , the maximum temperature of different fuel mixtures are recorded to have a maximum temperature of over 2800 K , which results in production of oxides of nitrogen.For $\lambda=0.9$,36.16 gram of NO is recorded at a CA of 27 degrees. Graph 80 concludes that lean mixture has the highest NO production over rich mixture and stoichiometric mixture during combustion of diesel. For $\lambda=1.1$,38.08 grams of NO are released and recorded as highest at a CA of 29 degrees after TDC. $\lambda=1$

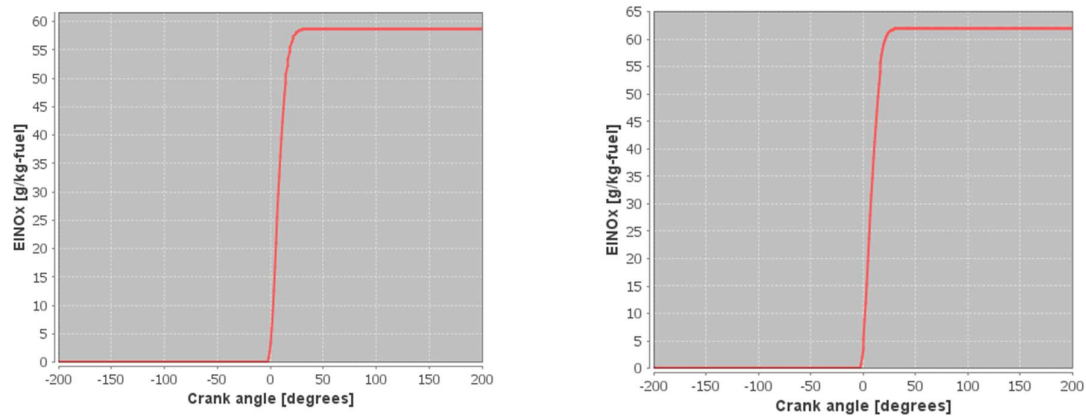
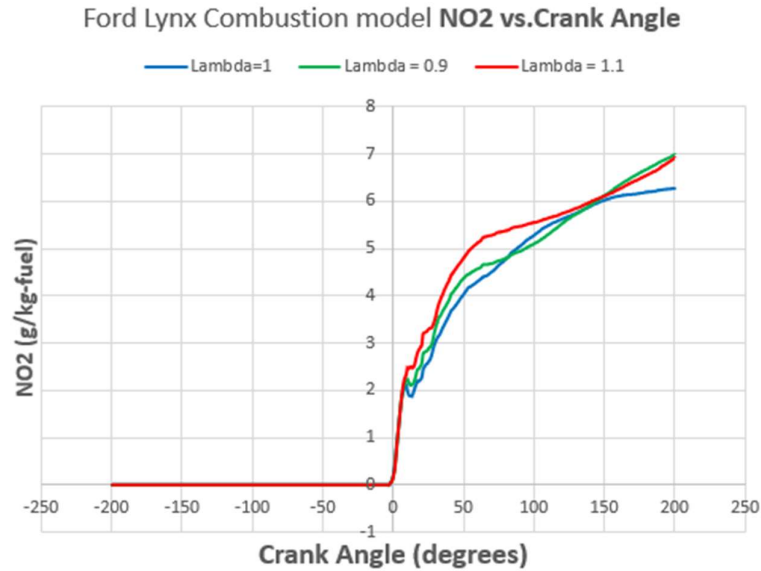
have least contribution towards NO emissions with a maximum of 34.05 grams produced at CA 27 degrees.



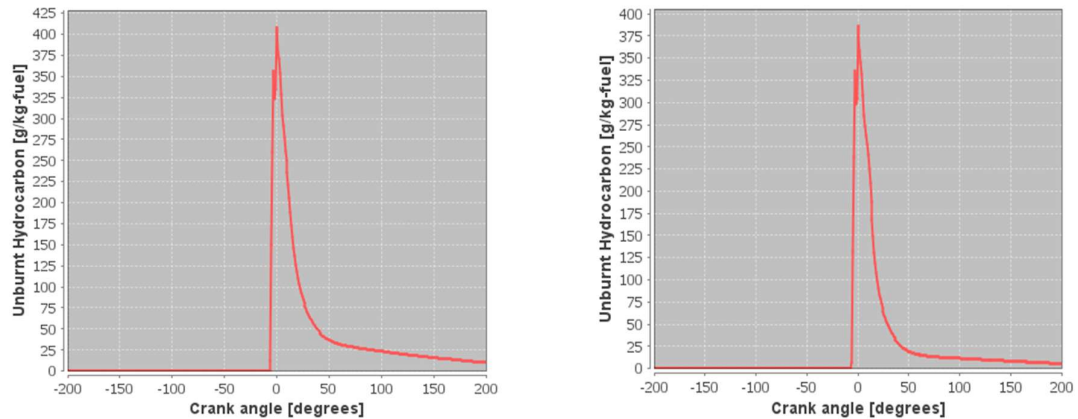
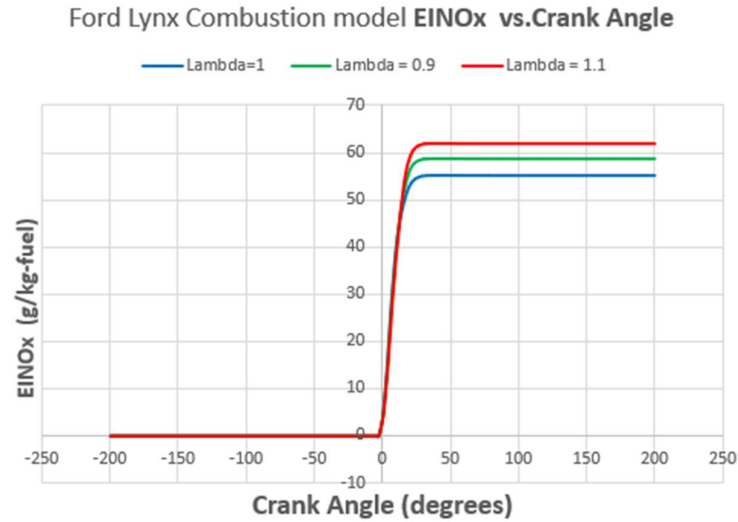
Graph 83 and 84 discusses about the nitrogen dioxide emissions due to combustion of n-dodecane for varying fuel mixtures. The NO₂ production is recorded to increase throughout the expansion stroke where the results conclude that rich fuel mixture have NO₂ emissions with negligible increase over lean mixture at CA of 200 degrees. For a lean mixture, NO₂ production is highest between CA 30 and 100 degrees. For $\lambda=0.9$, 7 grams of NO₂ is recorded to release over 6.27 grams for $\lambda=1$, and 6.92 grams for $\lambda=1.1$.



