CONCEPT OF REACTIVITY CONTROLLED COMPRESSION IGNITION ON A SINGLE CYLINDER AUTOMOTIVE ENGINE

A thesis report

Submitted in partial fulfilment of
requirements for the degree of

Master of Engineering
in

CAD/ CAM Engineering

by

Komal Virdi

Registration No.: 801784009

Under the Supervision of

Dr. Kaleemuddin Syed

(Associate General Manger in R&D Department at Greaves Cotton Ltd., Aurangabad)

And

Prof. S.K.Mohapatra

(Senior Professor, Department of Mechanical Engineering at TIET, Patiala)

MECHANICAL ENGINEERING DEPARTMENT

THAPAR INSTITUTE OF ENGINEERING & TECHNOLOGY, PATIALA

DECLARATION

I, Komal Virdi, student of M.E in CAD/CAM specialisation from Thapar Institute of Engineering and Technology, Patiala, hereby declare that I take full responsibility for the information, results, conclusion etc. provided in this thesis/project titled “Concept of RCCI on single cylinder automotive engine”. I further declare that in case of violation of the intellectual property rights or copyrights found at any stage, I will be solely responsible for the same.

Date: 9th August 2019

Komal Virdi

M.E (CAD/CAM)

Roll no. 801784009

CERTIFICATE

I, hereby declare that the thesis entitle "Concept of Reactivity controlled compression ignition on a single cylinder automotive engine" is an authentic record of my work carried out as requirements for the award of degree of Master of Engineering in CAD/CAM at Greaves Cotton Ltd., Aurangabad under the guidance of Dr. Kaleemuddin Syed (Industrial Guide) and Prof. S.K.Mohapatra (Thesis guide). No part of the matter embodied in this report has been submitted to any other university for the award of any degree.

Date: 9th Aug 2019

Komal Virdi

Roll no. : 801784009

T.I.E.T, Patiala

It is certified that above statement made by the student is correct to the best of our knowledge and belief.

Dr. Kaleemuddin Syed

(Associate General Manger)

R&D Department

Greaves Cotton Ltd., Aurangabad

Prof. S.K.Mohapatra

(Senior Professor)

Department of Mechanical Engineering

T.I.E.T, Patiala

ACKNOWLEDGMENT

Present project work won't be possible unless I express my feeling of gratitude towards number of persons for support, cooperation and patience.

Work was carried out under supervision of Dr. Kaleemuddin Syed at Greaves Cotton Ltd, Aurangabad for which I would like to express my deepest appreciation and thanks for guiding me since it was assigned to me and sharing his experiences. I am grateful and appreciative for all the knowledge shared with me during course of the project.

I express and extend my thanks to Prof. S.K.Mohapatra for constant supervision during course of this work.

I further extend my thanks to Mr. Atul Jahirabadkar (Associate General Manger-Design head) for his constant support and inspiration during design, simulation and experimental work.

I appreciate the support provided and patience shown by Prashant Dharmapurikar (DGM), Shashank Alegaonkar (Manager), Vikas Thorat (DGM), Suresh Nallaperumal, Uday Singh (Manager), Sanjeev Ronte (Asst. Manager) and whole CAD TECH team during design and simulation phase.

I also appreciate the support given by Pranjal Jagtab (Senior Manager), Prasant Nakhod (Manager), Ganesh, Salman Sayyed, Tanmey Deshpande during assembly and final engine testing phase. I would like to thank all those who have contributed directly and indirectly.

Last but not the least, I am thankful to my parents and my little brother for their support and inspiration during the project work.

(Komal Virdi)

Table of Content

ABSTRACT

The cost of conventional fuels is increasing day by day. To overcome this, alternative fuels are getting more and more attention with additional benefits of exhaust emission reduction and in delaying faster depletion of crude oil reserves.

Because of better fuel economy of Compression Ignition (CI) engines, users have been preferring these not only for off road vehicles but recently even for on highway passenger vehicles. An important feature of CI engine is that it can tolerate wide variety of fuels and this includes both liquid and gaseous fuels. Continuous research is going on in technologies like hybrid, biofuel vehicle, etc. to make them able to produce greater power than conventional diesel engine. Also, now a days for transportation, users are relying more on natural gas due to its lower price and not only it has cut running cost but has also made people less dependable on premium liquid petroleum products like Ultra low Sulphur Diesel.

To retain fuel economy benefits of CI engine and tailpipe emission of gaseous fuel spark ignition engine, one of the available options is the dual fuel technology, also now a days known as Reactivity Controlled Compression Ignition (RCCI) technique. This technique has an advantage that the dual fuel engines need very less modifications to convert it from diesel engine and by this technology one can vary the reactivity inside the combustion chamber by varying the quantity of fuels of entirely different reactivity. Here high reactive fuel is used only for making the flame propagate throughout the low reactive air fuel mixture in combustion chamber at constant rate. In most of the cases, Compressed Natural Gas (CNG) is used as a low reactive fuel and diesel or B20 fuel as a high reactive fuel. Fuels like ethanol and petrol can also be used as a low reactive fuel but CNG being in gaseous form requires less modification to use it as a low reactive fuel.

CNG can be mixed with air through port injection or direct injection. Subsequently, the amount of diesel injected inside cylinder is decreased in proportion of injected gaseous fuel. Amount of CNG that can be mixed is entirely limited by exhaust emissions of HC, CO and knocking produced inside the cylinder. According to the literature survey done, the maximum diesel substitution is 70%.

For this project 650 cc diesel, water cooled engine is taken as baseline engine with rated power of 10.8 kW @ 3200 rpm and maximum torque of 38 Nm @1600 rpm with compression ratio of 19.2:1 having re-entrant type piston bowl. By analysing the design and packaging of current engine, it is required to modify intake manifold for CNG port injector and Temperature Manifold Absolute Pressure (TMAP) sensor mounting, and finally modify the piston bowl volume and shape for lower compression ratio in the range of 15 to 17:1 for better air utilisation.

GT-Power software of Gamma Technology is used for one dimensional simulation of baseline and modified engine performance. This simulation predicts the peak cylinder pressure achieved during combustion, performance and emission. Initially separate models of baseline engine with Diesel and CNG as single fuel are made in the software. Then this model is converted to RCCI to study the effect on performance and emission by varying percentage of fuels. Then the results are verified with help of engine steady state performance test.

1.0 Introduction

1.1 Emission norms and effect of exhaust emission

Pollution on earth is increasing day by day leading to a number of problems like global warming, acid rain and increase in the number of diseases. Vehicles, whether on or off-road, are contributing large amount of air pollution. To keep pollution in control, emission norms have been developed and implemented in all developed and most of developing countries. Currently emission standard BS-4 is being followed in India. From 1st April 2020, Government of India is going to implement BS-6 emission norm for vehicles of all class. The emission limits for BS-6 are given as follows:

ARAI® Progress through Research — M & N Category Vehicle with GVW< 3.5 tons : BS VI — 27

The Emission Standards for Bharat Stage VI (BS-VI)
for category M and N vehicles having Gross Vehicle Weight not exceeding 3500kg, manufactured on or after 1 April 2020 for all models, as per GSR 889 dt. 16 Sept 2016.

		Reference Mass (RM) (kg)	Mass of Carbon Monoxide (CO)		Mass of Total Hydrocarbons (THC)		Mass of Non-Methane Hydrocarbons (NMHC)		Mass of Oxides of Nitrogen (NOx)		Combined Mass of Hydrocarbons and Oxides of Nitrogen (THC + NOx)		Mass of Particulate Matter (PM)		Number of Particles (PN)	
			L1 (mg/km)		L2 (mg/km)		L3 (mg/km)		L4 (mg/km)		L2 + L3 (mg/km)		L5 (mg/km)		L6 (numbers/km)	
Category	Class		PI	CI	PI	CI	PI	CI	PI	CI	PI	CI	PI (1)	CI	PI (1)(2)	CI
M (M1 & M2)	—	All	1000	500	100	—	68	—	60	80	—	170	4.5	4.5	6.0X 10^{11}	6.0X 10^{11}
N1	I	RM ≤ 1305	1000	500	100	—	68	—	60	80	—	170	4.5	4.5	6.0X 10^{11}	6.0X 10^{11}
	II	1305<RM ≤ 1760	1810	630	130	—	90	—	75	105	—	195	4.5	4.5	6.0X 10^{11}	6.0X 10^{11}
	III	1760<RM	2270	740	160	—	108	—	82	125	—	215	4.5	4.5	6.0X 10^{11}	6.0X 10^{11}
N2	—	All	2270	740	160	—	108	—	82	125	—	215	4.5	4.5	6.0X 10^{11}	6.0X 10^{11}

PI = Positive Ignition, CI = Compression Ignition

(1) For positive ignition, particulate mass and number of particles limit shall apply only to vehicles with direct injection engines.

(2) Until three years after date of implementation for new type approvals and new vehicles, particle number emission limit of 6.0 X 10^{12} /km shall apply to BS VI gasoline direct injection vehicles upon choice of the manufacturer.

Fig. 1 Emission norms of BS-6 for light duty vehicle [1]

1.2 Low temperature combustion (LTC) technique

There are various developed strategies like Exhaust Gas Recirculation (EGR), Selective Catalytic Reduction (SCR), catalytic converter, etc. to reduce emissions values. One of the modern techniques is LTC, both lower emission and better fuel economy can be achieved by this technique. This technique is based on concept of longer ignition delay, low equivalence ratio and combustion phasing

(CA50). One of the LTC techniques is Homogenized Charged Compression Ignition (HCCI), in which mixture of fuel and oxidizer enter into the cylinder, which is compressed and ignited. Premixed charge compression ignition (PCCI) came after HCCI which is more practical version of HCCI, in which pre-mixed charge (obtained by early injection) of air-fuel mixture is compressed and ignited. Followed by PCCI, the RCCI concept was introduced. All the above said techniques contribute in reducing NOx as well as particulate matter (PM).

1.2.1 RCCI or Dual fuel engine

RCCI stands for reactivity controlled compression ignition which is advanced version of HCCI. In this technique there is more control over the combustion rate which otherwise results in aggressive knocking inside HCCI engine. The main reason for which RCCI is known is to minimize **trade-off between NOx and PM**. NOx is formed because at higher temperature, when atmospheric nitrogen present, breaks down to reactive monotonic nitrogen radicals which further react with oxygen to form NOx. PM is formed locally due to lack of presence of sufficient air which leads to improper oxidation of charge. In RCCI, NOx is reduced because it is a low temperature technique and PM is reduced because the charge is more homogenised. The mixture inside combustion chamber of RCCI is more homogenised as compared to diesel engine because of use of low reactive fuel (gasoline, ethanol, CNG, etc.) which forms more homogenised mixture with air than diesel (high reactive fuel).

In RCCI/ dual fuel technique two completely different fuels of different reactivity are compressed and ignited. Reactivity of the fuel is basically related to the amount of n-cetane that exists in the fuel and amount of cetane in the fuel directly indicates self-ignition property of fuel. In this way performance of diesel engine with tail pipe emission of low reactive fuel (gasoline, CNG, etc.) can be achieved. Following figure shows P-V diagram of the dual fuel cycle.

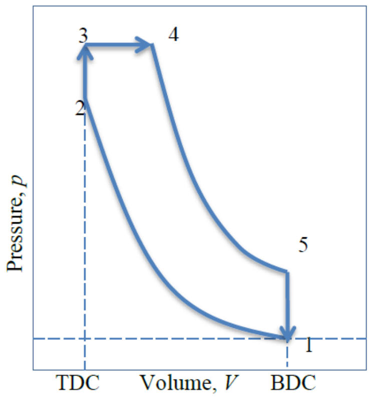


Fig. 2 P-V diagram of dual fuel cycle [2]

Dual fuel cycle is the combination of Otto and Diesel cycle. In this way dual fuel engine has advantages of both diesel and gasoline engine. 1-2 is isentropic compression, 2-3 constant volume heat addition, 3-4 constant pressure heat addition, 4-5 isentropic expansion and 5-1 constant volume heat rejection. Since compression ratio of the diesel engine is higher than Otto and Dual fuel cycle, therefore, it has the maximum efficiency. Dual fuel cycle efficiency lies in between Diesel and Otto cycle efficiency.

2.0 Applications:

1. **Cummins QSK50** engine used for land based gas and oil drilling application. With rated power up to 1864kW. With the maximum gas replacement of 50-70%, this engine has flexibility to operate on low quality, low cost gas at lower substitution ratio or to operate on higher quality of gas with higher substitution ratio in order to achieve environmental, economic benefits and also due to energy policy.

Fig. 3 Cummins QSK50 Dual fuel engine [3]

2. **Hanson et al. [4]** A hybrid vehicle was also developed using the dual fuel technology. It is 1.9L diesel with 2009 Saturn Vue chassis.

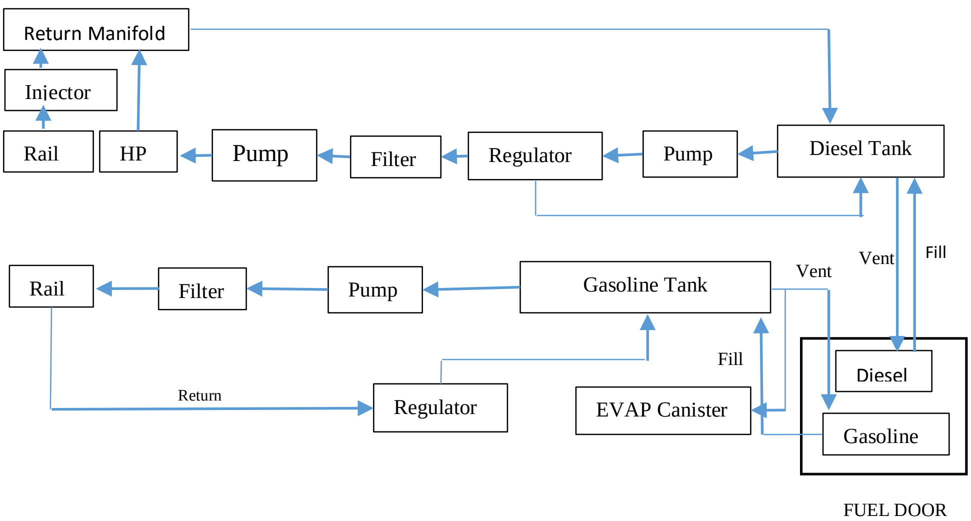


Fig. 4 Fuelling system of vehicle [4]

3.0 Literature Review

S S Thipse et al. [5] In RCCI CNG and air mixture enters in cylinder which is then compressed and ignited with the help of high reactivity fuel i.e. diesel. Dual fuel engine have flexibility to run on pure diesel operation. In the dual fuel mode, gas supplement ratio is another and important parameter, this requires number of experimental trials for engine optimization. Therefore simulation is performed to reduce cost as well as time. Diesel injection quantity was controlled by suitable mechanism to that controlled governor position. A test matrix is prepared with three different speeds and loads. Engine is operated under pure diesel mode and then on dual fuel mode with gas supplement ratio (GSR) varying by 20%, 40%, 60%, 80% and maximum depending on stability of operation.

$$GSR = \frac{Mass\ of\ natural\ gas}{Mass\ of\ natural\ gas + Mass\ of\ Diesel} \tag{1}$$

The engine was maintained at required speed and load under normal diesel operation, after which CNG is injected, which results in increase in load output from the engine, simultaneously the quantity of diesel is reduced until initial value of the load is achieved. Under idle condition or at 10% of load the engine works on pure diesel because dual fuel engine at lower load have lower combustion efficiency. At part load the diesel replacement may go up to 30% to 50%. Maximum replacement of diesel is 75%. Above this value there are high chances of engine knocking. At higher load there may be 40% or no replacement of diesel. Noting down the trends of emission, the value of the GSR is decided. Full Throttle Performance (FTP) and European Transient Cycle (ETC) is conducted under normal diesel operation as well as during dual fuel operation. The compression ratio of this dual fuel engine is 17.5:1, Bore= 97 mm, Stroke= 128 mm. In dual fuel operation, low thermal efficiency is due to incomplete combustion of natural gas which results in higher Hydrocarbon (HC) and carbon monoxide (CO) present in the exhaust emission. This is however compensated by low cost of CNG. Under certain condition dual fuel operation has more effective cost than diesel. HC and CO increases with increase in the percentage of CNG. Figure 4 shows the diesel percentage vs. load diagram for dual fuel and pure diesel operation.

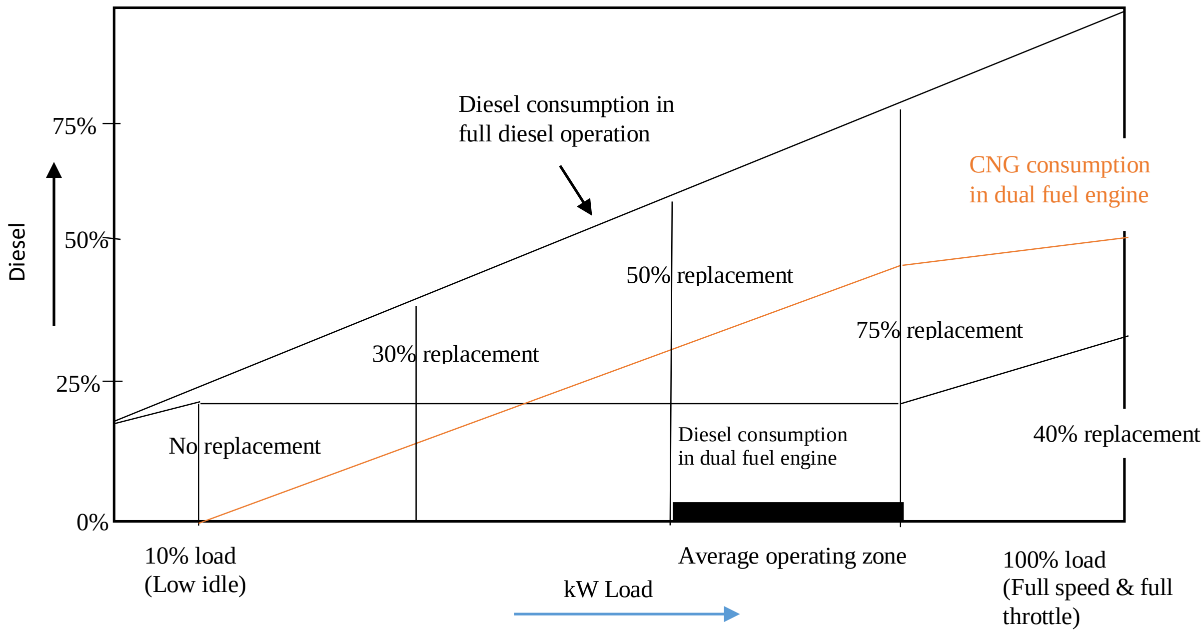


Fig. 5 Diesel quantity and load diagram [5]

Derek Splitter, et al. [6] In dual fuel combustion process, burn rate event is controlled by zones of reactivity from the most to the least reactive. Excess lean mixture increases the gross efficiency. However, pumping work is increased. This leads to net and gross thermal efficiency to be mutually exclusive. Brake thermal efficiency (BTE) in terms of gross thermal efficiency (GTE), pumping (PMEP), friction (FMEP) and indicated mean effective pressure (IMEP) can be formulated as follows:

$$BTE = GTE \times \left[1 - \frac{PMEP+FMEP}{IMEP}\right] \tag{2}$$

In order to achieve BTE of 55%, extreme low pump mean effective pressure (PMEP) and friction mean effective pressure (FMEP) is required. It is finally evaluated by simulation that with lean mixture, high compression ratio (CR) and 50% lower heat transfer and incomplete combustion 47% of BTE and 60% of GTE can be achieved. Combustion phasing is controlled by varying the percentage of high and low reactive fuel. Around **50-70% of the heat flux in piston crown in RCCI technique is reduced at same load**. The highest GTE happens when volumetric efficiency is high and heat transfer losses are minimum. Both the cooling of piston and volumetric efficiency are inversely proportional. **High compression ratio results in higher energy extraction inside cylinder which reduces exhaust temperature**. Both the gross and net thermal efficiency are inversely proportional, because as GTE increases the exhaust enthalpy reduces, the turbine efficiency thus decreases NTE (Net Thermal Efficiency). NTE can be optimized by either increase in efficiency of turbocharger or by improving combustion efficiency along with reduction in heat transfer.