For $\lambda=0.9$, 7 grams of NO₂ is recorded to release over 6.27 grams for $\lambda=1$, and 6.92 grams for $\lambda=1.1$.

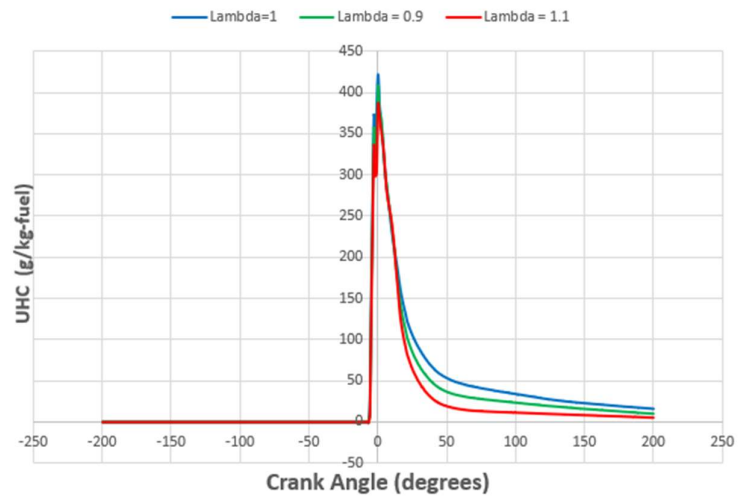


Graph 85 and 86 discusses about the influence of AFR on NOx emission index during the combustion of diesel. For $\lambda=0.9$, NOx is recorded to have an amount of 58.72 grams per kilogram of fuel burned at 42 degrees after TDC. For $\lambda=1$, NOx is recorded to have the least contribution amount of 55.25 grams per kilogram fuel burned. For $\lambda=1.1$, NOx is recorded to have highest contribution, an amount of 62.01 grams at 42 degrees after TDC. The simulation data concludes that NOx is higher for $\lambda=1.1$, whereas stoichiometric mixture ($\lambda=1$) has least NOx emissions throughout the combustion and expansion stroke over $\lambda=1.1$ which is later released to exhaust.



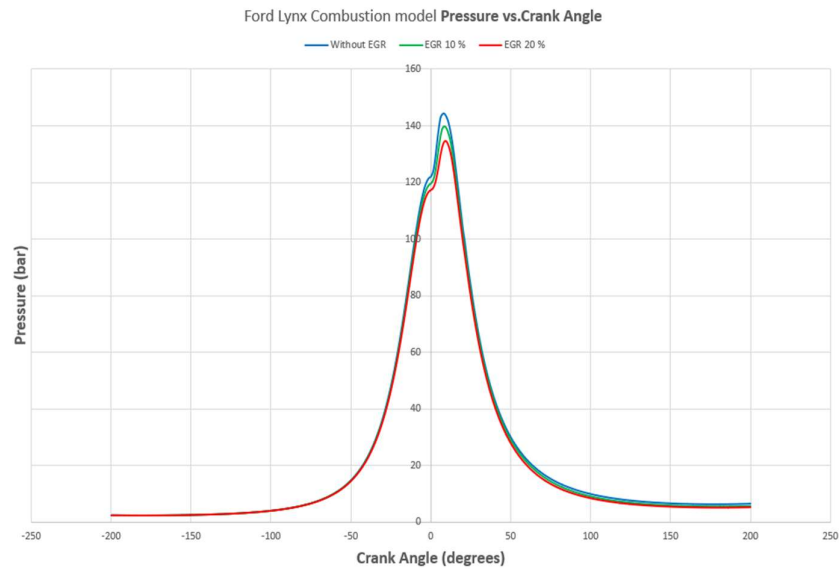
Graph 87 and 88 discusses about the influence of AFR on unburned hydrocarbon emission index during the combustion of diesel. AT CA 0 degrees, the fuel is about to combust where, for $\lambda=0.9$, UHC is recorded to have an amount of 408 grams unburned at 0 degrees, the injected fuel mass. For $\lambda=1$, UHC is recorded to have 421 grams at 0 CA. For $\lambda=1.1$, UHC is recorded with an injected diesel fuel of 386 grams at 0 degrees. The simulation data at 200 degrees on beginning of exhaust stroke concludes that unburned fuel is least for $\lambda=1.1$ with 5.33 grams, higher for $\lambda=1$ with 16.04 grams and $\lambda=0.9$ with 10.09 grams. Stoichiometric fuel mixture is recorded to have the highest contribution towards unburned hydrocarbon emissions and carbon monoxide emissions which is discussed previously in same chapter.