Willi, Martin et al. [7] Factors effecting thermal efficiency are combustion timing, duration, heat losses, combustion efficiency, pumping and friction losses. HC and CO major source is Crevice volume and squish volume. HC is crevice dependent. It depends on compression ratio, volume of crevice and engine rpm. Based on experiment RCCI peak pressure has reached 90 bar however in case of pure diesel operation it is 70 bars. RCCI piston usually runs at 60ºC lower temperature than CDC engine and according to the prediction done in simulation the piston shape is important, reducing the squish has the potential of reducing HC. Bowl has to be designed to reduce heat losses. With same crevice the performance of low CR matches with that of higher CR. Thermal losses in case of RCCI engine are less due to more expansion work available. **With help of RCCI there is 5-15% rise in the brake thermal efficiency+ NOx and PM reduction in cylinder+ high pressure system can be operated at low pressure this further efficiency increase**. With the reduction of crevice with its height reduced by 1.5mm and undercut reduced to 0.5mm HC can be reduced by 60% and gives 1% increase in GTE. **With reduction of CR helps in getting the intake temperature and EGR quantity to increase.**

Adwait Jadhav et al. [8] Engine testing is done on Speed- throttle mode. Performance at six different speeds was taken and it was evaluated that it is more economical at lower load to run the engine on pure diesel. **BSFC of dual fuel is reduced by 8% as compared with pure diesel operation**. However, the volumetric efficiency alongside has reduced by 5%.There are two type of dual fuel engines:

1. CNG dedicated method: In this method the mixture of CNG and air along with diesel in cylinder is ignited by means of spark plug. However this method increases the cost, reduces the compression ratio.
2. CNG /Diesel dual Fuel method: In this method the air and CNG mixture is ignited by means of diesel direct injection. This method includes characteristics of both the Otto and Diesel cycles therefore initial cost is less.

Steady and transient test is performed for engine emission and performance test. Testing of dual fuel engine is done with GSR value of 20%, 40%, 60%, and 80% at different speed and load conditions. Up to 94% reduction in PM emissions observed, but NOx emissions is found to be higher and HC & CO were found to be more than CDC engine. **Closed loop lambda control system for lean combustion is incorporated in engine to achieve performance and emission targets**. By simulation in GT-Power, it is observed that in dual fuel engine, there is 10% less torque and 5% less BSFC. BSFC at 50% load has increased by 13% and by 9% at 70% load for keeping the same brake power as CDC. However, fuel economy has improved by 10% in dual fuel mode. Minimum BSFC is obtained at 60% of substitution rate of diesel. It was observed that in order to keeping same engine speed and torque, there is drop in engine load carrying capability after **80% substitution of diesel**. Peak pressure in case of dual fuel mode is made same as that in pure diesel operation so that it does not harm the engine structures and transmission systems. Comparing diesel with dual fuel, brake power is same in both modes. However volumetric efficiency is reduced due to gas fumigation in intake manifold and slow combustion is observed. Reduction in oxygen or air inside cylinder also reduces amount of energy going inside the system. With increase in the gas supplement ratio, PM decreases whereas the HC and CO increase. But at higher load that the HC and CO concentration decreases with higher GSR.

Benajes Calvo et al. [9] This paper discuss about the effect of geometry of piston bowl on RCCI engine at all loads and how combustion profile is effected with single and double injection. **The equivalence ratio and reactivity gradient is maintained in cylinder with the help of gasoline fraction and direct injection time variation.**

$$Equivalence\ ratio = \frac{Actual\left(\frac{Fuel}{Air}\right)}{Stoichiometric\left(\frac{Fuel}{Air}\right)} \tag{3}$$

Main source of HC and CO is crevice and squish region. Combustion efficiency is given as:

$$Comb.Eff = \left(1 - \frac{HC}{m_f} - \frac{CO}{4*m_f}\right) \times 100 \tag{4}$$

HC and CO are the amount of unburned hydrocarbon and carbon monoxide, m_f is total mass of fuel injected.

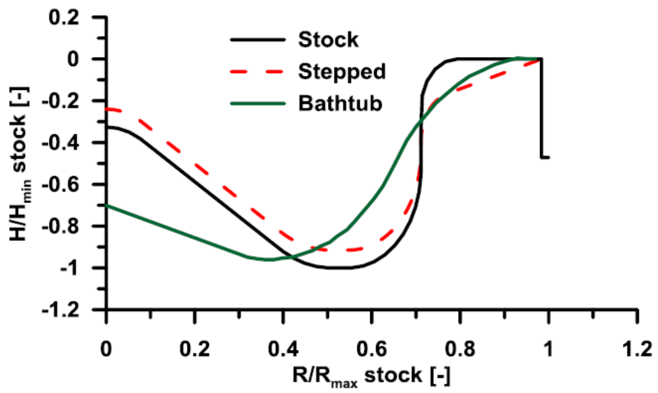


Fig. 6 Different piston geometry [9]

Bowls are designed so as to reduce squish and crevice regions also considering lower heat transfer losses. Stepped bowl helps in reducing surface area by 4.6%. Bathtub type piston reduces the area by 16% as compared with stock geometry. During testing indicated mean effective pressure (IMEP), start of combustion (2% increase in cumulated heat) and CA50 along with ringing intensity (RI) are determined.

RI is determined by formula:

$$RI = \frac{1}{2\gamma}\frac{\left[0.05*\left(\frac{dP}{dt}\right)_{max}\right]^2}{P_{max}}\sqrt{\gamma RT} \tag{5}$$

γ is the ratio of constant pressure by constant volume of specific heat, dP/dt is peak pressure rise rate (PRR), P_{Max} is the maximum in-cylinder pressure, R is ideal gas constant and T is maximum in-cylinder temperature. The combustion start is mainly governed by timing of main injection. It was also evaluated that with the advancement of both main and pilot injection the **combustion in two stages is converted to one stage which was observed to be beneficial for reducing HC and CO, simultaneously it keeps the NOx and PM emission below Euro 6 norms level.** In cylinder temperature is same in stock, stepped type and bathtub type piston. Therefore heat transfer losses is directly related with bowl surface area. As the diesel SOI is retarded, the first rate of heat release (RoHR) at first stage is reduced, whereas the second stage of combustion has higher RoHR, outcomes of this are higher in cylinder temperature and NOx. For the medium load bathtub piston is most preferred because of lower surface area. Shorter mixing time promote combustion of diesel in less premixed manner. This results in smaller first stage of combustion and higher heat release in the second stage of combustion resulting in higher NOx. At medium load higher RoHR is achieved by bathtub piston because of lower surface area and due to less mixing of charge.

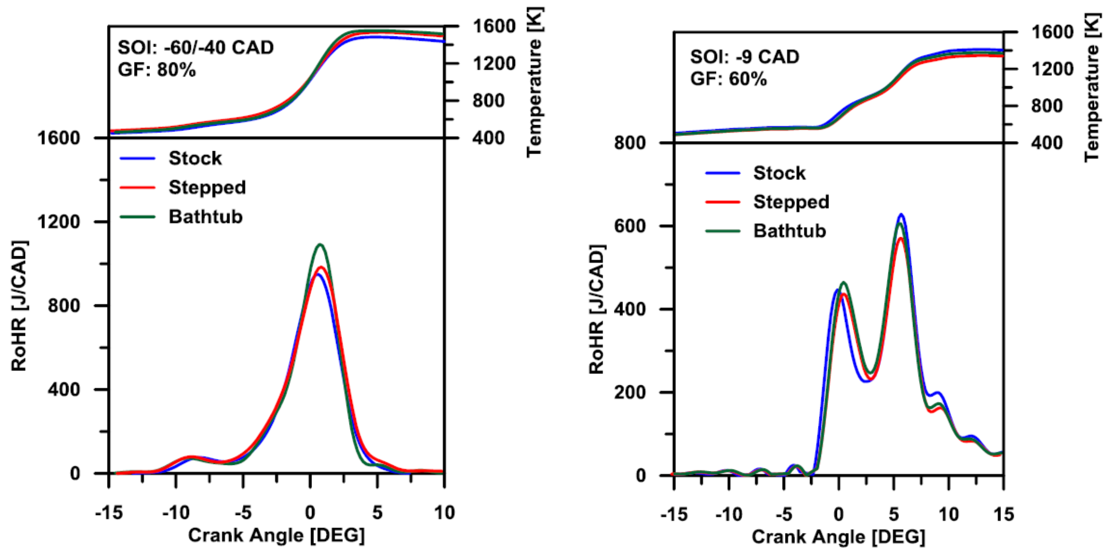


Fig. 7 Double (Left) and single (right) injection heat release w.r.t crank angle and mean temperature at medium load (GF- Gasoline fraction) [9]

At higher load there is significant increase in soot emission for bathtub type piston. This is because the excessive shallow bowl reduces the charge motion. Stepped piston is seen to have lower BSFC, bathtub piston worsen brake specific fuel consumption (BSFC) as compared with stock piston. **Also HC emission in all the pistons are same whereas CO emission is improved in stepped type**

piston. At the medium and low load stock piston gives highest combustion efficiency whereas in case of higher load stepped piston shows the highest combustion efficiency.

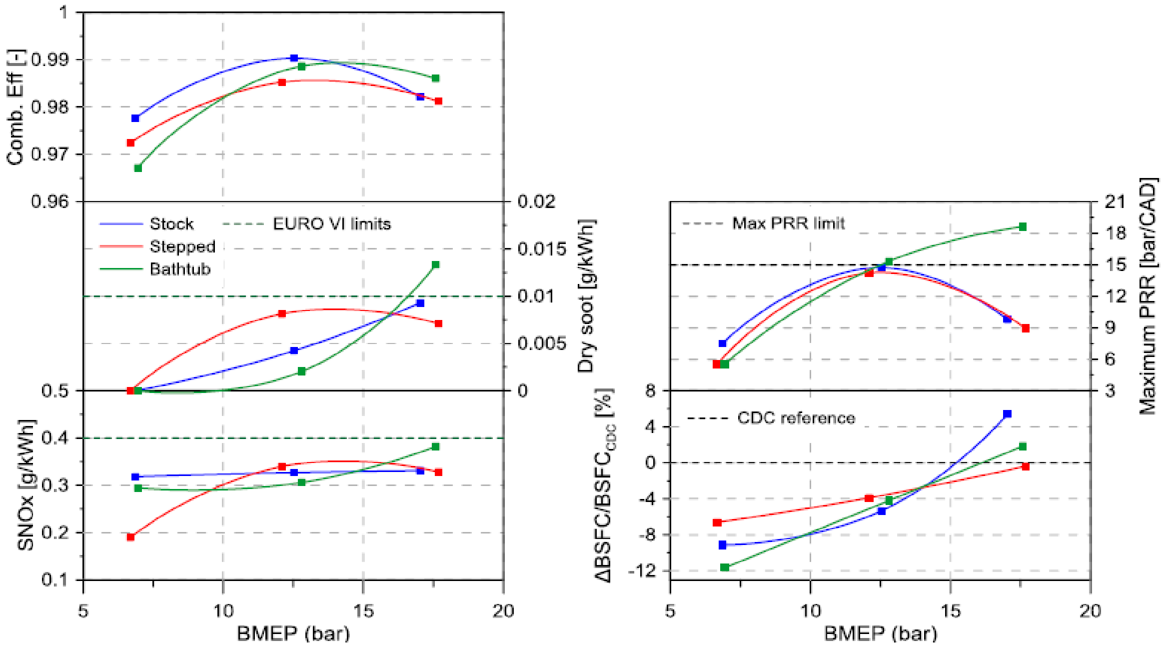


Fig. 8 Combustion efficiency, soot, NOx, Peak pressure rise rate and BSFC (%) for different piston [9]

Scott J Curran et al. [10] In RCCI technique helps in getting reactivity stratification, temperature and equivalence ratio gradient in cylinder. **The first injection strategy controls the reactivity in the squish region and this charge is then ignited by second injection which act as an ignition source.** Swirl ratio is directly proportional to the BTE. This also indicates a benefit from increased in-cylinder mixing also increase heat transfer losses. RCCI combustion leads to increase in both aldehydes and ketones. Furthermore, the increase in carbonyl species from RCCI. The RCCI particle number concentration was less than CDC or PCCI by 10 to 470 nm particles numbers. RCCI technique also reduces particles geometric mean diameter Particle mass measurements were collected on Teflon-coated quartz-fibre filters.

Pushpendra Upadhyay et al. [11] Combustion phasing in RCCI engine depends on gasoline to diesel ratio. RCCI engine are less prone to knocking than HCCI engine but as compared to RCCI, combustion is more complete in HCCI. In RCCI have comparatively lower heat transfer losses than in HCCI and CDC because in RCCI engine cylinder wall temperature (fuel mixture near cylinder wall is low reactive/lean) is less thus result in less heat transfer. Heat transfer in HCCI and CDC is same as evaluated by experiment. Ringing intensity is reduced by retarding CA 50 but this also reduces combustion efficiency. Also another factor that was observed is that the ringing intensity of RCCI is lower than CDC and HCCI engine. Combustion efficiency of HCCI is higher than CDC and RCCI because of lower HC and CO out emission.

Engine	RI ($MW=m^2$)	EISFC (g/KWh)
CDC	12	196
	7.5	200
	6	204
	7.5	207
HCCI	12.5	180
	9.5	179
	5	178
	2.5	183
RCCI	2	184
	2.5	186
	1.5	188
	1	195

Fig. 9 Comparison of equivalent indicated specific fuel consumption (EISFC) and ringing intensity of three different technology [11]

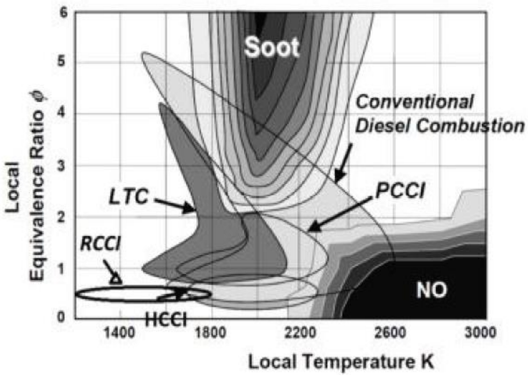


Fig. 10 Operating region of CDC and other LTC techniques in equivalence ratio vs. temperature diagram [11]

Liu. Z et al. [12] Ignition delay in case of dual fuel engine is very different than in case of diesel engine. Delay period increases initially then gradually decreases with increase in the gasoline quantity. The ignition delay is increased because of presence of gaseous fuel. This delay can be reduced by charge temperature and increase of pilot quantity. Ignition delay depends on charge pressure, equivalence ratio of fuel and air and exponentially depends on inverse of mean charge temperature.

S L Kokjohn et al. [13] For optimal thermal efficiency of any engine, the combustion phasing must be accurately controlled over a wide range of operating condition but **each fuel has its own ignition delay as well as operating range where peak efficiency can be achieved.** Inlet valve closing timing has been retarded, so as to better combustion efficiency control. KIVA-3v software for study of CFD analysis for the study of flow inside the piston bowl. This software was paired with CHEMKIN II solver which is a chemical analysis software, to study the chemical reaction taking place inside the combustion chamber. A mechanism of 45 species and 142 reaction so as to describe the combined oxidation of n-heptane (diesel) and iso-octane (gasoline). Simulation is performed with 60 deg sector mesh as shown in fig. 11.

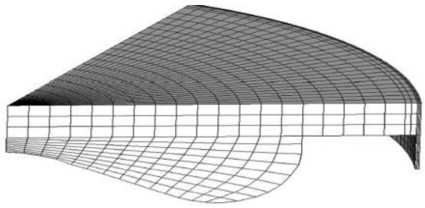


Fig. 11 Computational grid with crevice volume. [13]

The first injection helps in improving the reactivity of charge in squish region, it is the second injection that helps in ignition of charge. Since in case of RCCI engine the ringing intensity is a big problem**, the limit set is 5 MW/m^2** as an acceptable combustion noise. In RCCI at lighter load mainly combustion losses results in reduction in efficiency, at higher load the exhaust losses increases (higher exhaust temperature), heat losses and combustion losses decreases.

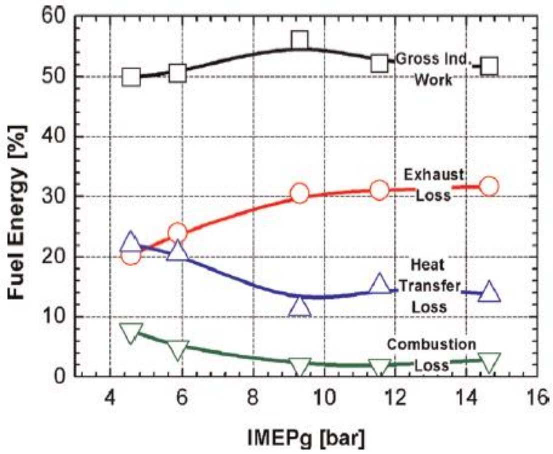


Fig. 12 Fuel energy losses (%) and load curve (IMEP) [13]

At lower load, UHC is found in centreline and crevices. At the peak efficiency region, CO is found near the cylinder wall as flame gets cold when they reach there, for UHC crevice is the main region. At higher load CO is formed near cylinder liner and centre, HC near crevice volume. CO at centre is due to rich region at centre. This can be reduced by increasing boost pressure**. UHC can be further improved by improving ring packing. In RCCI the peak equivalence ratio is 0.6**, **this reduced PM formation.** Peak combustion temperature as observed in CDC engine is 2500K, whereas in case of RCCI it is 1700 K. The combustion efficiency in RCCI is 2% lower.

$$\eta_{thermal} = \frac{W_g}{(Fuel\ energy) * \eta_{Comb}} \tag{6}$$

W_g is output work and η_{Comb} is combustion efficiency.

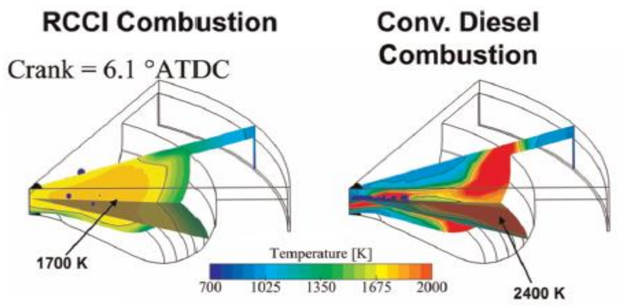


Fig. 13 Temperature contour on the plane of spray axis at 6.1º ATDC (50% burn location) for RCCI and conventional [13]

Improvement in work extraction can be improved by further working on the start and end of combustion. Though experimental observation, the RCCI gave 5.6% more fuel energy in useful CDC engine. Under RCCI and high EGR diesel engine, it is evaluated that at identical operating condition, NOx is reduced by two order of magnitude and soot by 10 factor.

Nwafor [14] The combustion characteristics of the RCCI engine are that of CI and SI engine both. Combustion process takes place in five stages: AB is the ignition delay of the diesel fuel injection, BC is the actual ignition of diesel, CD is the ignition delay of primary fuel, DE is the ignition of the primary fuel and EF is diffusion combustion stage.

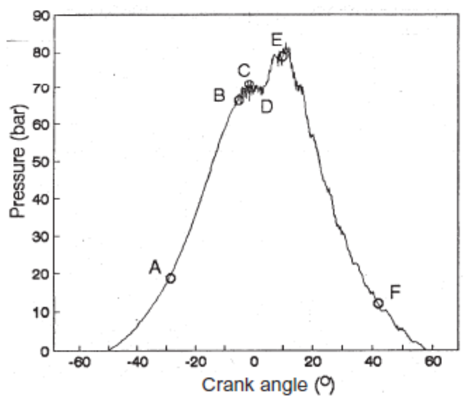


Fig. 14 Pressure crank angle diagram for RCCI/ dual fuel engine [14]

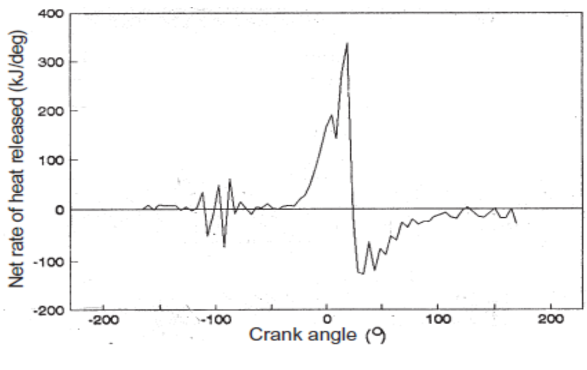


Fig. 15 Dual fuel pilot injection heat release diagram (Speed=3600 rpm; torque output=5.15 Nm). [14]

Combustion Knock in dual-fuel engine:

In diesel engine, knocking generally occurs in cold condition. Ignition delay is set to increase with dual fuel technology. Whereas in case of spark ignition engine knocking is due to auto ignition of end gases. In dual fuel engine, knocking is increased as quantity of low reactive fuel is increased in cylinder temperature, load and oxygen concentration (improve combustion rate) also effect the knock formation. In case of diesel engine ignition delay increases with increase in speed whereas in case of RCCI this delay reduces with speed. Therefore at lower speed the primary fuel has to be injected in advance. The main factors that influences knock are **pilot quantity of high reactive fuel, ignition delay period, engine load, engine speed, and gas flow rate and time interval for secondary ignition. Increasing the primary fuel and reduction in secondary fuel reduces the knocking phenomena in dual-fuel engines.** The degree of knock that occur in cylinder depends on timing between first and second ignition. It was also observed that peak pressure and initial rate of pressure rise is reduced as compared to diesel engine.

Table 1. Effect of pilot fuel/gas ratio on knock characteristics of dual-fuel engine. Engine speed = 3000 rev/min..

Load (N)	Pilot fuel (kg/h)	Gas supply (kg/h)	Total fuel (kg/h)	% Pilot fuel	% Gas supply	Mixture strength
6.61	0.305	0.626	0.931	32.76	67.24	0.557
13.18	0.292	0.720	1.012	28.85	71.15	0.620
17.72	0.271	0.792	1.063	25.49	74.51	0.667
23.78	0.295	0.846	1.141	25.85	74.15	0.730
28.82	0.313	0.882	1.195	26.19	73.81	0.779
33.87	0.331	0.907	1.238	26.74	73.26	0.823
38.92	0.343	0.936	1.279	26.82	73.18	0.866
44.47	0.307	1.008	1.315	23.35	76.65	0.922

Oxygen concentration has greater effect on smooth running of engine. In dual fuel due to substitution of gas with air quantity of air/ oxygen inside the cylinder decreases. The combustion is also improved with increase in pilot quantity at lower load.

GH Abd Soliman et al. [15] One of the major problems in dual fuel engine is that there is drop in thermal efficiency at lower load. This may be because of the reason that HC and CO amount is quite high at lower load as the flame cannot propagate as mixture is lean, which further leads to partial oxidation of fuel. Whereas in case of higher pilot quantity leads to faster flame propagation. **Therefore, the intake temperature and pilot quantity needs to be properly controlled to prevent failure in combustion.** For the same equivalence ratio for diesel engine, the amount of diesel quantity increases thus temperature as well as NOx increases.

Amey Parshuram Amate et al. [16]

Piston bowl Parameters:

I. Throat diameter: Distance between bowl piston edges (lip) near piston crown. The ratio of diameter of throat to maximum bowl diameter defines amount of re-entrance. Bowl lip is often the hottest part of piston bowl internal surface.

II. Maximum diameter: It is the largest diameter of the piston bowl. It is one of the main and the first parameter set while designing shape of new piston.

$$\text{Aspect ratio} = \frac{maximum\ bowl\ diameter}{bowl\ depth} \tag{7}$$

III. Central pip: If there is no central pip swirl that are generated leads to poor air/ fuel mixing. Central pip results in better mixing by maintain mean air flow velocity. It controls the activity of swirl and vortex.

IV. Bowl depth: Distance between piston crown to toroidal radius. It is closely linked with the spray impingement area. It also depends on piston pip.

V. Toroidal radius: Major combustion takes place in this toroidal radius volume. Lager toroidal radius helps in better air/ fuel mixing.

VI. Impingement area: It is the area where high velocity fuel impinges on the bowl. It affects the delay period, initial pressure rise rate. It divides the fuel injected into two parts: one below the lip and other one above the lip. First stage of combustion takes place in upper bowl and then in the lower bowl, this helps in reducing PM. This type of combustion generally takes place in stepped lip.

Dahlstrom, J. and Andersson [17] The ratio of surface of piston to it's the volume is an important parameter for heat transfer. The injection events and piston design should be such that the high tangential velocity is at the centre of bowl, rather near bowl. This prevent convective heat transfer to the outer bowl. All experiments were performed at 1500 rpm and at 10.5 bar IMEP, whereas speed load test is performed at 2000 rpm. For stepped piston suitable nozzle protrusion was decided as spray target is important for PM and emission. Injector tip protrusion of stepped is slightly below the lip so that maximum amount (70%) of fuel goes in the bowl, so nozzle protrusion has to be increased by 1mm. Speed and load are the input whereas output is rail pressure. High pressure reduces injection duration and thus combustion duration. Stepped type perform better than conventional because of reduced CO and PM because of better air utilisation. Soot emission decreases with larger nozzle protrusion but this also increases combustion pressure and increases NOx formation. The cylinder head is responsible for losing about 50% of total heat transfer to the cooling water. For stepped piston exhaust losses are much more at higher speed and load and also higher piston cooling losses. Head losses are same to base line engine. **No change in combustion phasing with different piston is observed.** The shorter combustion duration results in larger amount of heat release in short duration of time, thus combustion duration and heat release are inversely proportional to each other. Stepped piston leads to faster combustion as compared to re-entrance type. Short combustion duration also leads to higher temperature inside the chamber, which can lead to higher exhaust losses. With increase in the heat release rate the indicated power is slightly increased, simultaneously losses in the exhaust and slight increase in piston cooling losses however cylinder head losses is slightly reduced. Therefore the exhaust losses can be more beneficial as energy can be utilized to run turbo charger so as to recover exhaust energy. In stepped piston exhaust losses are more than baseline piston.

Felix Leach et al. [18] Stepped lip increases mass fuel burning by 3° crank angle as compared to re-entrance type. Comparison is done between two pistons with same compression ratio only one is re-entrance type and other is stepped type with no change in omega part. In stepped piston, it is observed that there is recirculation formed at the stepped region this reduces the flame velocity near squish region, this mechanism is also called ‘flame hold up’. At higher load there is high post flame oxidation at 4000 rpm therefore the PM at these load is low thus NOx seems to increase. ISFC for stepped piston is marginally low than in re-entrance type at higher load. Stepped piston seems to slow down the flame in the step region and prevent rapid movement of flame in the fire land, thus reduces soot formation in squish region. In the stepped piston the velocity at the lip is almost zero at 10° ATDC because of re-circulation of stepped lip. This portion helps in holding up or slowing down the diesel into fire land. This recirculation increases mass to diesel burn, slows down burn rate due to vortex formation and this also reduces ignition delay.

Fig. 16 Comparison between re- entrance and stepped type piston bowl shown above respectively [18]

Oivind Andersson et al. [19] Height squish region should be kept as small possibly between 0.6-0.8mm so as to improve air utilisation i.e. increase k-factor (ratio of piston bowl to clearance volume), this is beneficial for both CO, HC reduction. This however depends on the manufacturing tolerance. Too small squish height leads to increase in the heat transfer.

Swirl ratio can be given as $Rs = \omega/2\pi N$. Where Rs is swirl ratio, ω is angular velocity and N is rotational speed of the engine. The designed swirl ratio generally depends on no. of holes in injector nozzle, higher holes require lower cylinder head swirl ratio. Higher swirl may also results increase in HC and CO (due to flame quenching). Piston bowl diameter usually kept 60% of bore diameter. For full load operation wider bowl is beneficial because they reduce wall wetting issues. Thus they are beneficial for high pressure injector. Wider bowl also increases the tolerance of variation in the spray penetration. One of latest technique is to split the fuel in two portion by using stepped lip. The upper split is directed towards the head so that penetration in the squish portion is avoided results in lower HC and CO formation. This stepped lip reduces heat losses by improving surface to volume ratio.

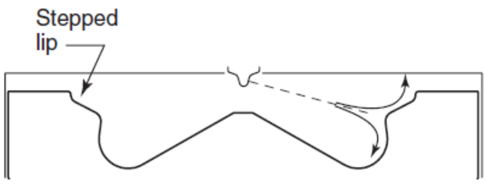


Fig. 17 Stepped lip bowl geometry [19]

Dockoon Yoo et al. [20] Final Emission Regulation without Diesel Particulate Filter (DPF) in order to improve the combustion efficiency concept of stepped lip has been introduced. This helps in utilising the air that is at the top of the piston. The upper portion forms an additional vortices. The spray position and the quantity of the fuel in top and the bottom main vortices depend on the lip position and amount of volume of fuel in the top portion and the bottom main combustion bowl portion. With help of this the k-factor is improved due to better air utilisation and the dead volume is reduced such as valve pocket volume. In the stepped piston, the low temperature zone (path at which fuel spray is split into two portions) is small, this results in smaller rich region present, thus lower soot formation. Soot emission has been reduced by 33% and NOx is increased by15%. NOx is further controlled by EGR control. The best spray angle is 142º for soot reduction. Tier-4 has been achieved by engine, final emission is measured without DPF and these piston will be available in the commercial market by 2014.

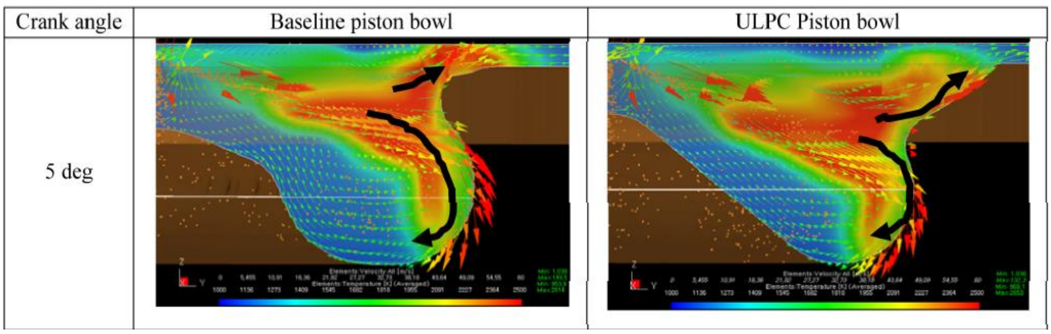


Fig. 18 Comparison of flow pattern in the baseline and ULPC piston [20]

Marc E. J. Stettler et al. [21] Dual fuel engine operates on two fuels blended inside the cylinder at the same time. Use of natural gas is further more cost effective, as it is cheaper than diesel and also the CO_2 quantity is also low in natural gas than as compared with diesel. Engine testing is done by European Transient Cycle and it is also been evaluated during transient cycle emission is reduced up to 9%. Engine speed and torque is systematically varied by 20Nm and 200rpm respectively. Then run on different modes that are urban, rural, motorway and combined the energy substitution ratio (ESR) is the portion of energy supplied by natural gas to engine.

$$\text{ESR [\%]} = \frac{LCV_{NG,IN} \cdot m_{NG,IN}}{LCV_{NG,IN} \cdot m_{NG,IN} + LCV_{Diesel,IN} \cdot m_{Diesel,IN}} \tag{8}$$

$LCV_{NG,\,IN}$ and $LCV_{Diesel,\,IN}$ is lower calorific value of natural gas and diesel respectively. $m_{NG,IN}$, $m_{Diesel,\,IN}$ is mass flow rate of natural gas and diesel. Emission are measured at tailpipe and post-turbo (engine out).

$$CH_4 \text{ slip} = \frac{m_{CH_{4,tailpipe}}}{m_{CH_{4,in}}} X\ 100 \tag{9}$$

The specific fuel consumption is maximum during urban phase than on motorway and rural phase for vehicles of all configuration. In dual fuel engine the energy efficiency is low as compared with baseline engine especially at lighter and medium load because of low combustion efficiency. The dual fuel system used in tractors leads to increase in tailpipe greenhouse gases emission compared with diesel engine by 50-127%, which is due to incomplete combustion of CH_4. The main reasons for this increase were:

1. Valve overlap
2. Incomplete combustion in region like crevice.
3. Incomplete combustion due to lean mixture which prevent flame propagation.

NOx of the dual fuel engine is comparatively lower than diesel engine, at the same time, CO emission is simultaneously increased. **It is also advised that low ESR at high speed could have a beneficial effect on CO_2 emission. However with this trade-off increases of NOx formation.** A method to reduce CH_4 is by direct injection of natural gas. Further reduction can be done by means of oxidation catalysts convertor which successfully oxidize CH_4 below 400ºC. As effectiveness of catalyst is at higher temperature at 400°C its performance is only 10- 15%. Since most of the previous studies have stated that dual fuel engine PM is reduced, however in this case **PM is increased at high loads (80%) with ESR in range of 30-50% which is possibly due to high rate of flow of diesel and natural gas quantity**. However PM was reduced by 50% at lower load which is due to tendency of premixed combustion of natural gas than in diffusion mode combustion. Higher exhaust temperature during the transient test is due to burning of unburnt HC.

The use of dual fuel technology can considerably effect SCR system working as N_2O emission in dual fuel mode is more than pure diesel run in following ways:

1. NO_2 produced due to high temperature are converted to N_2O due to high oxidation rate of ammonia. Also by direct ammonia above 350º C results in N_2O formation.
2. NH_4NO_3 (Ammonia nitride) decomposition results in N_2O formation.

Ghazi A. Karim [22] For the conversion of diesel engine to dual fuel engine retrofitted equipment should be avoided when the cost effective conversion is required. The cost depends on number of engines to be converted to dual fuel, cost of primary fuel available, effect on efficiency with the addition of primary fuel equipment's like fuel tank and additional equipment required to control the flow rate of fuel inside cylinder. Generally, it has been observed for dual fuel engine with gaseous fuel as the primary fuel cost conversion from diesel engine is less, as number of the design changes that are required are comparatively less. However variable cost for liquid and gaseous fuel in dual fuel engine may include fuel container, insurance and maintenance cost and also for gaseous fuel the cost of gas compression also has to be considered. However the main reason to use gaseous fuel is that it helps in cost saving because they are cheap and also more environmental friendly. The volumetric efficiency of the engine when the throttle is wide open determine the maximum power that can be achieved, as when gaseous fuel is injected in intake manifold some amount of air get displaced. Also the effect of cooling intake has to be taken into account. The conversion kit required for dual fuel engine include an ECU which continuously compute the amount and the timing of injection of diesel in dual fuel mode and also when engine is running in pure diesel. Sometimes retrofitted approach is not fully beneficial as it may take over the cost of benefits obtained by use of cheaper fuel, NOx and PM reduction. Cost saving can be further achieved removing large tank of primary fuel and parts of emission control equipment. Simpler engine design and control further add to reduce operating and capacitive cost.

John Heywood [2] Hiro Hiroyasu et al. [24] Spray penetration is a very important design parameter to decide piston bowl shape because in some cases spray impinges on the surface when walls are wet and during high swirl (spray penetration decreases with swirl), greater problem arises when there is wall wet at engine start. This leads in increase in PM, soot and HC. Also there should not be under penetration as this will cause poor air utilisation.

Spray penetration formula:

$$S = 3.07 \left(\frac{\Delta P}{\rho_g}\right)^{\frac{1}{4}} (td_n)^{\frac{1}{2}} \left(\frac{294}{T_g}\right)^{\frac{1}{4}} \qquad (10)$$

S = Spray penetration in meter

ΔP = Pressure drop across nozzle in Pascal

ρ_g = density of gas in kg/m^3 (15-25 kg/m^3)

t = time after start of injection in seconds.

d_n = nozzle diameter in meter

T_g = Temperature of hot gases in Kelvin (1000K)

Spray penetration increases with time at constant velocity.

$$t_{break} = \frac{29\rho_l d_n}{(\rho_l \Delta P)^{\frac{1}{2}}} \qquad \rho_l = \text{density of liquid in kg/m}^3 \tag{11}$$

$$S = 0.39\left(\frac{2\Delta P}{\rho_l}\right)^{\frac{1}{2}} t \qquad \text{for } (t< t_{break}) \tag{12}$$

$$S = 2.95\left(\frac{\Delta P}{\rho_l}\right)^{\frac{1}{4}} (td_n)^{\frac{1}{2}} \qquad \text{for } (t> t_{break}) \tag{13}$$

Spray penetration under the condition of swirl:

$$\frac{S_g}{S} = \left(1 + \frac{\pi R_s NS}{30 v_f}\right)^{-1} \tag{14}$$

S_g = Spray penetration with swirl in meter

R_s = Swirl ratio

N = Engine speed

v_f = Initial velocity of jet through injector nozzle

Mansor, W., and Nudriyan, [25] Presence of gaseous fuel reduces diffusion combustion which effect the rate of NOx formation. **NOx formation in dual fuel engine at higher load is more than diesel engine reason being in dual fuel mode the temperature at TDC remains higher for long duration than in case of diesel engine.** However due to larger ignition delay in dual fuel engine heat release rate of diesel engine is much better. Thermal efficiency depends on combustion duration and it directly dependents on in cylinder temperature. **In dual fuel engine with no throttle results in lean mixture at lower load which results in lower thermal efficiency, thus at lower load diesel quantity is increased.** Also considering flame wall quenching at lower load dual fuel engine works on pure diesel. For test plan ISO 8178 type D2 test cycle which involves five different loads corresponding to maximum load is used.