Ford Lynx Combustion model **Unburned HC vs.Crank Angle**



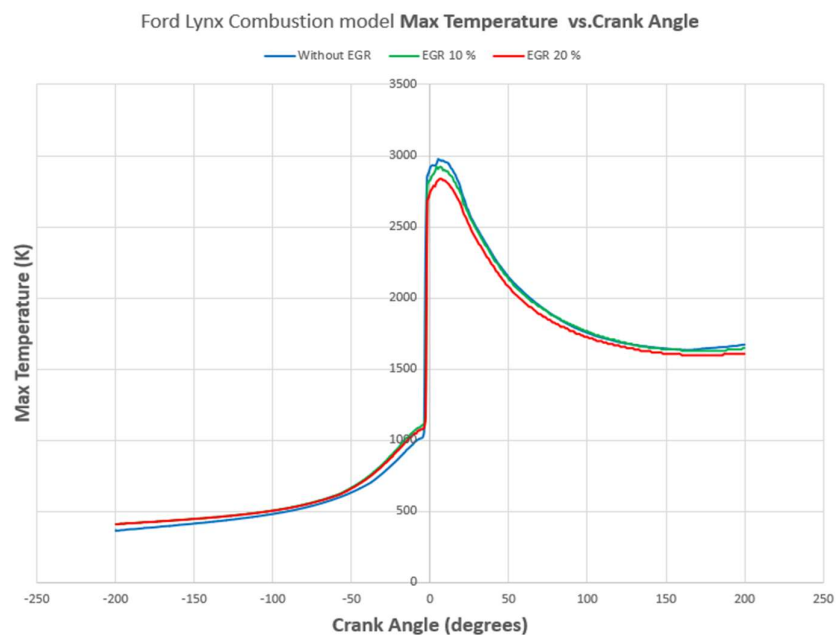
Graph 88 : Merged Hydrocarbon emissions data for varying fuel mixtures

5.5.2 Simulation analysis of influence of EGR on Ford Lynx engine performance



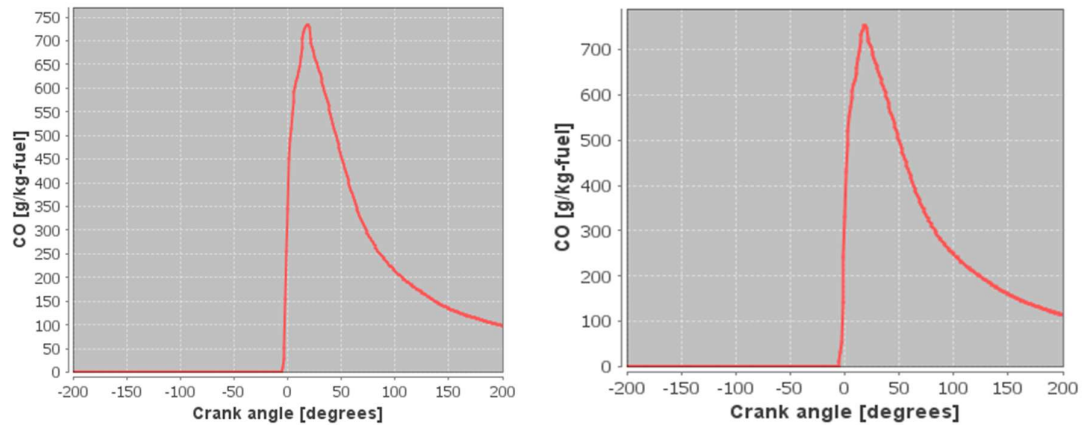
Graph 89 : Influence of EGR on combustion pressure

Graph 89 concludes that EGR reduces the pressure at the end of compression stroke , followed by complete combustion of dodecane. For EGR=10%, a maximum pressure of 139.69 bar is recorded at 8 degrees after TDC, whereas for EGR=20%, a maximum pressure of 134.79 bar is recorded at 9 degrees after TDC. Without EGR stoichiometric diesel combustion generated a maximum cylinder pressure of 144.31 bar at CA 8 degrees. This concludes that as the percentage of EGR increases the combustion phenomena turns late with decrease in cylinder pressure.

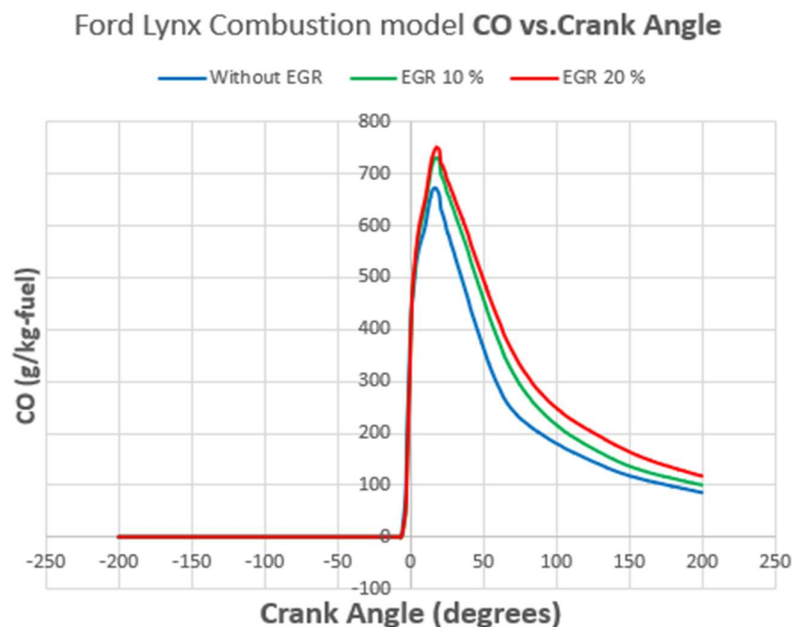


Graph 90 : Influence of EGR on combustion zone temperature

Graph 90 discusses about the influence of EGR on combustion zone maximum temperature. EGR decreases the overall cylinder temperature, considering when added to intake system with an initial intake temperature of 140 degrees (413 K). For EGR=10%, a maximum temperature of 2920 K is recorded at 4 degrees after TDC, whereas for EGR=20%, a maximum temperature of 2840 degrees is recorded at 6 degrees after TDC. Without EGR stoichiometric diesel combustion generated a maximum temperature of 2969 K bar at CA 5 degrees. This concludes that EGR have strong influence in reducing combustion zone temperature and have control in nitrogen oxides emission index, which are discussed in following section.

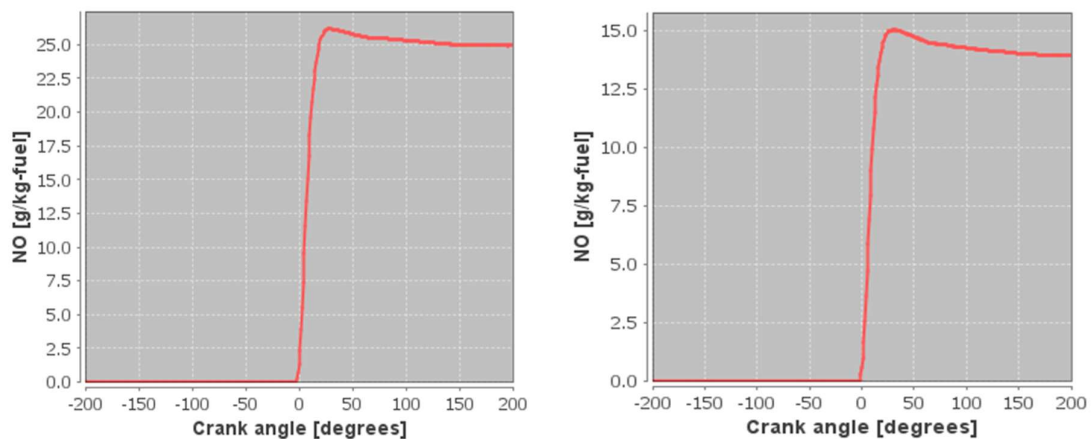


Graph 91 : Role of EGR on carbon monoxide emissions (A) EGR = 10 % (B) EGR = 20 %

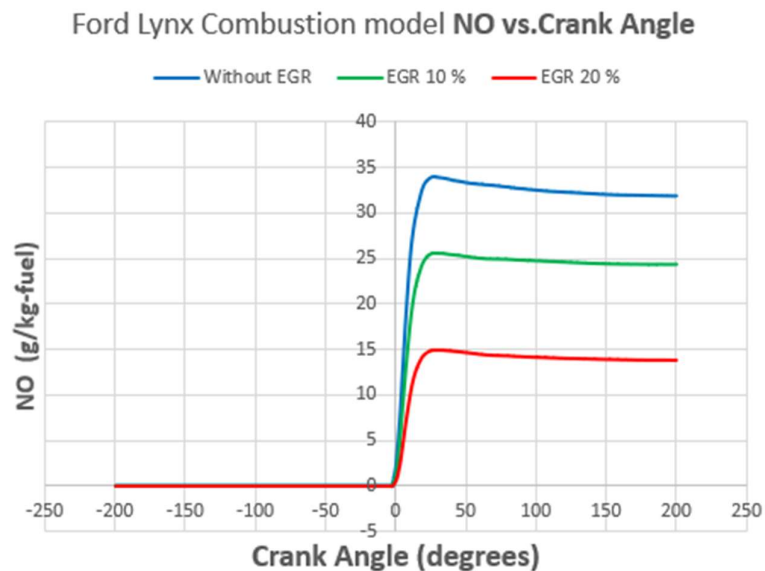


Graph 92 : Influence of EGR on CO emissions

Graph 91 and 92 discusses about the influence of EGR on CO emissions. The data concludes that addition of EGR results in increased CO emissions due to the presence of CO₂ mixture in combustion zone. For EGR=10%, CO is recorded to have 731 grams per kilogram fuel burned, whereas for EGR=20%, a maximum CO of 750 grams is recorded at 18 degrees after TDC. Without EGR stoichiometric diesel combustion generated CO of 673 grams at CA 8 degrees. Concentration of CO emissions decreases throughout the expansion stroke.