Sunmeet Singh Kalsi, and K.A. Subramanian [26] Experimental engine has power at rated point 1500 rpm of 7.4 kW. CNG is injected with timed manifold gas injection system. **EGR percentage vary from 8-30%. It has been found out that with 8% helps in reducing HC and CO, along with advantage of additional NOx reduction. However smoke is increased marginally**. Decrease in injection pressure reduces NOx emission. Over higher load the BTE reduces, which can be improved by improving combustion characteristics like pilot fuel quantity and advanced injection. Torque increases with increase in the pilot, which simultaneously increases inside temperature of combustion chamber results in increase in NOx formation. **Advanced timing of injection reduces torque formation and thermal efficiency has more of the combustion is taking place in the compression stroke**. EGR helps in improving performance of dual fuel engine by radical effect of exhaust gases. These **radical concentration helps in improving flame velocity** at lower and part load because at that time in- cylinder temperature is low and high specific heat of the charge (CNG %) . HC and CO reduces because of high temperature of exhaust gases. As the EGR is increased delay period increases, the peak rise rate and heat release rate decreases. Thermal efficiency increases with EGR. HC and CO can further reduced in by proper time setting so as to prevent the exhaust of CNG during intake during valve overlap. At part load only 15% EGR is used. Also one has to be careful with EGR percentage as higher value of it, increases the HC and CO at exhaust due to lack of oxygen present inside the chamber. Experiment output for part load (5 kW) are: NOx emission is 3.89 g/kWh, with 85% CNG it decreases to 1.48 g/kWh. For 30% of EGR NOx value is 1g/kWh at almost all load.

Junghwan Kim et al. [27] Two different stepped or lip piston were made one with single step other with two steps. Performance of these piston is compared with the re-entrance piston. By using the KIVA software it was evaluated that the air utilisation was increased with the use of double step piston. It gave wider fuel distribution inside the bowl and thus enhance combustion.

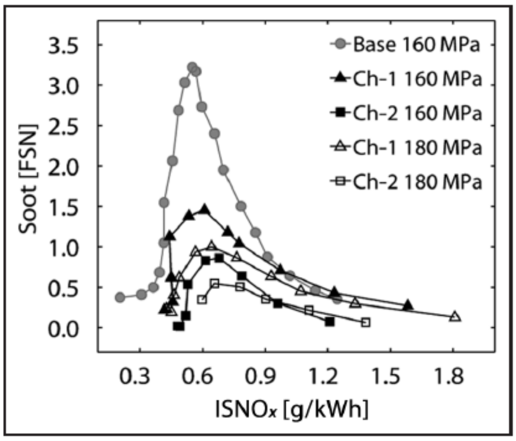


Fig. 19 Comparison of NOx and PM trade-Off for different piston bowl [27]

In figure 19 Ch-1 is single stepped piston, Ch-2 is double stepped piston and base is the re-entrance bowl piston bowl with the corresponding injection pressure. From figure 19, it is observed that the bump is the highest in case of base piston bowl and further bump in case of double lip is lower than in case of single lip because of better air utilisation and atomisation. Charge is more uniform in case of Ch-2 due to wider distribution as said above. All three pistons exhibit same level of NOx emission.

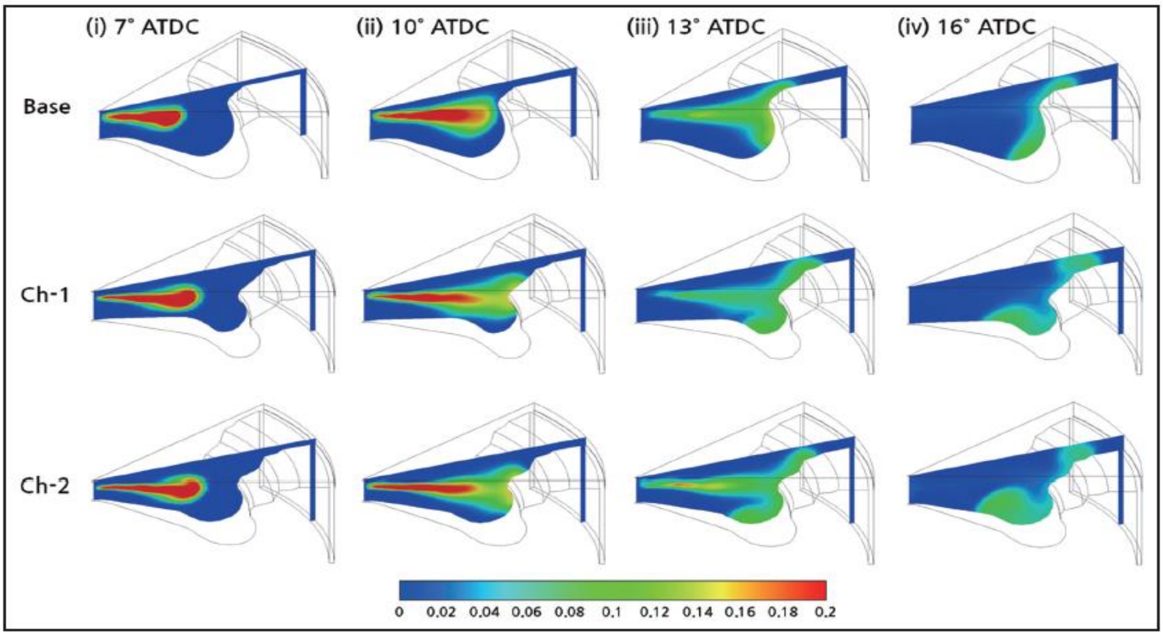


Fig. 20 Fuel charge distribution [27]

Due to better utilisation of air, in cylinder temperature increases. This higher temperature results in combustion timing, this helps in advancement in the injection which helps in reducing soot and higher thermal efficiency.

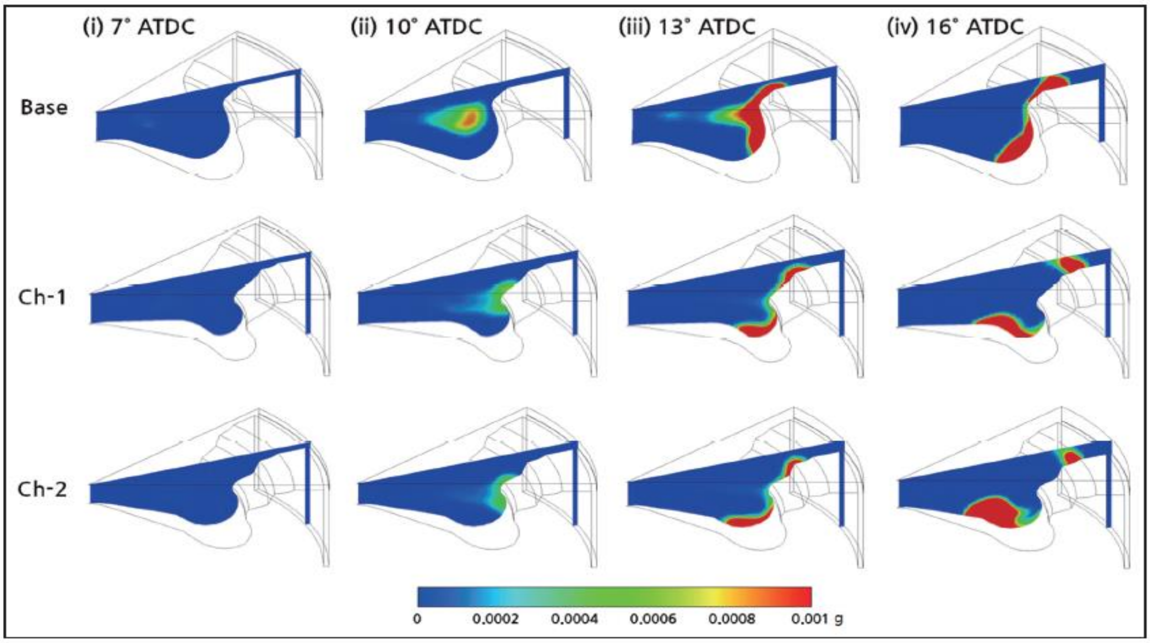
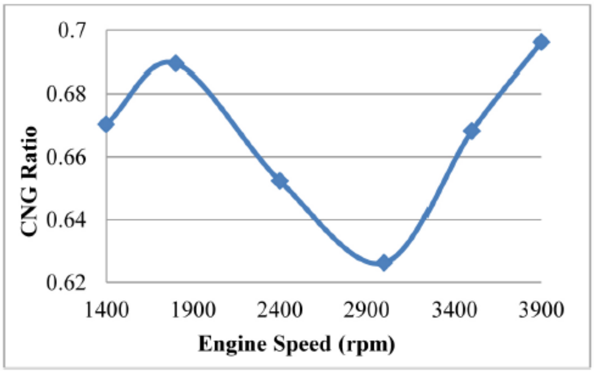


Fig. 21 Soot distribution in bowl for different piston [27]

V.P. Chougule, and K.C. Vora [29] GT-Suite has been used for examining the BSFC. For experimental work 59 kW diesel engine with compression ratio of 18.5 has been converted to dual fuel engine. **A closed loop lambda sensor is used which helps in improving performance with lean mixture**. Initially base line engine is modelled, results of which are compared with experimental results. Then dual fuel engine model is built using GT-POWER. For testing dual fuel engine a test matrix is prepared with different engine speed and at three different loads (50, 70 and 100%). At all the given test conditions, engine is first run on pure diesel operation then natural gas is injected, which results in increase of load at the same time diesel injection quantity is reduced till the initial conditions are reached. **Gas mixture venturi is used for mixing CNG in intake manifold.** In dual fuel engine, injection timing is advanced in order to reduce chances of premixed knocking. Retarding the injection timing moves the peak curve away from TDC, this reduces in cylinder temperature to such an extent that the flame cannot propagate throughout the combustion chamber therefore injection timing is advanced. From simulation it is observed that the BSFC is increased as compared with diesel engine. However, due to low cost of CNG, at higher speed fuel economy is improved by 8%.



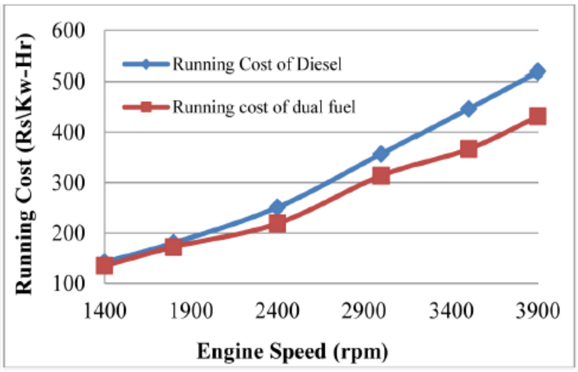
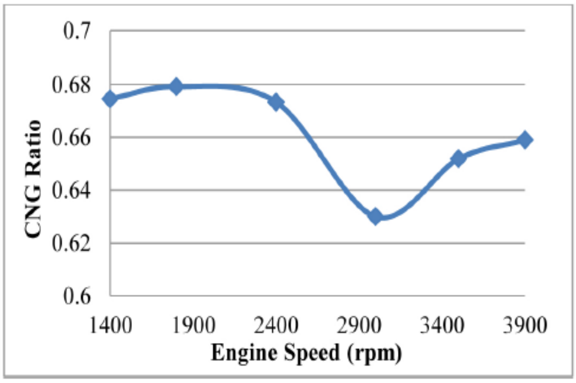


Fig. 22 CNG ratio vs. rpm at 50% load. Fig. 23 Running cost on diesel vs. dual fuel at 50% load



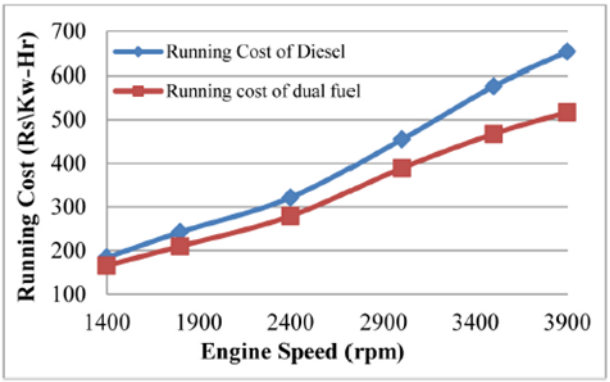


Fig. 24 CNG ratio vs. rpm at 70% load. Fig. 25 Running cost of diesel vs. dual fuel at 70% load

Brake specific fuel consumption is increased by 13% and 9% for 50% and 70% loads, respectively, keeping the torque same.

Sohan Lal, and S.K. Mohapatra [30] In this study biomass waste is the secondary fuel in CI diesel engine. Producer gas is formed from sawdust and cotton stalks which is the secondary fuel in this paper. Dual fuel engine have higher peak pressure and ignition delay as compared with diesel engine. With higher compression ratio ignition delay decreased due to higher pressure and temperature. Not only does the peak pressure increases but it also get shifted toward the expansion stroke, lesser the compression ratio more will be the shift. At higher compression ratio the consumption of gaseous fuel i.e. combustion efficiency also increases because higher compression ratio helps in getting higher temperature and more complete in combustion. In experimentation the sound level range of 80.9-89.6dB (A) was observed for dual fuel operation whereas diesel engine operate in the range of 81-89dB (A) at 3.2kW brake power. **Also NOx emission was reduced by 35.26-56.09% in case of dual fuel engine as in case of diesel engine, this is because of less intense premixed combustion**, less temperature produced due to producer gas and low concentration of oxygen. CO emission is about 81-84% higher in case of dual fuel engine than diesel engine at 3.2kW. HC emission has also increased by 63.41-74.04%. The SOx emission in dual fuel run is lower than in diesel mode because of lower sulphur content in biomass (0.01%) than in diesel (0.05%).

P. Rosha, A. Dhir, and S.K. Mohapatra [31] Dual fuel engine main objective is to minimize the consumption of fossil fuel by using renewable fuel in place of it. This paper includes the effect of using hydrogen, biogas and syngas as secondary fuel in dual fuel engine. When Hydrogen is used as secondary fuel combustion characteristics are high flame speed, low ignition energy and high energy density. Due to this hydrogen as secondary fuel has the problem of premature ignition. High speed flame propagation in hydrogen surely increases the BTE. It also leads to quicker combustion rapid increase in the pressure. Increase in the hydrogen thus increases the NOx formation. Biogas as secondary fuel is cheaper and easily available. Energy content in biogas depends on the quantity of methane present. With biogas as secondary fuel can however cause carbon decomposition inside cylinder. CO_2 present in the biogas results in slowing down flame propagation thus the BTE reduces. Beyond 40% of CO_2 in biogas surely effect dual fuel engine performance. When using biogas that the peak pressure under all compression ratio for dual fuel engine was lower than in diesel engine under all given compression ratio, also the exhaust gas temperature is much higher the reason begging the late combustion of biogas. When using Syngas (H_2 and CO) it provides advantages like high self-ignition temperature and low CO emission. **It is observed that NOx emission reduces, with increase in CO as the flame propagation reduces.** The use of syngas in dual fuel engine helps in reducing diesel consumption and also improves combustion quality. This is done by correct proportion of CO and H2.

Ryskamp, R., and Thompson et al. [32] RCCI is a technique in which fuels of entirely different reactivity are used to control the combustion phasing. Nine different fuels were taken with varying cetane number, aromatic content and distillation temperatures. Cetane number lower than 33 results in higher NOx formation compared to 44 to 54. For controlling combustion phasing of fuel with higher cetane number with lower aromatic content need larger percentage of CNG. Higher cetane number increases the ability of engine to take higher load, it also reduces ignition delay. Control of CA50 depends on ratio of high and low reactive fuel. Retard in CA50 is obtained by increasing in CNG quantity. **Increase in cetane number of high reactive fuel reduces NOx formation in RCCI technique as pressure rise rate is reduced in case of fuel with high cetane, also this increase load carrying range.** However, soot emission was slightly increased with use of medium and high cetane fuel (soot increases for the fuel having reactivity higher than T10).

3.1 Research Gaps:

1. The small four wheeler passenger and commercial vehicle segment forms a very important first-and-last mile travel mode of transportation in Indian cities and towns due to kind of road network available.
2. The already proven technologies of CRDI, DPF, and SCR are very expensive for this cost sensitive market.
3. There can be a cost effective solution to meet forthcoming emission norms with fuel efficiency benefits.
4. A number of projects using RCCI has been used for stationary, marine, larger multi-cylinder engines for buses, trucks etc. Hardly any work for automotive application with single cylinder engine is known.

3.2 Conclusion:

Studying the design of existing engine it is evaluated that CNG direct injection is difficult due to packaging constraints in the cylinder head, it may also requires change of piston bowl offset and development cost involving would be too high for up gradation to RCCI engine. Therefore, it is decided that CNG has to be port injected. CNG fumigation in the intake manifold decreases volumetric efficiency and its effect has to be considered. Spray penetration in RCCI engine would increase because of use of CNG which is lighter than air. Compression ratio of existing engine also has to be reduced in order to prevent expected knocking. In order to decrease number of experiments simulation is must. In the company simulation is performed with the help of GT-Power for one dimensional simulation. For accuracy of results initially baseline engines will be simulated, then results will be verified with experimental work. Initial simulation models will be modified for RCCI engine. From this simulation we will get the value of performance and emissions. For RCCI engine main critical part is piston. It is always required during design of piston to keep crevice volume and surface area minimum so as to increase the thermal efficiency. From the experimental work done in literature, it is concluded that HC and CO in case of RCCI engine is high as compared with diesel engine; therefore use of EGR would help in reducing these emission value. Also, according to literature survey, dual fuel engine is to be run on pure diesel operation at lower load due to lower thermal/ combustion efficiency in case of dual fuel run and also to retain good starting characteristics of diesel engine.

4.0 Engine Specification (Baseline engine):

Displacement	650cc (Single cylinder)
Application	Compact 4-wheeler
Cooling	Water cooled
Bore	94 mm
Stroke	94 mm
Compression ratio	19.2:1 (re-entrance type piston bowl)
Rated power	10.8kW @ 3200rpm
Max. torque	38Nm @ 1600 rpm
Max. combustion pressure	75-80 bar
Injector	Solenoid type with operating range 250- 1100 bar

5.0 Strategy of conducting project work:

1. To study the concept of RCCI engines and its past practices (literature survey).
2. To study the existing engine (650cc) and to evaluate the parts that are to be redesigned to convert existing engine to RCCI engine.
3. Simulation using GT-Power, of the baseline engines (Diesel and CNG) and verify results with their experimental steady state performance.
4. Manufacturing and assembly of parts (RCCI).
5. Predicting performance of RCCI engine by combining diesel and CNG in GT-POWER.
6. Final testing of dual fuel engine for validating the model.

6.0 Design up gradation

6.1 Intake manifold design

Intake manifold is connector that essentially connects the environment with the engine. It is required to design the intake manifold in such a way that flow of the air converge while flowing from air filter to cylinder head, this helps in reducing unnecessary loss of expansion or eddy formation. A new Intake manifold has been designed in order to accommodate CNG injector, TMAP sensor, EGR port and oil mist port.

The sequential order of arrangement from air filter to engine is given as follows:

- Oil mist port
- EGR port
- TMAP sensor
- CNG injector mounting bracket

As in baseline engine the EGR port is near cylinder head. It is recommended to shift EGR port position away from the cylinder head due to two main reason

1. In RCCI engine, this port has been shifted near air filter so as to get more homogenous air and exhaust gases mixture.
2. There are chances of CNG gas flow to pass EGR port when located near cylinder head. Increase in distance certainly reduces the chances of CNG gas passing through EGR port.

CNG injector is placed in such a way that the spray of CNG gas is directly above the intake valve and TMAP sensor is accommodated in order to control the pressure and quantity of CNG gas.