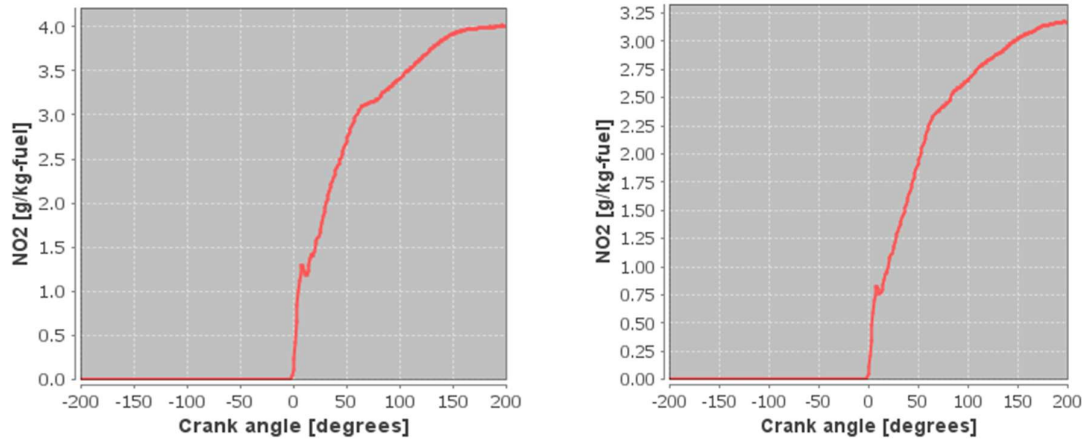


Graph 93 : Influence of EGR on Nitrogen monoxide emissions (A) EGR = 10 % (B) EGR = 20 %

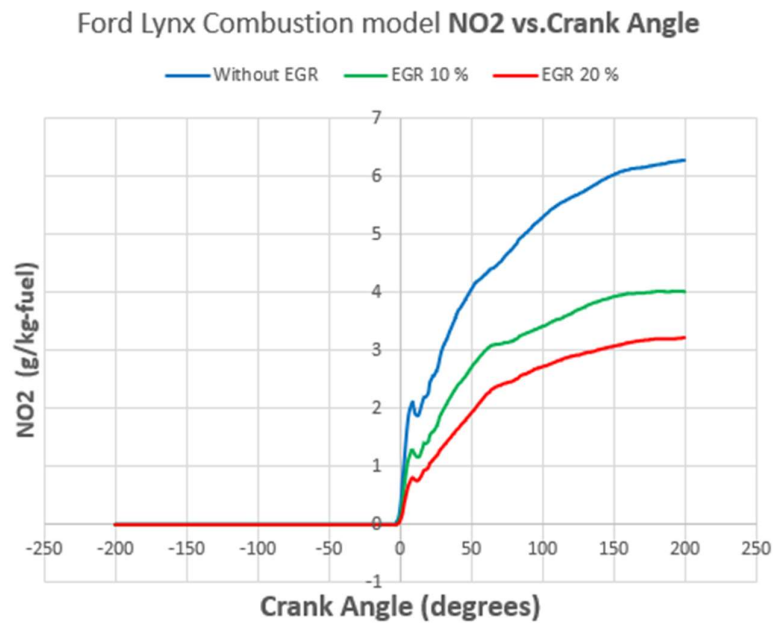


Graph 94 : Influence of EGR on NO emissions

Graph 93 and 94 discusses about the influence of EGR on nitrogen monoxide emissions. The data concludes that for stoichiometric combustion a maximum of 33.99 grams nitrogen monoxide is released during combustion whereas 10% EGR reduces the NO emissions to 25.55 grams and 20 % EGR is recorded to have 14.91 grams in a Ford Lynx engine emission.

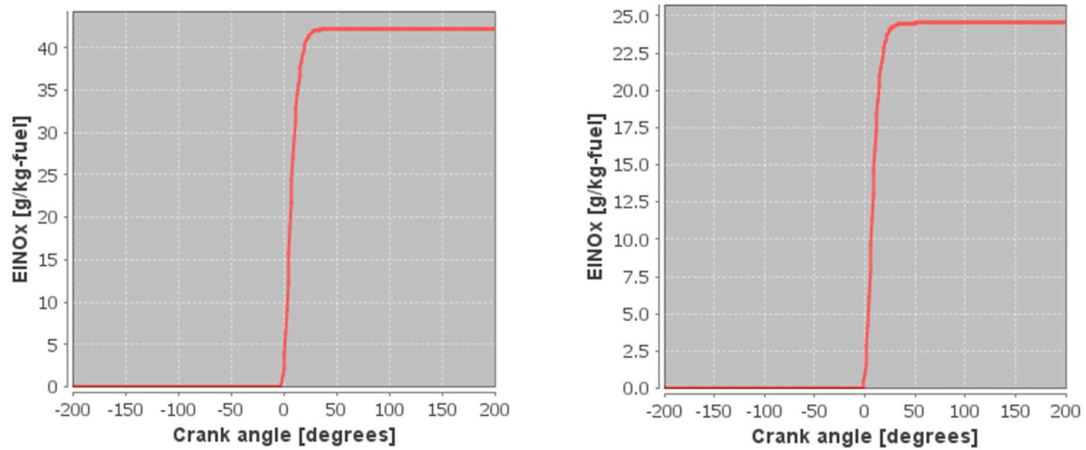


Graph 95 : Influence of EGR on Nitrogen dioxide emissions (A) EGR = 10 % (B) EGR = 20 %

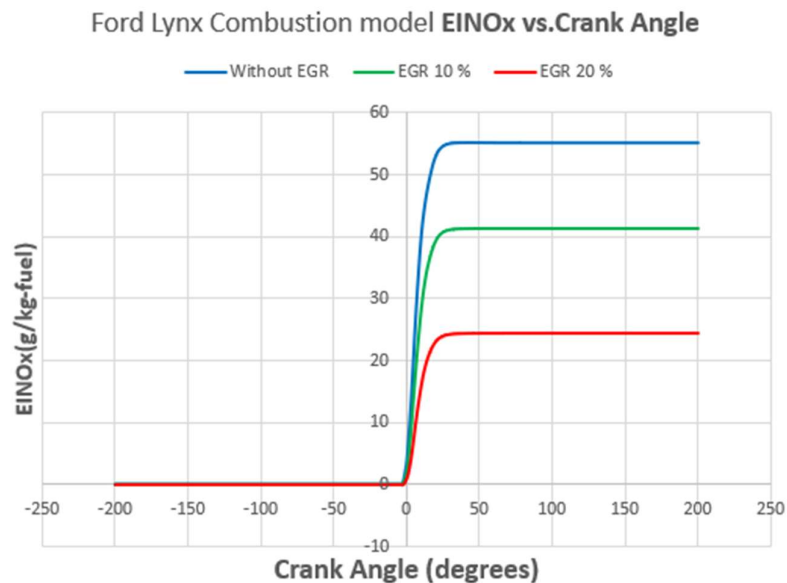


Graph 96 : Influence of EGR on NO2 emissions

Graph 95 and 96 discuss about the nitrogen dioxide emissions developed during combustion of diesel. For the validation model at CA 200, NO2 have the highest value of 6.27 grams. 10% EGR reduces the emissions into 4 grams and 20% EGR reduces the emissions to 3.21 grams. It has been noted from the model results that EGR have a strong influence in nitrogen-based emissions due to decreases in cylinder temperature during combustion. From NO and NO2 perspective, the emission rate has decreased in substantial quantity due to addition of EGR.



Graph 97 : Influence of EGR on Nitrogen monoxide emissions (A) EGR = 10 % (B) EGR = 20 %



Graph 98 : Influence of EGR on NOx emission index

Graph 97 and 98 discusses about the influence of EGR on NOx emission index. For the validated Ford Lynx model, a maximum emission index of 55.21 g/kg-fuel has been released to tailpipe during combustion. By adding 10 % EGR, the data concludes that the emission index has reduced to 41.3 g/kg-fuel, 25.19 % reduction to emissions without EGR. For 20 % EGR the emission rate has decreased to 24.38 grams, 55.84 % of net emissions without EGR.

6.Evaluation

6.1 Combustion modelling and analysis

The study investigated gasoline and diesel engine combustion performance and emissions in a 1D/3D computational simulation environment. A wide range of simulations are performed initially to validate both the manufacturer's combustion output and simulation output in the Ricardo WAVE and ANSYS Forte R2 combustion models. This is followed by research into RON, molecular diffusion, and EGR in a wide range of flammability.

Research A-The first stage of research (5.1) of Ricardo WAVE 1-Dimensional combustion modelling analysis was done to study the effects of Research Octane Number (RON) and flammability (AFR) of gasoline mixtures on a Ford Sigma gasoline powertrain. RON 100 and RON 91 exhibit similar properties as high-performance fuels of 5 different fuels studied in chapter 5.1. For 10 % rich mixture ($\lambda=0.9$) and 10 % lean mixture ($\lambda=1.1$), despite the fact that RON 100 has 4.55 % less lower heating value than RON 91 (5.1.2), RON 100 have highest indicated power, brake power, IMEP and BMEP over RON 91 between 1000 and 5000 RPM. RON 91 has better performance than RON 100 between 4000 and 6000 RPM during lean combustion, and between 5500 and 6000 RPM at rich combustion. Frictional performance analysis concludes that energy loss due to friction is highest for RON 95 and RON 98 for both $\lambda=0.9$ and $\lambda=1.1$.

At 3000 RPM, RON 95 followed by RON 98 has highest brake thermal engine efficiency over RON 100. RON 91 with 4.05 % less BTEE at 3000 RPM than RON 95 as a performance fuel contributes to higher greenhouse emissions. For lean combustion, RON 100 has the highest BTEE over RON 95 and RON 98, where RON 91 has the least BTEE by .63 % than RON 100. Heat transfer rate of RON 91 is highest of all fuels due to its high calorific value, where more heat transfer loss is obtained for low performance fuels RON 95 and RON 98 when $\lambda=0.9$. High performance fuels RON 91 and RON 100 shows peak heat transfer loss when mixture turns lean ($\lambda=1.1$). Study also concludes that cylinder pressure is a function of calorific value of fuel and AFR. Rich mixture have higher cylinder pressure and exhaust gas temperature over lean mixture. RON 91 shows a maximum EGT of 1376 K when $\lambda=0.9$, 50 K more than $\lambda=1.1$. The same degradation in EGT is noted in all fuels throughout the engine speed.