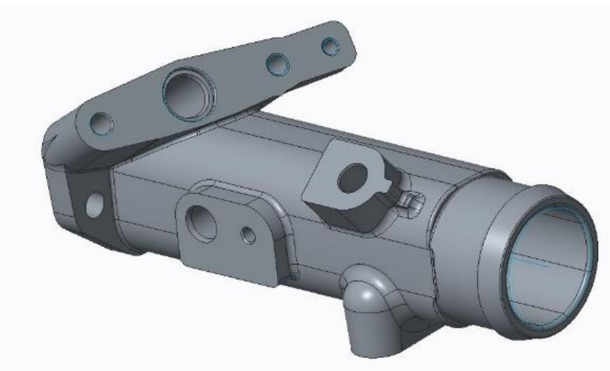


Fig. 26 Designed intake manifold

6.2 Piston bowl design

Initial compression ratio calculation for diesel engine is given as follows:

S.no.	Volume	Units	Mean value	Max. dim.	Min. dim.
1	Bore	cm	9.4	9.401	9.4
2	Stroke	cm	9.4		
3	Diameter of intake valve	cm	4.25	4.26	4.24
4	Top land	cm	1.05	1.07	1.03
5	Diameter of exhaust valve	cm	3.78	3.79	3.77
6	Protrusion vol. of intake	cm^3	0.56	0.71	0.42
7	Thickness of valve protrusion	cm	0.04	0.05	0.03

8	Protrusion vol. of exhaust	cm^3	0.45	0.56	0.33
9	Piston top land dia.	cm	9.33	9.32	9.32
10	Cyl. head gasket inner dia.	cm	9.5		
11	Bumping clearance	cm	0.09	0.1	0.08
12	Swept volume (Vs)	cm^3	652.00	652.21	652.00
13	Bowl volume (Vb)	cm^3	26.9	27.2	26.6
14	Bumping clearance volume	cm^3	6.37	7.08	5.66
15	Top land volume	cm^3	1.11	1.13	1.11
16	Intake valve pocket volume	cm^3	2.3	2.38	2.22
17	Exhaust valve pocket volume	cm^3			
18	Total protrusion volume	cm^3	1.01	1.27	0.75
19	Injector protrusion	cm^3	0.0043		
20	Total dead volume (Vc)	cm^3	35.66	36.52	34.83
21	C.R.		19.28	18.85	19.71
22	k-factor (Vb/Vc)		0.75	0.74	0.76
23	Bowl surface area	cm^2	48.29		

Table 2 Compression ratio calculation of existing diesel engine

Existing piston is re-entrance type with k-factor (Bowl volume/Total clearance volume) of 0.75. The modified piston (Stepped type) has compression ratio of 17.11:1. With its bowl depth/bowl diameter ratio remained same. It is essential to make piston shallow so as to get better mixing and prevent chances of wall hitting as CNG has lighter density than air so spray penetration in dual fuel engine is certainly more than diesel engine. Following is the calculation of 17.11:1 CR calculation.

S.no.	Volume	Units	Mean value	Max. dim	Min. dim
1	Bore	cm	9.4	9.401	9.4
2	Stroke	cm	9.4		
3	Diameter of intake valve	cm	4.25	4.26	4.24
4	Top land	cm	1.05	1.07	1.03
5	Diameter of exhaust valve	cm	3.78	3.79	3.77
6	Protrusion vol. of intake	cm^3	0.56	0.71	0.42
7	Thickness of valve protrusion	cm	0.04	0.05	0.03
8	Protrusion vol. of exhaust	cm^3	0.44	0.56	0.33
9	Piston top land dia.	cm	9.32	9.32	9.32
10	Cyl. head gasket inner dia.	cm	9.5		

11	Bumping clearance	cm	0.09	0.1	0.08
12	Swept volume (Vs)	cm^3	652.0	652.2	652.0
13	Bowl volume (V_b)	cm^3	32.76	33.06	32.46
14	Bumping clearance volume	cm^3	6.37	7.08	5.66
15	Top land volume	cm^3	1.11	1.13	1.11
18	Total protrusion volume	cm^3	1.015	1.27	0.75
19	Injector protrusion	cm^3	0.0043		
20	Total dead volume (Vc)	cm^3	40.46	41.23	39.71
21	C.R.		17.11	16.81	17.41
22	k-factor (Vb/Vc)		0.81	0.826	0.84
23	Bowl surface area	cm^2	52.32		

Table 3 Compression ratio calculation for RCCI engine

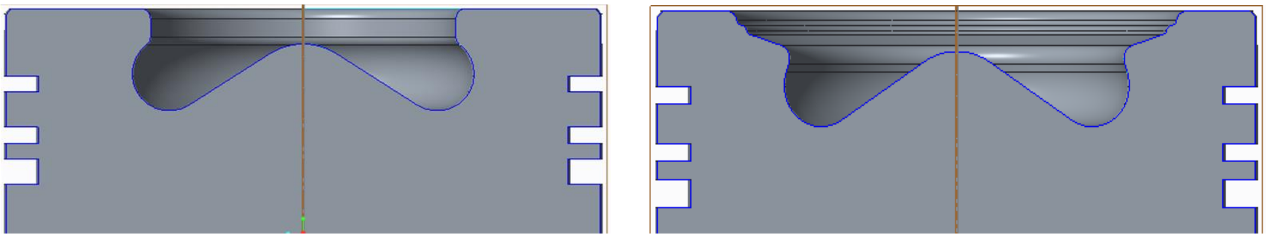


Fig. 27 Existing and modified piston bowl geometrty

Fig. 28 Manufactured modified piston CR 17.11:1

With the use of modified piston (stepped type), k-fator was increased by 8 % as the valve pocket volume is reduced, this helps in better air utilisation inside piston because bowl volume has increased.

7.0 Simulation (GT-POWER)

Simulation in the GT includes following steps:

1. Benchmarking of existing diesel and CNG (650cc) engine. Predicting performance and emission and verify its results within 10% of deviation from their experimental value.
2. Predicting performance of model with CR 17.11:1 on pure diesel operation and verifing its results with experimental data.
3. Finally building RCCI model by combining CNG and diesel model, predicting its performance and comparing it with benchmarked diesel engine.

7.1 Benchmarking of existing diesel engine (650cc)

Existing engine is model using direct injection diesel wiebe model. Burn rate profile in this model is predicted by using three Wiebe functions [33] which are as follows:

1. Premix duration function
2. Main duration function
3. Tail duration function

The Wiebe equations are given by following equation:

$$\text{Wiebe premix} = \left[\frac{\mathrm{D_P}}{(2.302)^{\frac{1}{(\mathrm{E_P}+1)}} - (0.105)^{\frac{1}{(\mathrm{E_P}+1)}}}\right]^{-(\mathrm{E_P}+1)} \tag{15}$$

$$\text{Wiebe main} = \left[\frac{\mathrm{D_M}}{(2.302)^{\frac{1}{(\mathrm{E_M}+1)}} - (0.105)^{\frac{1}{(\mathrm{E_M}+1)}}}\right]^{-(\mathrm{E_M}+1)} \tag{16}$$

$$\text{Wiebe tail} = \left[\frac{\mathrm{D_T}}{(2.302)^{\frac{1}{(\mathrm{E_T}+1)}} - (0.105)^{\frac{1}{(\mathrm{E_T}+1)}}}\right]^{-(\mathrm{E_T}+1)} \tag{17}$$

$$F_M = (1 - F_P - F_T) \tag{18}$$

D_P = Premix duration

D_M = Main duration

D_T = Tail duration

E_P = Premix exponent

E_M = Main exponent

E_T = Tail exponent

F_M = Main fraction

F_P = Premix fraction

F_T = Tail fraction

The main purpose of these equations is to model premixed, main and diffusion portions of combustion process curve. Final diesel model building is done by following steps:

1. Initially all the input data regarding fuel injector (injection quantity, SOI, injection duration, fuel properties, etc.), inlet and exhaust valves timing, port forward and reverse discharge coefficients, FMEP, heat transfer model, intake and exhaust manifold dimensions and finally experimental pressure crank angle value at all given full throttle performance points are collected of existing engines.
2. Build TPA (Three pressure analysis) model to predict the burn rate profiles from the given experimental pressure crank angle values. Pressure profile with the filter of IEEE-54 is used in order to provide more smoothening to the experiment pressure curve. However TPA model lacks the ability of predicting emission, therefore DI Wiebe model is built in separate new file.
3. Various input constants in DI Wiebe combustion model are as follows:

Ignition delay	Tail fraction
Ignition delay multiplier	Tail duration
Premixed fraction	Tail duration multiplier
Premixed duration	Premixed duration exponent
Premixed duration multiplier	Main duration exponent
Main duration	Tail duration exponent
Main duration multiplier	Combustion efficiency

The above parameter are set to such as a value that the combustion profile (Cumulative burn rate/ heat release rate) in DI Wiebe matches with the combustion profile that is predicted in the GT TPA model. In TPA combustion profile is plotted from pressure vs. crank angle graph given by INDICOM software (Engine testing software).

4. Prediction of performance by TPA and DI Wiebe model are within 10% of error w.r.t. to experimental data.

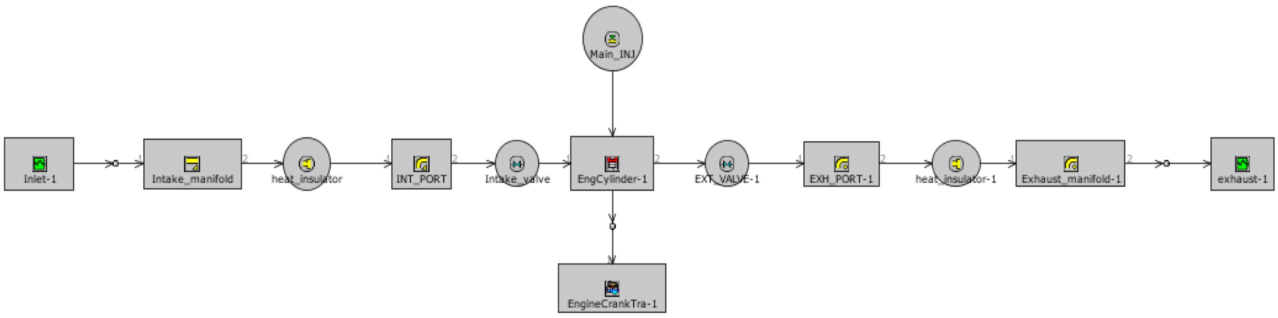


Fig. 29 TPA model of benchmark engine for prediction of burn rate profile from experimental pressure crank angle value

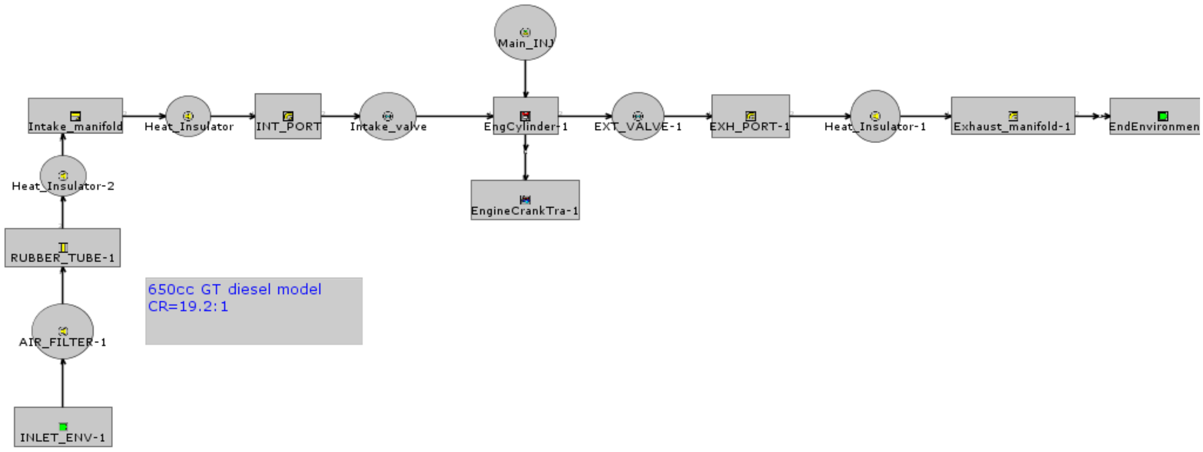


Fig. 30 DI Wiebe model for prediction of performance and emission of existing engine

The predicted and experimental pressure crank angle at different full throttle performance points are given as follows:

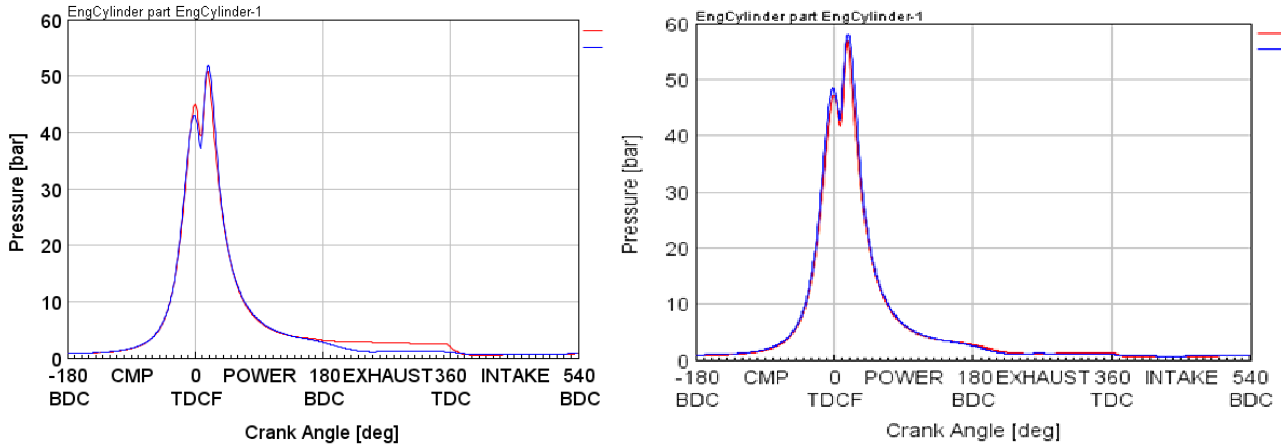


Fig. 31 Pressure vs. Crank angle at 3200 (rated power point) and 2400 rpm respectively.

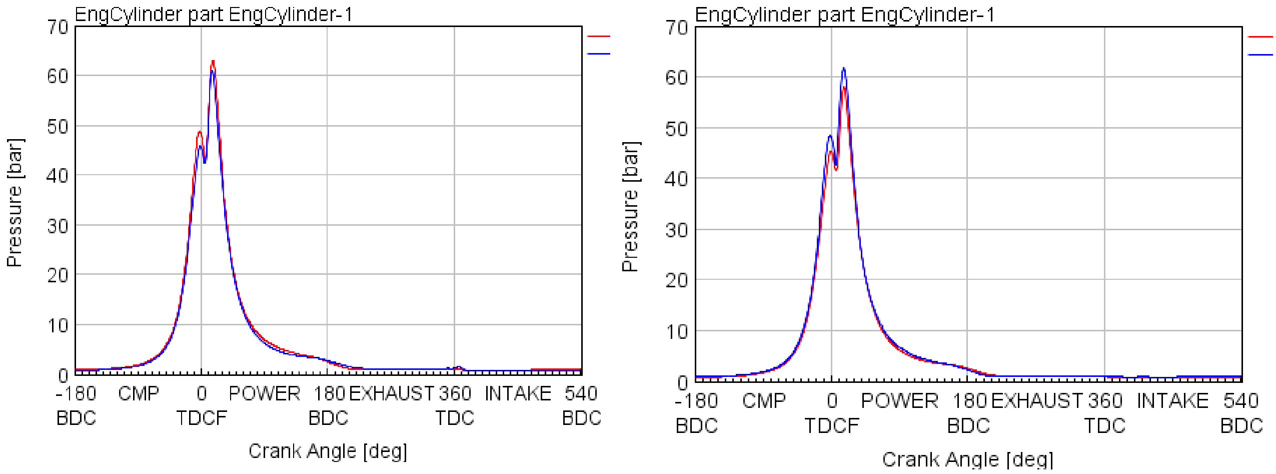


Fig. 32 Pressure vs. Crank angle at 1600 (Max. torque) and 1400 rpm respectively.

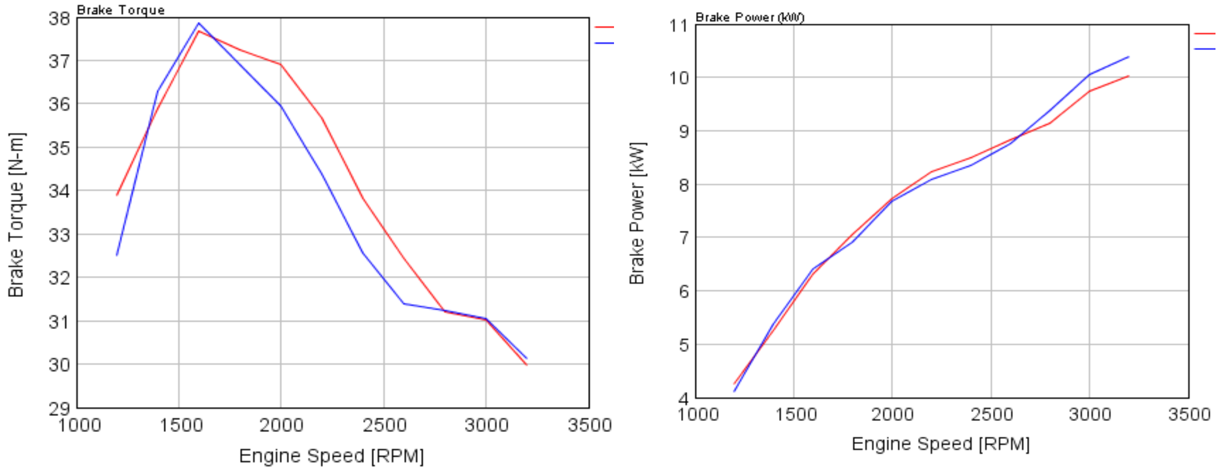
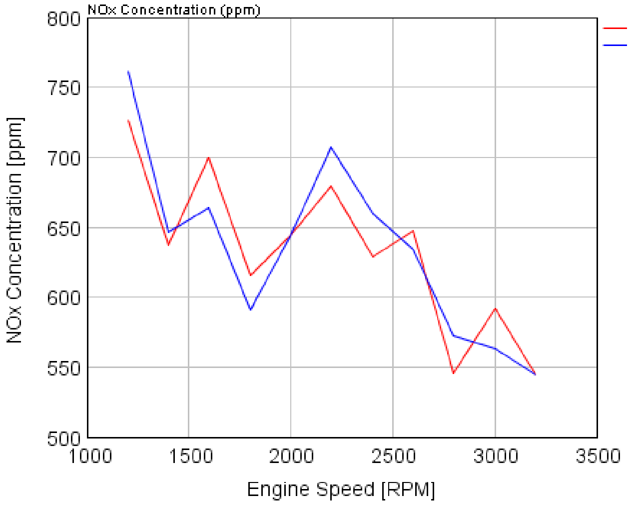


Fig. 33 Predicted and Experimental torque and power comparison respectively.



Simulated

Experimental

Fig. 34 Predicted and Experimental NOx comparison respectively.

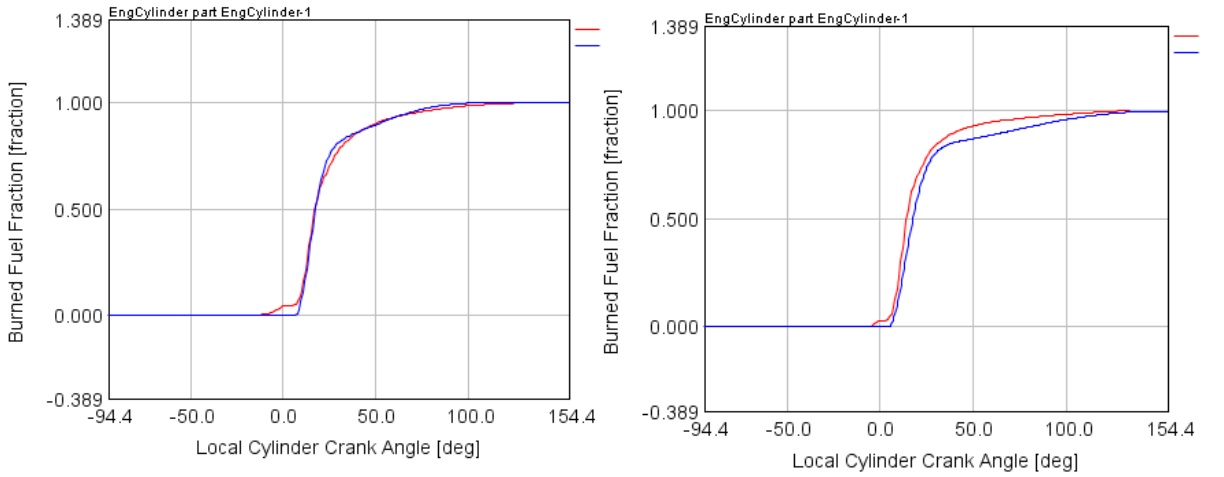


Fig. 35 Cumulative burn fraction at 3200 rpm (Rated power point) and 1600 rpm (Max. torque point) respectively.

TPA Experimental

Simulation

Following are the main assumption made during model building:

1. FMEP (Friction Mean Effective Pressure) - For calculation of FMEP Chen-Flynn engine friction model is used. Engine friction is calculated by following equation:

   $$\text{FMEP}=\text{FMEP}_{\text{Const}} + \left[A \times P_{\text{Cyl.max}}\right] + \left[B \times c_{\text{mean}}\right] + \left[C \times {c_{\text{mean}}}^{2}\right] \quad (19)$$

   $\text{FMEP}_{\text{Const}} = \text{Constant part of FMEP in bar (0.4)}$

   $P_{\text{Cyl.max}} = \text{Max. cylinder pressure}$

   $c_{\text{mean}} = \text{Mean piston speed}$

   $A = \text{Peak pressure factor (0.005)}$

   $B = \text{Mean piston speed factor (0.09)}$

   $C = \text{Mean piston speed factor (0.0008)}$
2. Cylinder wall solver model has been used with head coolant temperature of 121°C, piston oil temperature of 131°C, cylinder coolant and cylinder oil temperature 98°C. Head coolant heat transfer coefficient of 5500W/m^2-K. Piston oil heat transfer coefficient 200W/m^2-K. Cylinder oil heat transfer coefficient 500W/m^2-K. Cylinder coolant heat transfer coefficient 9500W/m^2-K.
3. Valve lash is said to be constant throughout all rpm.

The percentage of error for the performance parameters are kept within 10%. For torque, the maximum percentage of error is 4.8 % at 1200 rpm. For power curve, it is 3.4% at the same point. However, for NOx prediction, the percentage of error is kept under 20% as there are always some percentage error in exhaust measured data.

7.2 Benchmarking of existing CNG engine (650cc)

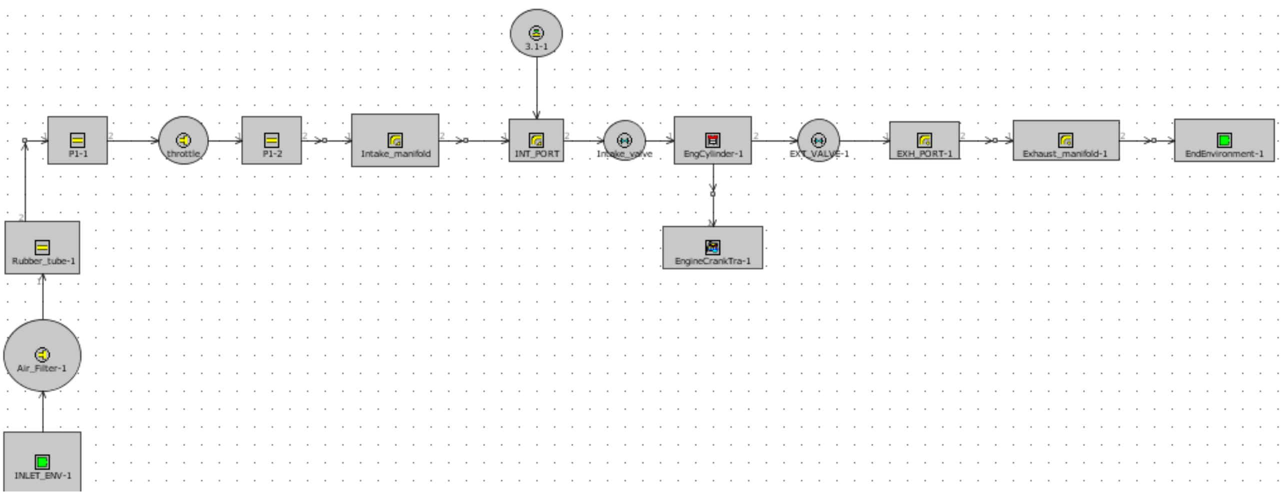


Fig. 36 Benchmarked CNG GT-POWER model building

Combustion model that is used for spark ignition engine is EngCylCombSIWiebe. Various inputs in SI Weibe equation [33] are as follows:

AA = Anchor Angle

D = Duration

E = Wiebe Exponent (default = 2.0)

CE = Combustion Efficiency

BM = Burned Fuel Percentage at Anchor Angle (default = 50%)

BS = Burned Fuel Percentage at Duration Start (default = 10%)

BE = Burned Fuel Percentage at Duration End (default = 90%)

θ = Instantaneous crank angle

Calculated constants:

Burned Midpoint Constant (BMC) = -ln (1-BM) (20)

Burned Start Constant (BSC) = -ln (1-BS) (21)

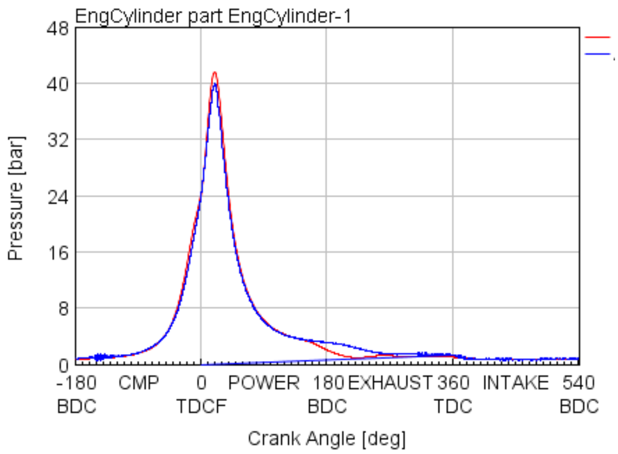
Burned End Constant (BEC) = -ln (1-BE) (22)

$$\text{WC (Wiebe constant)} = \left[\frac{D}{BEC^{1/(E+1)} - BSC^{1/(E+1)}}\right]^{-(E+1)} \tag{23}$$

$$\text{SOC (Start of combustion)} = \text{AA-}\left[\frac{(D)(BMC)^{1/(E+1)}}{BEC^{1/(E+1)} - BSC^{1/(E+1)}}\right] \tag{24}$$

$$\text{Combustion } (\theta) = (\text{CE})\left[1 - e^{-(WC)(\theta - SOC)^{(E+1)}}\right] \tag{25}$$

Following graphs show pressure vs. crank angle, experimental and predicted, for CNG at FTP points:



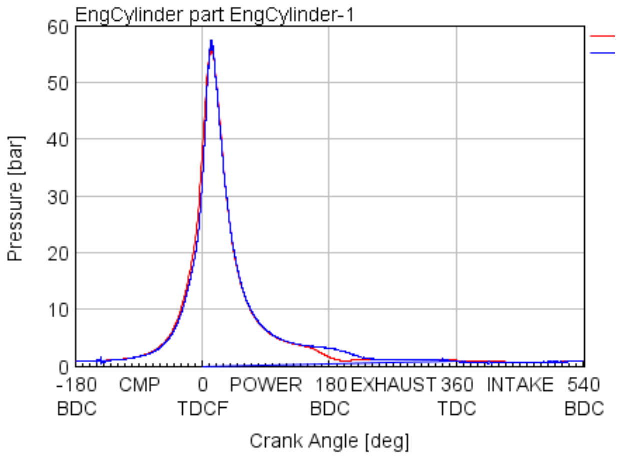


Fig. 37 Pressure vs. Crank angle at 3100 and 1800 rpm respectively.

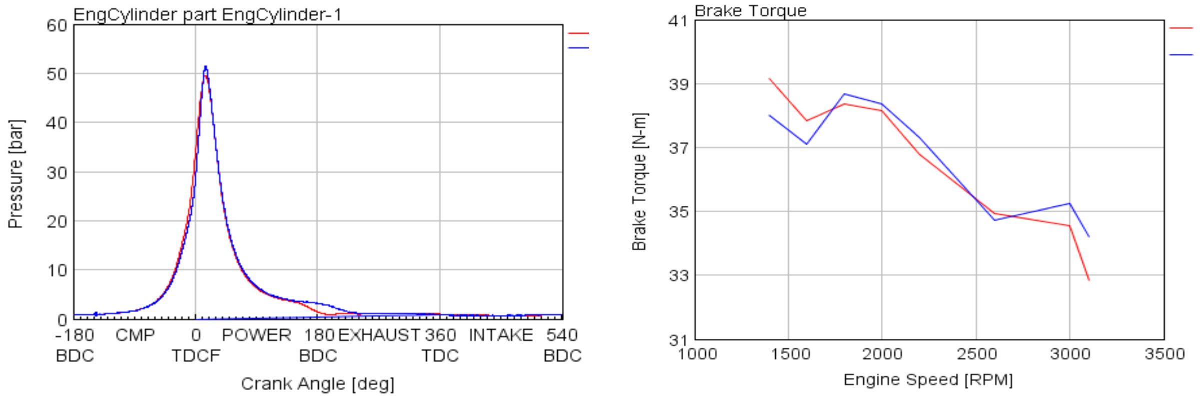


Fig. 38 Pressure vs. crank angle at 1600 rpm, predicted and experimental torque comparison

Simulated

Experimental

For CNG model, the FTP points at which input data available are:

3100, 3000, 2600, 2200, 2000, 1800, 1600 and 1400 rpm. Maximum percentage of error in torque curve is at 3100 rpm of 3.95%. Engine is operated at 17.8:1 air fuel ratio (stoichiometric ratio). Following shows the composition of CNG by mass percentage (mole percentage) as an input in GT-POWER fluid properties:

Methane = 93.24	Propane = 0.091
Ethane = 6.523	n-Pentane = 0.021
Nitrogen = 0.109	n-Butane = 0.016

7.3 Diesel engine performance predicted with CR 17.11.

With modified piston of compression ratio 17.11:1, following is the effect on performance and emission.

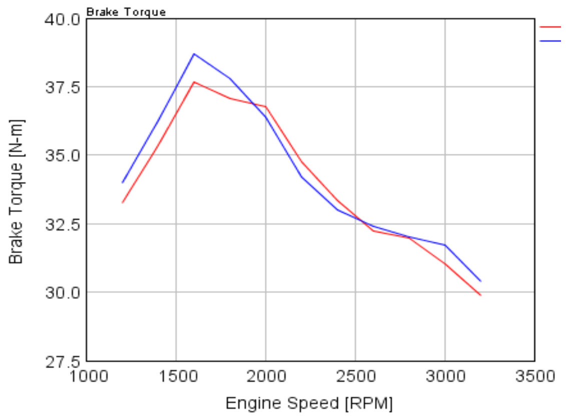


Fig. 39 Torque variation with CR

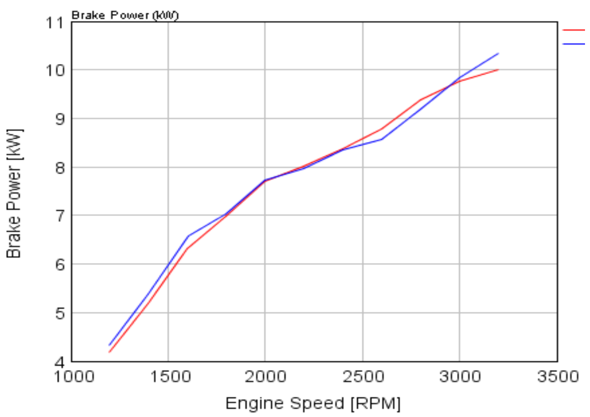


Fig. 40 Power variation with CR

Simulated

Experimental

Max. percentage of error in torque and power is observed to be 2.59% and 3.93%, respectively, at 1600 rpm. With the reduction in the compression ratio, it was observed that exhaust temperature has reached to its alarming point this may be because of two main reason that are:

1. Reduction in compression ratio has reduced the thermal efficiency therefore energy of fuel extracted inside the cylinder has reduced and exhaust losses has increased.
2. At higher rpm, usually the time of burning the fuel in cylinder reduces, due to this fuel continue to burn during the exhaust stroke, which results in increase of exhaust temperature.

Considering the exhaust losses, the dynamic injection timing (DIT) has been advanced by 4º in experimental work. This advancing is done at all the points of DIT base maps in INCA software.

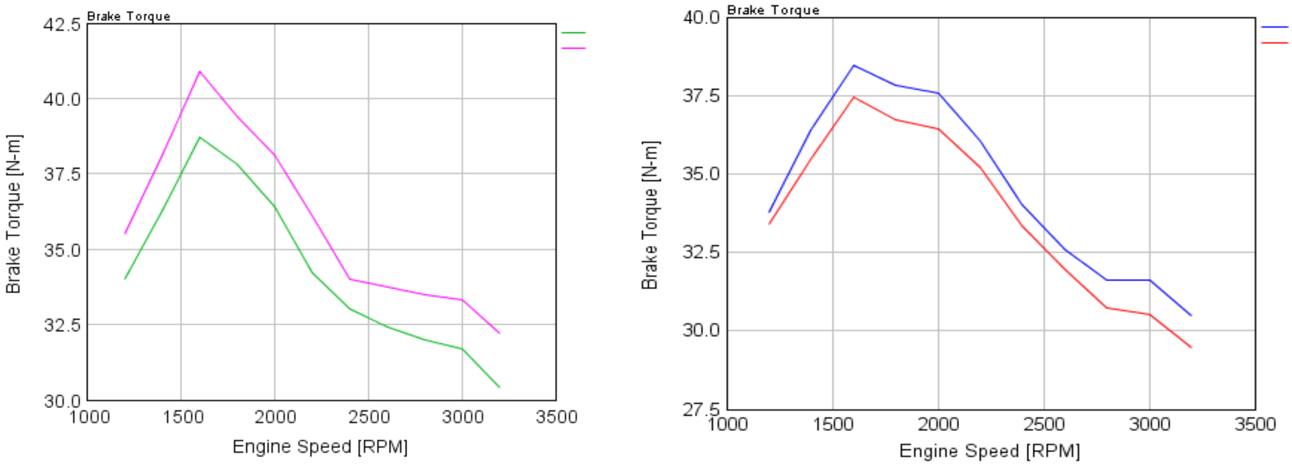


Fig. 41 Experimental and simulation torque comparison with and without DIT variation.

Experimental work shows 4.7% average whereas simulation showed 3.68% average increase in torque with advancing.

Torque with advanced DIT (Experimental)

Initial torque (Experimental)

Torque with advanced DIT (Simulated)

Initial torque (Simulated)

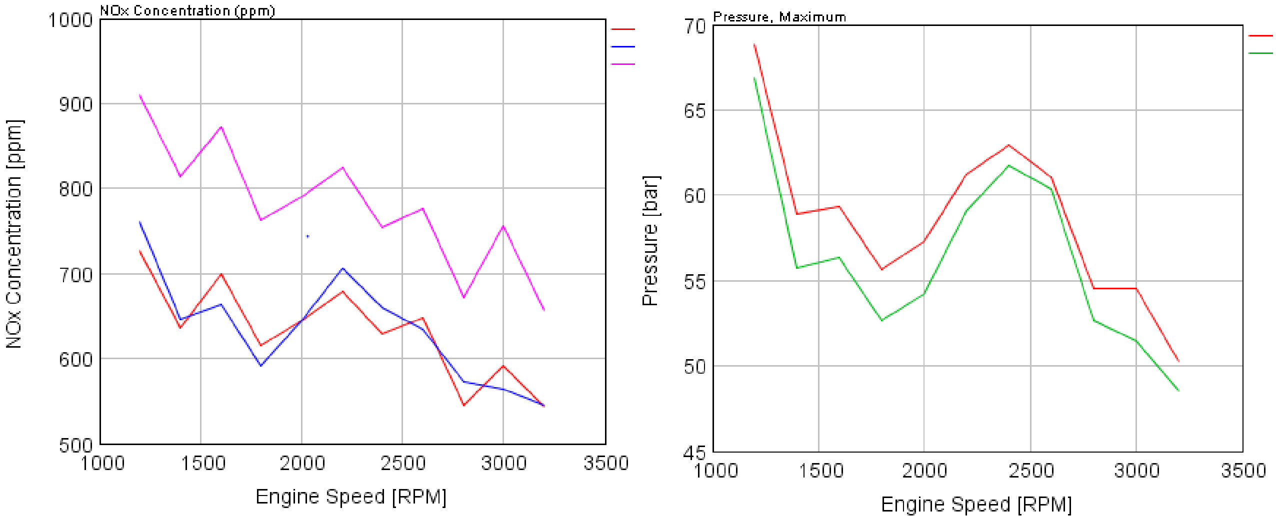


Fig. 42 with advancing the DIT by 4°

NOx has increased by 23% average overall and rise in peak pressure is observed.