From emission analysis, brake specific CO emissions is higher for lean combustion ($\lambda=0.9$) where RON 91 contributes substantially higher CO emissions with an elevation of 100.85 g/KW/hr than $\lambda=1.1$ at 6000 RPM. RON 91 and RON 100 as performance fuels have more CO tailpipe emissions, where 10 % rich mixture produces 26408 ppm CO than 10 % lean mixture. For $\lambda=0.9$, RON 95 has the highest contribution of 2707 ppm of NOx over RON 98 with 2691 ppm followed by RON 91 and RON 100. Whereas when $\lambda=1.1$ NOx emissions are overturned with highest proportion noted in RON 91 with 6379 pp. Unburned HC emissions are least recorded for RON 91 and 100 where $\lambda=0.9$ and $\lambda=1.1$ have similar combustion emissions throughout all operating engine speed.

Research B – The second stage of research (5.2) of Ricardo WAVE 1-Dimensional combustion modelling analysis on a Ford Sigma gasoline powertrain was done to study the unburned hydrocarbon emissions due to molecular diffusion of lubricant and fuel in piston top land crevice volume for a standard radial clearance of .35 mm. The cumulative fuel mass absorbed by the crevice volume is recorded higher at an engine speed of 4000 and 5000 RPM with .001145 grams of fuel for $\lambda=0.9$. The accumulated fuel mass is 3.10 % of injected fuel mass, later released into combustion chamber at suction stroke which can lead to preignition or knock. Lean AFR ($\lambda=1.1$) has the least fuel mass absorption in crevice volume, where 24 % reduction in fuel absorption is noted at 5000 RPM. A proportion of unburned fuel in top land crevice is being absorbed by oil layer where at 4000 RPM,

highest composition of .000148 grams is shown when $\lambda=0.9$ and 15.54 % decreases in oil layer consumption is shown at $\lambda=1.1$. This simulation results conclude that 12.92 % of fuel mass in piston top land crevice are absorbed by the lubricant layer over a 720-degree cycle. Unburned fuel mixtures are accumulated at maximum rate when the piston is at TDC at end of compression stroke. These characteristics are of higher concentration between 4000 and 5000 RPM.

Research C – The third stage of research (5.3) of Ricardo WAVE 1-Dimensional combustion modelling analysis on a Ford Sigma gasoline powertrain was done to study the unburned hydrocarbon emissions due to the radial clearance of piston top land crevice volume for a standard value of 0.25,0.35,0.45, and 0.55 mm. Study shows that hydrocarbon emissions and unburned fuel absorption rate increase with increasing crevice radial clearance. An analysis at 1000 RPM concludes that HC emissions increase by 19.53 % when radial clearance increases from 0.35 mm to 0.45 mm. A steep inclination of 48.53 % higher emissions than 0.35 mm is noted for a radial clearance of 0.55 mm during analysis. Varying radial clearance simulation analysis also shows that, the unburned fuel mass trapped in lubricant layer remains the same as the lubricant layer have limitations in accumulation of fuel droplets.

Research D -The fourth stage of research (5.4) of Ricardo WAVE 1-Dimensional combustion modelling analysis on a Ford Sigma gasoline powertrain was done to study the influence of varying lubricant dynamic and kinematic properties on engine friction characteristics. Simulation analysis concludes that lubricant specifications like kinematic viscosity and density do influence on frictional energy loss and friction torque is a microscopic scale. SAE 10W-40 have the highest FI with .000625% more energy loss than SAE 0W40 which is recorded to have least frictional loss. Lubricants SAE 0W-40 and SAE 5W-50 show similar FEL at an engine speed of 6000 RPM.

Research E – The fifth and final stage of research (5.5) of ANSYS Forte R2 3-Dimensional combustion modelling analysis on Ford Lynx diesel powertrain was done to investigate the influence of AFR, and EGR on thermodynamic properties and emissions. The study done on $\lambda=0.9$, stoichiometry and $\lambda=1.1$ concludes that higher the injected mass into combustion zone, more the chemical heat release. The temperature increases at CA -2 degrees releases NOx in higher concentrations. $\lambda=1.1$ have more combustion zone temperature with 2846 K over 2816 K for $\lambda=0.9$. This results in a NOx emission index of 58.72 grams per kilogram of fuel burned for $\lambda=0.9$, where $\lambda=1.1$ release 62.01 grams.

Introduction of 10% and 20% EGR resulted in varying combustion properties. The key observation was reduction of in cylinder pressure by 2.99 % to 139.69 bar for 10% EGR and 6.39% decrease in cylinder pressure for 20 % EGR. Moreover, cylinder combustion zone temperature was reduced by 49 K for 10 % EGR and 129 K for 20 % EGR. Simulation analysis highlights about the increased CO emissions due to EGR. An 8.61 % increase in CO emissions are noted for 10% EGR whereas 11.44 % increase is noted for 20% EGR compared to that of stoichiometrically validated analysis without EGR. The research concludes that 10% EGR resulted in 25.19% reduction in NOx emission index, whereas 20% EGR reduced the NOx concentration by 55.08 %.