Predicted NOx for 19.2 CR

Predicted NOx for 17.1 CR with 4° advanced DIT

Experimental NOx for 19.2 CR

Peak pressure with CR 19.2:1

Peak pressure with CR 17.11 with 4° of advance

7.4 Dual fuel engine model

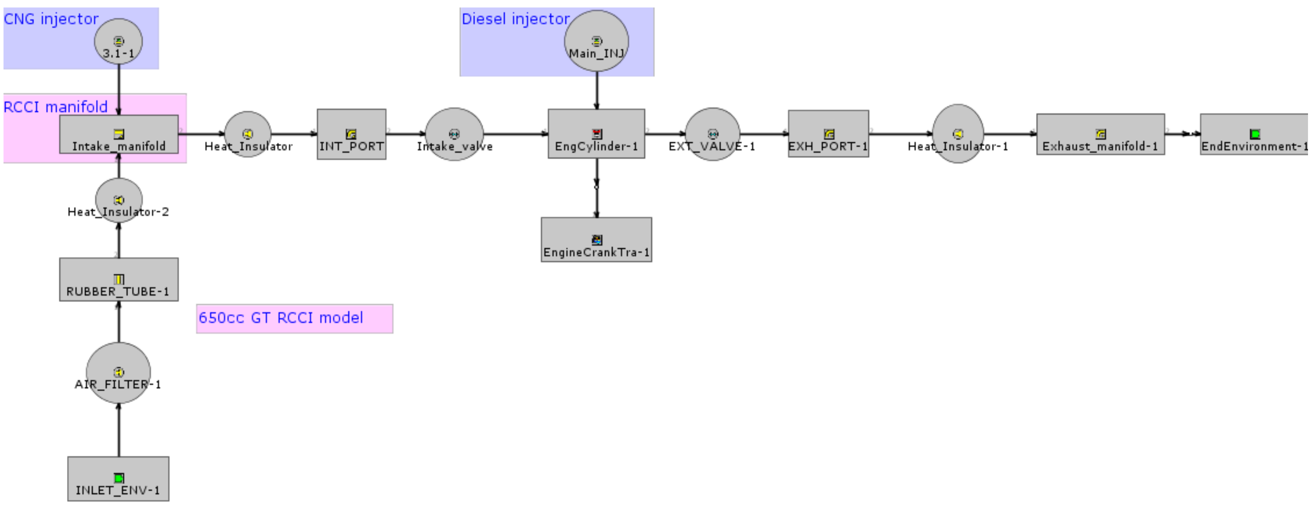


Fig. 43 RCCI GT-POWER simulation model

7.4.1 GSR (Gas Supplement Ratio) 20%

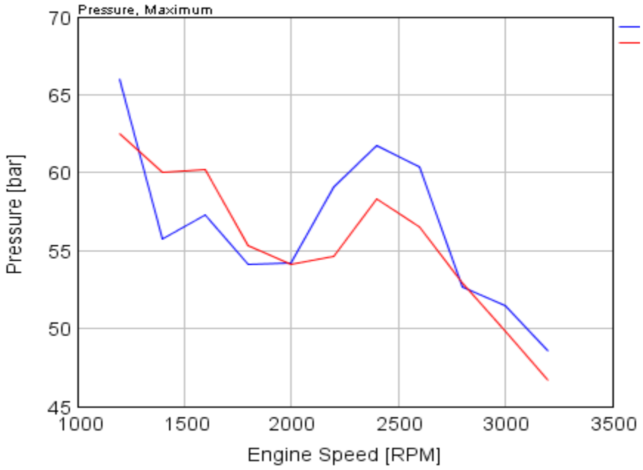


Fig. 44 Predicted peak pressure vs. RPM

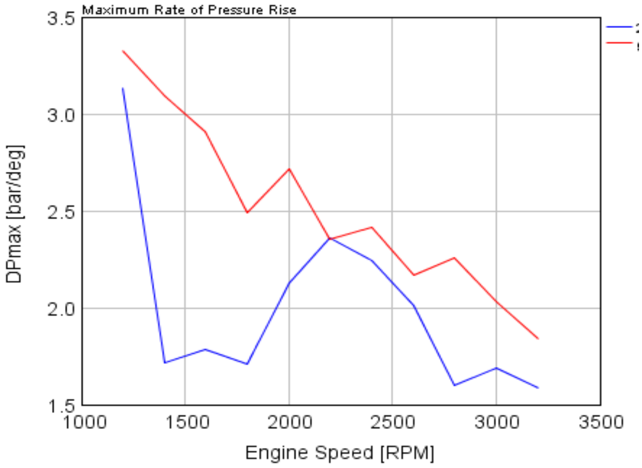


Fig. 45 Predicted PPR vs. RPM

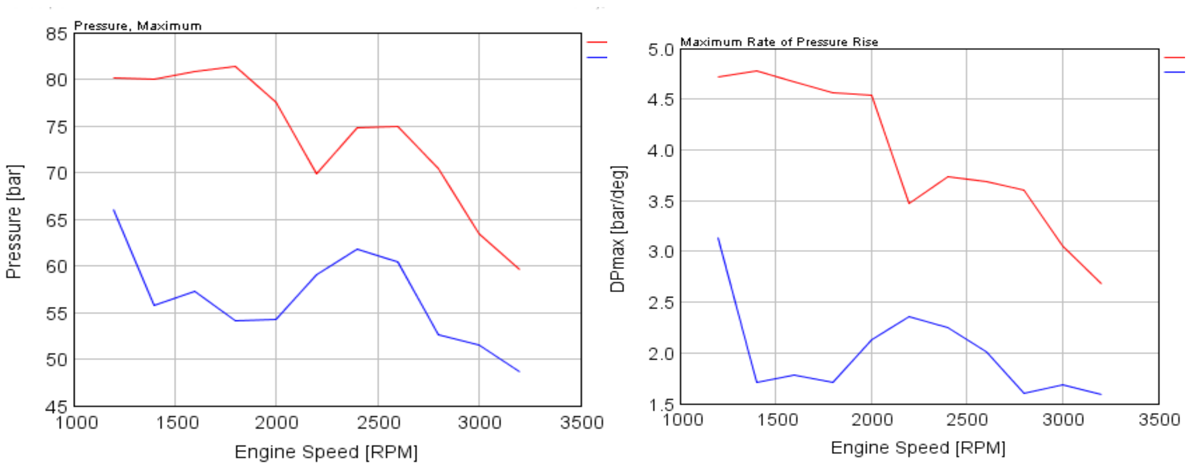


Fig. 46 Predicted peak pressure and PPR vs. RPM with 8° of advancement.

Peak pressure is increased by average of 1.33 times of initial value whereas PPR has increased 2.1 times of initial value.

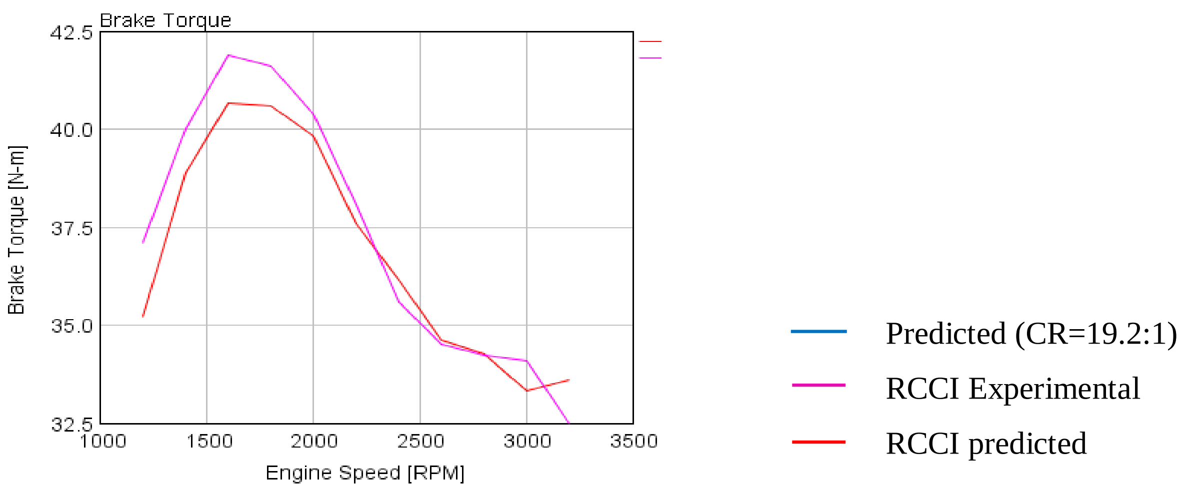


Fig. 47 Torque vs. RPM

7.4.2 GSR 30%

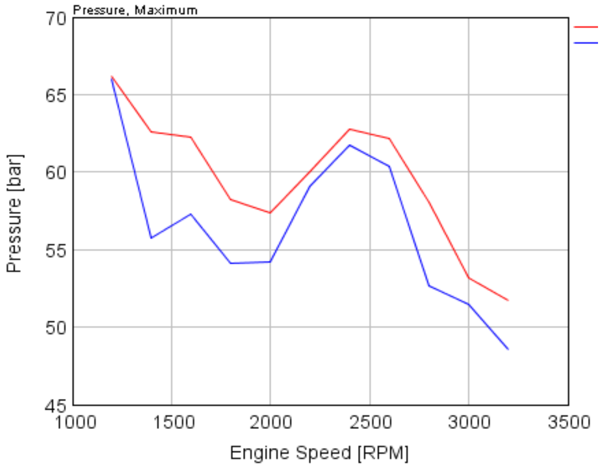


Fig. 48 Predicted peak pressure vs RPM

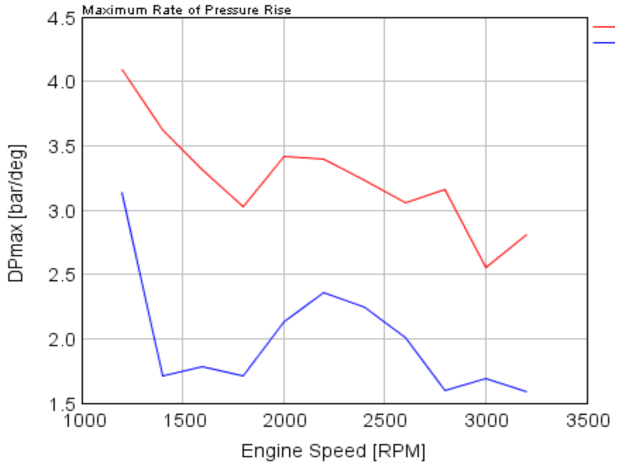


Fig. 49 Predicted PPR vs RPM

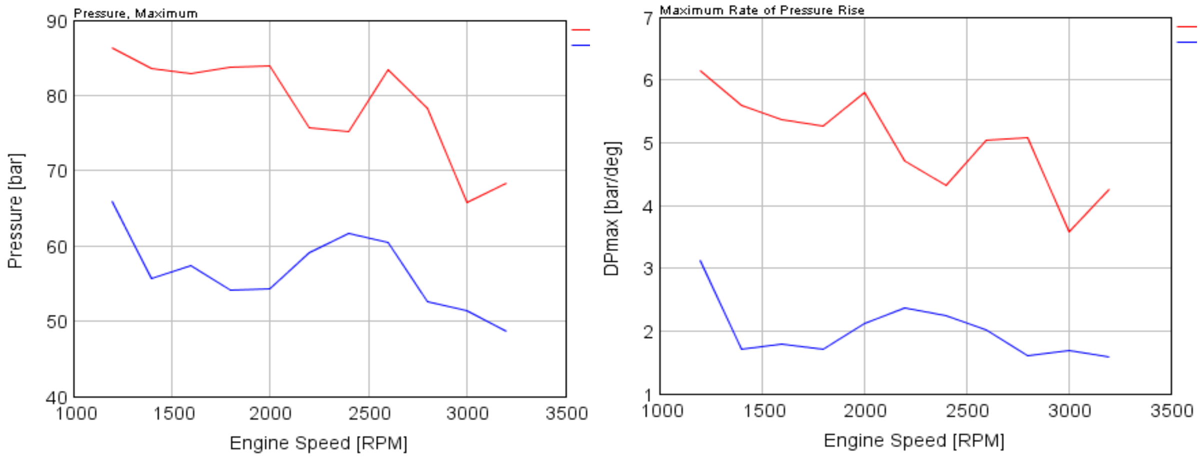
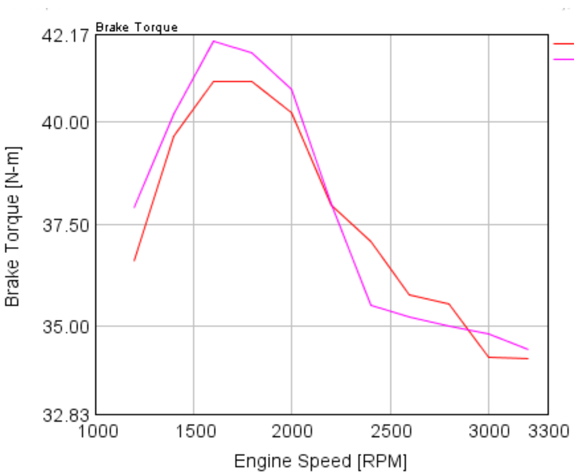


Fig. 50 Predicted peak pressure and PPR vs. RPM with 8° of advancement.

Peak pressure has increased by average of 1.39 times of initial value whereas PPR has increased by 2.57 times of initial value.



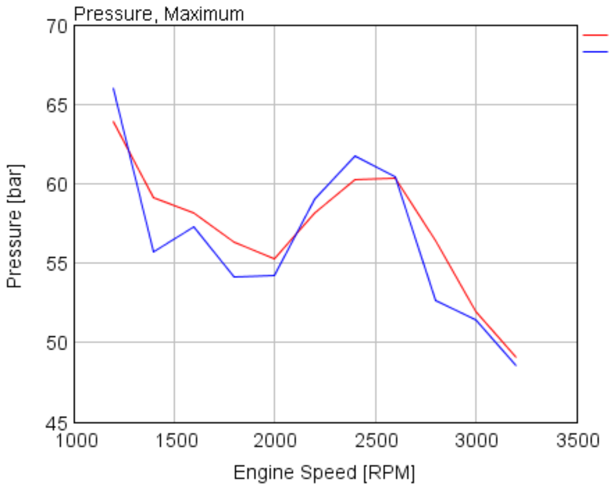
Predicted (CR=19.2:1)

RCCI Experimental

RCCI predicted

Fig. 51 Torque vs. RPM

7.4.3 GSR 40 %



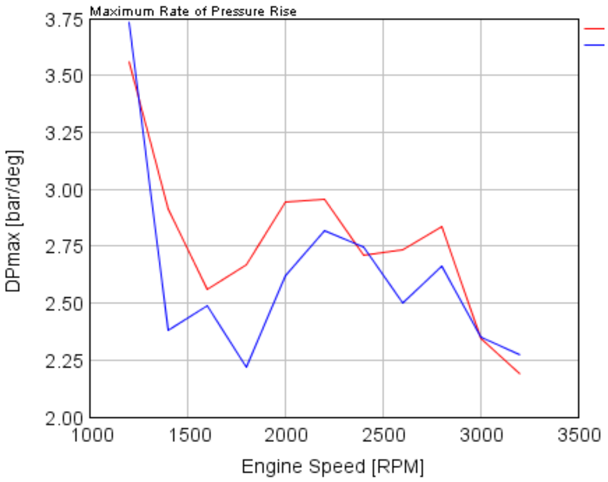


Fig. 52 Predicted peak pressure vs. RPM

Fig. 53 PRR vs. RPM

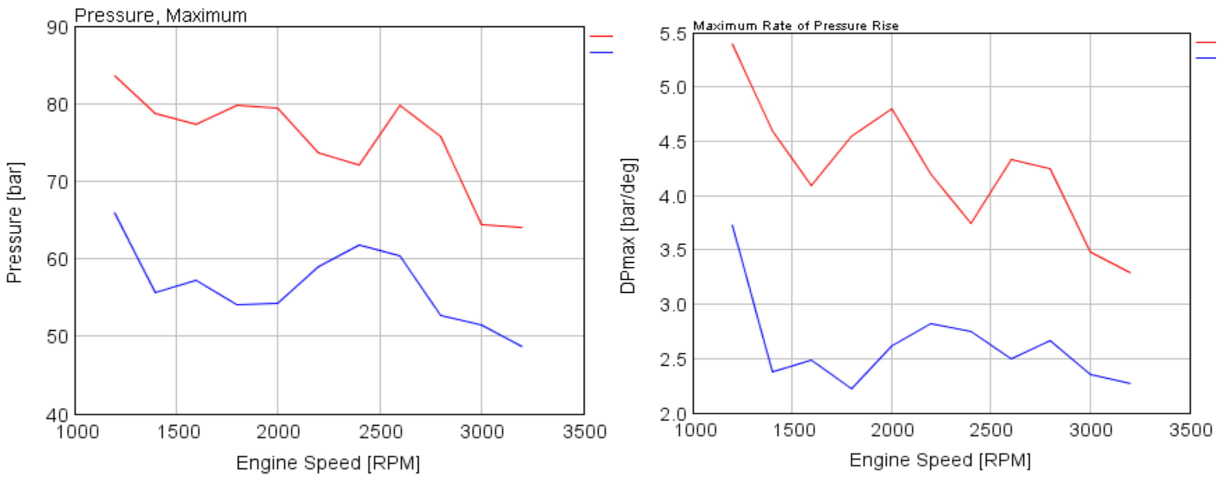


Fig. 54 Pressure and PPR vs. RPM with 8° of advancement. Peak pressure has increased on average of 1.31 times of initial value whereas PPR was increased by 1.63 times of initial value.

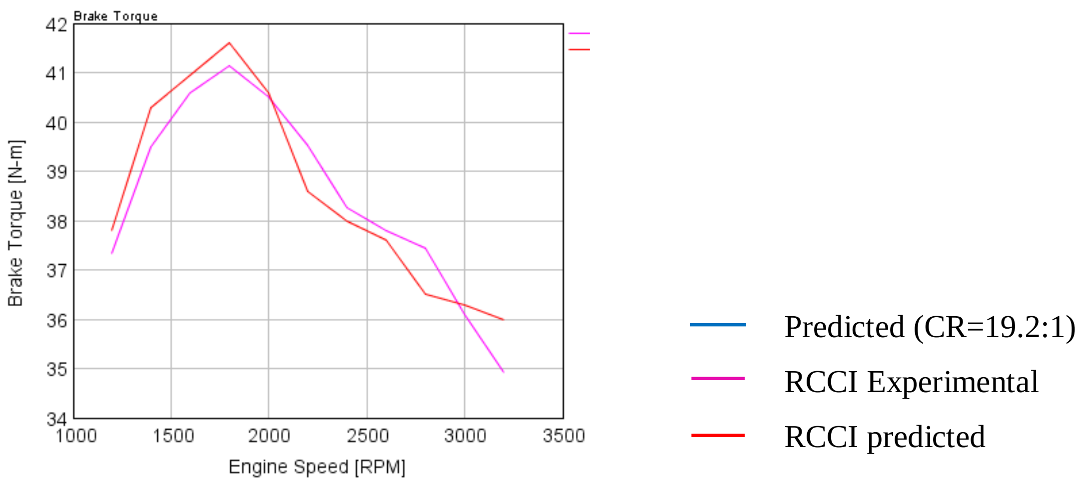


Fig. 55 Torque vs. RPM with 8° of advancement percentage of error between simulation and predicted within 10%.