6.2 Software

Validation and evaluation of Ford Sigma and Ford Lynx combustion model on Ricardo WAVE and ANSYS Forte R2 are done in both software's to study reliability of combustion modelling in 4 different simulation validation. Apart from the research done after successful validation of gasoline combustion modelling in Ricardo WAVE and diesel powertrain modelling in ANSYS Forte R2, validation of Ford lynx combustion model in Ricardo WAVE was unsuccessful due to the insignificant compression and expansion data compared to manufacturer data. Combustion resulted in

uncontrolled pressure hike which can damage the components with intense knock. Certain simulations with burn duration and 50 percent mass fraction burned point was changed in numerous simulations to combust without knock. This unsuccessful validation could be due to the turbo diesel variant of Lynx with transient over boost where introduction of turbo is also done to validate the model. A polytropic index of 1.25, 1.3 and 1.35 could not fit the compression, combustion and expansion with Lynx cylinder geometry and parameters. Meanwhile the study was successful in ANSYS Forte R2 as the injection events and parameters gave an efficient pressure vs crank angle diagram for an engine speed of 4000 RPM. As the research was comparing the characteristics of research octane number, key flame characteristics like burning radii and flame velocity were not shown when requested, even with the sub models on. This data should have been an important way to compare the influence of RON on combustion performance and emissions.

After successful validation of Ford Sigma geometry in WAVE, the validation was unsuccessful in Forte with analysis on various spark events, throttle body injection and direct injection, chemistry sets of iso-octane, introduction of EGR and cylinder bowl geometry as the pressure rise in combustion zone results in intense pressure rise over 90 bars for the lean fuel mass injected. Certain simulation run on throttle body injection helps to fit the compression whereas combustion and expansion could not be regulated to fit to a maximum bar of 62 bar as in manufacturer data. This navigated the research to study the influence of RON in Ricardo WAVE and combustion of diesel, EGR on Forte R2 where simulation thermodynamic analysis could not be compared in both combustion modelling platforms.

Both simulation software's were efficient in providing carbon and nitrogen oxide emissions outputs, whereas lubricant consumption and lubricant emissions could not be addressed in simulation outputs. Even with the introduction of chemistry of compounds in a lubricant to the chemistry sets, simulation software like ANSYS Forte R2 does not have an indication to sulphur emissions which was one of the strategies of research that could not be studied. Due to the presence of wide variety of chemical compounds in a lubricant, with Research B on molecular diffusion showing that the fuel being absorbed by oil layer, it is important to analyse the emissions due to combustion of lubricant as lubricant manufacturer data in section 5.4 shows a flash point temperature below 250 degrees for all multigrade lubricants with performance additives.

7. Conclusion

The study shows that lean mixture is not a fundamental AFR for manufacturers due to volatile thermodynamic emissions of NO_x, unburnt hydrocarbon with compromised performance output from the designed engine geometry. Rich AFR have better performance and emission characteristics where CO emissions are of higher proportion which is a compromise done by manufacturers while extracting the potential of the powertrain. From a performance perspective, RON 100 has more performance than RON 91, which makes it the feasible performance fuel for engines. RON 100 and RON 91 have similar output as high-performance fuels followed by RON 98 and 95, which are considered as low performance fuels for both $\lambda=0.9$, $\lambda=1.1$. From an emission perspective, for all fuels taken for analysis, 10 % rich mixture have highest contribution to CO emissions, whereas lean mixtures have high rate of NO_x production. Even though RON 95 and RON 98 have least engine performance, NO_x production and unburned hydrocarbon emissions, another key advantage of both fuels is that they are superior in the reduction of CO emissions.

Molecular diffusion is an inevitable phenomenon in ICE due to the presence of piston top land crevice volume. Internal combustion engines with continuous operation can result in wear and tear of the cylinder block in the long-term run even with efficient or non-efficient periodic maintenance. This concludes that the fuel absorption and accumulation in piston top land and oil layer increases as a function of powertrain life. Certain combustion phenomena like pre ignition and knocking are expected to rise in engines over the long term, which raises questions about the reliability of components due to variation in combustion zone pressure and temperature. It is noted from simulation that these unburned fuel mixtures are released into combustion zone during the event IVC where combustion zone turbulence release the mixture from crevice land. Controlling the molecular diffusion are strictly related to crevice volume parameters whereas introduction of IVC event between IVO and EVC might release more UHC to atmosphere.

Lubricants have a very negligible impact on the frictional performance of an engine. Simulation results conclude that different grades with varying oil properties like viscosity and density does have minute influence in friction characteristics. The choice of lubricant grade should be a consideration of engine operating atmospheric conditions where the atmospheric conditions vary globally. The transformation of viscosity from 40 degrees to 100 degrees are a key characteristic over the choice of additives to determine the oil performance which helps the powertrain to increase efficiency in operation.

EGR have a strong influence on combustion characteristics, where engine operating temperature and pressure are influenced by the composition. Increase in EGR also notified the pressure loss in the system which can affect the performance output of an engine. CO emissions increased due to the presence of recirculated CO₂ whereas NO_x is regulated to great extent according to simulation results.

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