7.4.4 GSR 50%

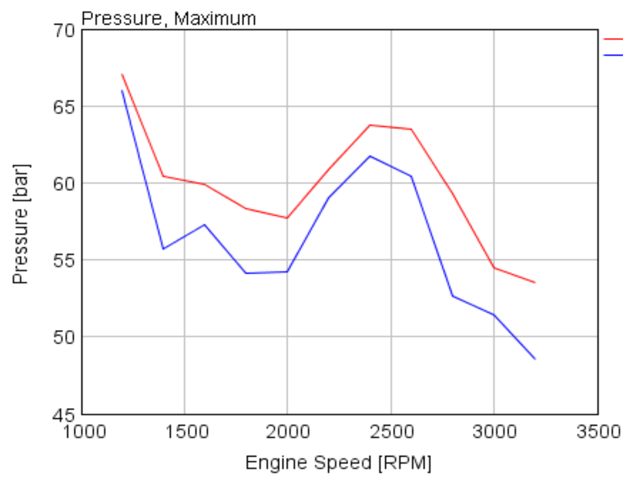


Fig. 56 Peak pressure vs. RPM

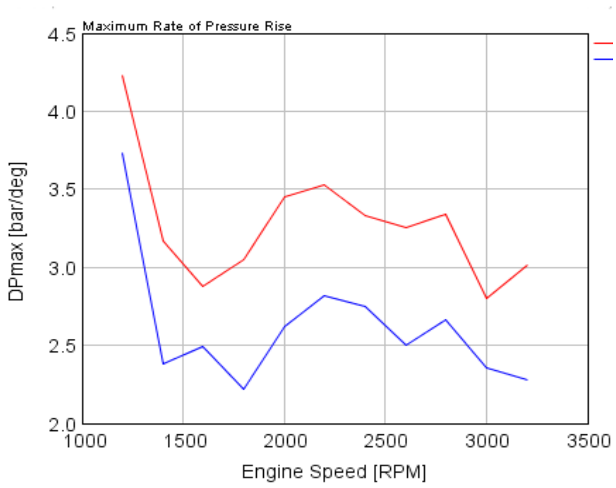


Fig. 57 PPR vs. RPM

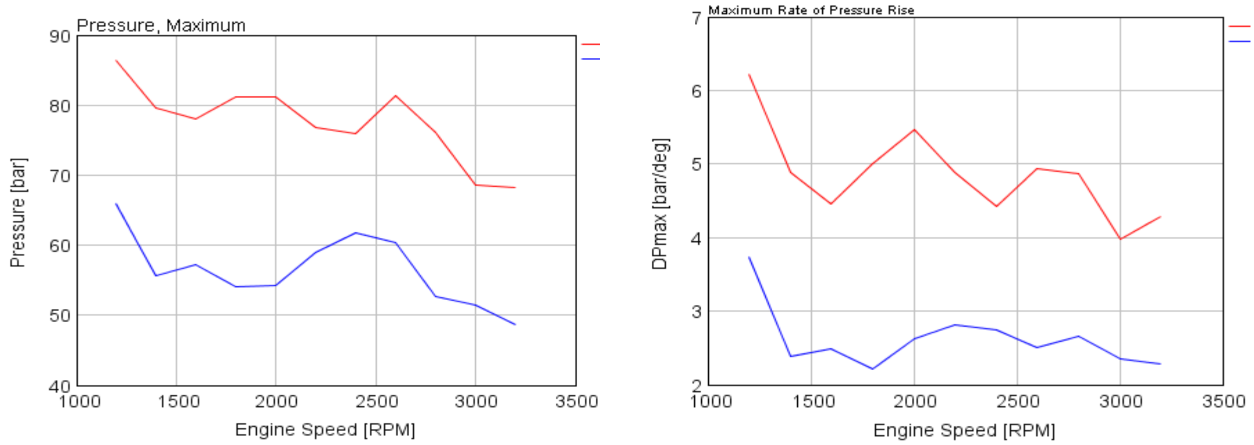


Fig. 58 Pressure and PPR vs. RPM with 8° of advancement.

Peak pressure has increased on average by 1.47 times of initial value whereas PPR has increased by 1.87 times of initial value.

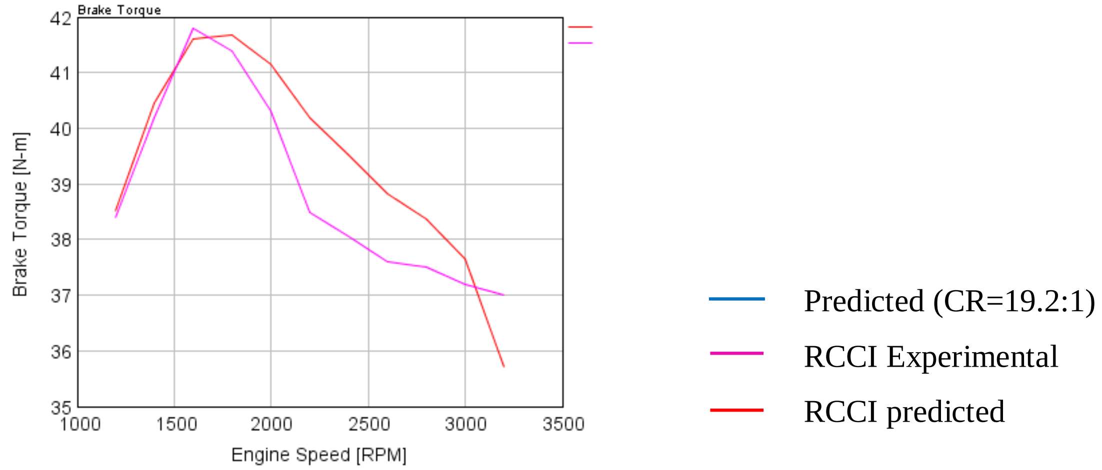


Fig. 59 Torque vs. RPM with 8° of advancement

Advancing the DIT results in higher in-cylinder temperature and pressure, the formation of NOx is also higher. If DIT is retarded, the peak of pressure crank angle occurs after TDC. Retarding injection timing also reduces ignition delays. Further retarding the injection timing past TDC leads to low in-cylinder temperature and energy losses resulting in low thermal efficiency, high brake specific fuel consumption and high exhaust temperature. Considering the higher exhaust temperature condition advancing of timing was considered more suitable than retarding the time. Therefore during experimentation the DIT has been advance to its initial value (baseline engine DIT).

8.0 Effect on torque output with compression ratio and GSR (Simulation)

We know that torque output from engine directly depends on the compression ratio. However, higher compression ration can also lead to knocking problem in dual fuel engine. Therefore, according to the literature survey done engine with compression ratio 15.5, 16 and 17 are more suitable for the dual fuel operation. According to LIM Pei Li [34] dissertation report for the compression ratio between 16.6 -19.3, the in cylinder temperature was recorded below 900K that is the self-ignition temperature of CNG. Therefore dual fuel (CNG-diesel) engine can operate within these two CR.

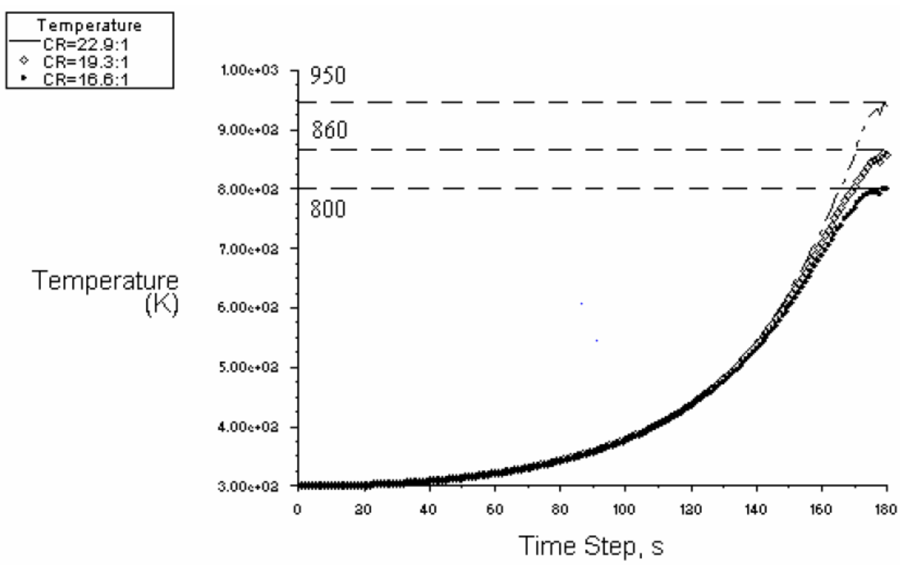


Fig. 60 Effect of in cylinder temperature with different compression ratio [34]

Figures 61, 62 and 63 show the effect of three different CR 15.5, 16 and 17.11 on the torque considering the same DIT. For each CR, GSR is varied from 20, 30, 40 to 50%.

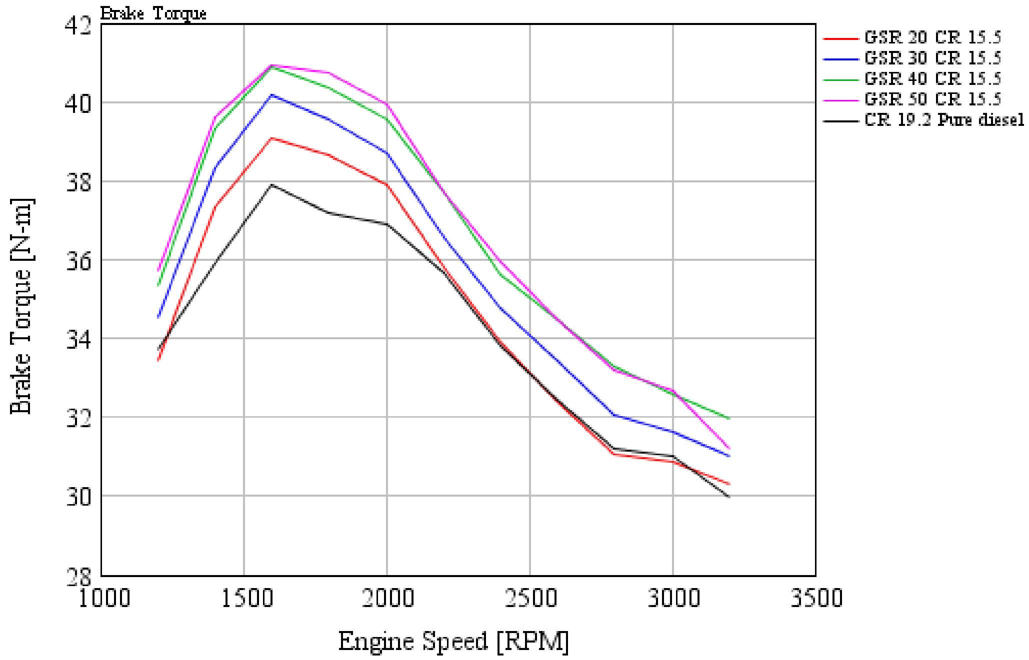


Fig. 61 Variation of Torque curve with different GSR at CR 15.5:1

For CR 15.5 (50% GSR), torque has increased by 7.9% at max. torque point and by 4.2% at rated point relative to baseline engine torque curve.

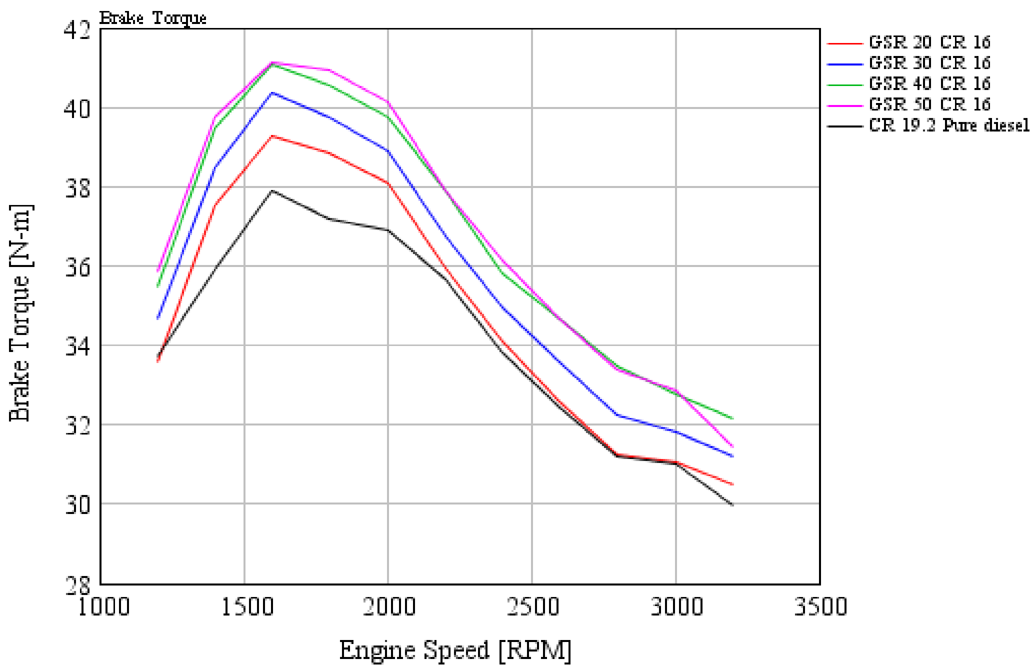


Fig. 62 Variation of Torque curve with different GSR at CR 16:1

For CR 16 (50% GSR) torque has increased by 8.4% at max. torque point and by 4.9% at rated point relative to baseline engine torque curve.

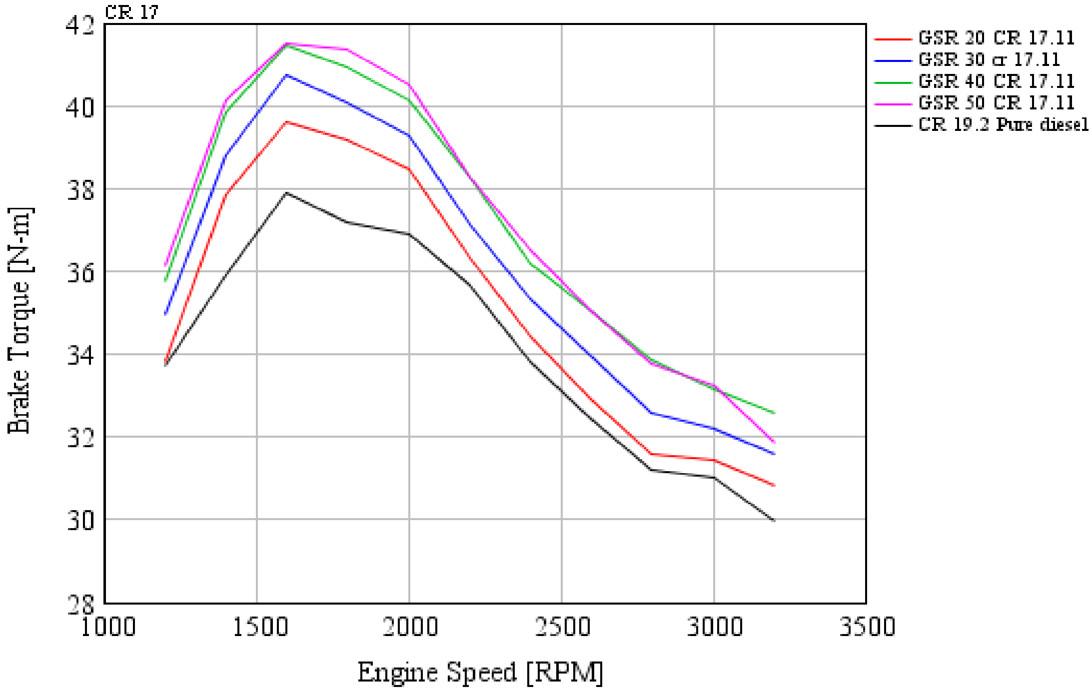


Fig. 63 Variation of Torque curve with different GSR at CR 17.11:1

For CR 17.1 (50% GSR) torque has increased by 9.4% at max. torque point and by 6.3% at rated point relative to baseline engine torque curve.

9.0 Experimental work

9.1 Logic building

Presently for working of engine on dual fuel mode, two ECUs are used as engine is electronically controlled. One ECU controls diesel system other controls the CNG system. Both the diesel and CNG ECU have two common sensors that are crank position and accelerator pedal sensor. Therefore, their respective signals are input to both ECUs.

Initially, engine assembly with modified intake manifold and 17.11:1 CR piston, is fired on pure diesel and then with diesel and CNG both on test rig. This is done in order to verify engine firing, coordination of sensors and ECU's working. Once all the connection were made, engine is fired and made to accelerate to full throttle on pure diesel and then on combined diesel-CNG. The shown test rig has provision for CNG cylinder mounting, diesel fuel tank, engine coolant system, exhaust manifold muffler and engine mounting brackets. The main advantage for which engine was tested on test rig is to minimise the time of testing on engine test bed, availability of which is a challenge, due to other ongoing projects.

Following figure shows the test rig setup:

Fig. 64 Test rig set up

9.2 Test operation

Test cell set up in the given company consists of:

1. Eddy current dynamometer with dyno controller outside the test cell by means of which an operator can conduct test. The dyno controller not only contains speed and throttle controller of engine for controlling the speed and load of the engine, it is also equipped with indicating lamps for any malfunctioning inside the cell. Dyno constant for dynamometer is 7021.54. All accessories required for normal working of engine inside the cell including exhaust temperature and back pressure, lubricating oil pressure and temperature sensors are controlled by means of Orbit software which is directly linked with dyno controller.
2. Fuel conditioning system is a heat exchanger that maintains the temperature of the diesel (40º C). Out fuel from this unit is directly supplied to the fuel pump or to fuel control unit in case of ECU controlled engine. Fuel in this unit is brought to desired temperature by means of warm water maintained at a certain temperature, this warm water container carries fuel tubes through which diesel is passed.
3. Water conditioning unit is used while testing water cooled engine. This unit keeps the inlet water temperature to engine at required temperature for fast warm up of engine. Maximum temperature limit is set to 90º C.
4. Fuel metering system measures the amount of fuel consumed by the engine. The unit of fuel consumption given by this system is in kg/hr. This system consist of a burette filled with

diesel. System measures the amount of time taken to empty the burette for calculation of fuel consumption.

Also test cell is having facility of encoder which controls throttle of mechanical and electronic engine (accelerator pedal). Additionally, fuel unit called fuel control unit (FCU) has been used in order to supply fuel from tank to injection pump at the desired pressure. For diesel and CNG ECU, INCA and SPARROW software are used respectively. Following figure shows the test bed setup:

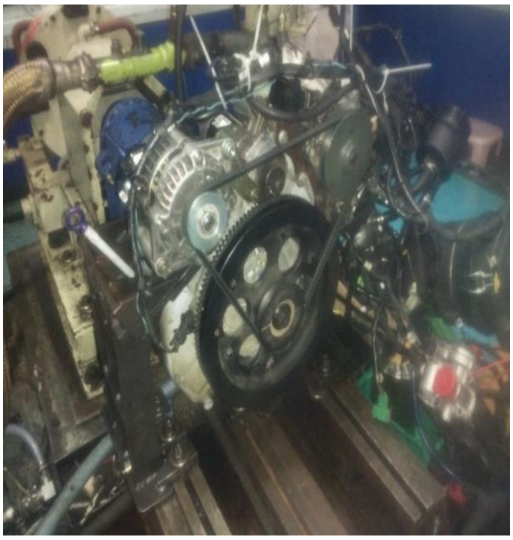


Fig. 65 Test bed set up

Propeller shaft connection of engine are made with dynamometer once water and fuel connections, acceleration pedal and engine working is verified. Once verification is done, engine is connected to the dynamometer with suitable adaptor that is connected between engine and dyno cardon shaft considering proper linear and angular alignment required adjustment is done.

For the CNG supply, gas cylinder (250 bar capacity) was set up inside the test cell. This cylinder is connected to the CNG port injector through CNG filter and regulator. CNG supply through cylinder is controlled by means of solenoid valve controlled regulator which maintains 6 bar constant pressure in the gas line.

9.3 Effect of GSR on performance and smoke (Experiment)

Performance at all Full throttle points (FTP) is taken on

a) Pure diesel operation
b) Dual fuel mode operation

Finally, comparison is done between simulated and experimental test data obtained at these points.

With initial conditions of fuel injected quantity and DIT of 19.2 CR, it was observed that under pure diesel operation with 17.1 CR, ignition delay increases due to decrease in the compression ratio. This increase in ignition delay while maintaining same fuel quantity as in baseline engine, results in increase of exhaust temperature. At 3200 rpm the exhaust temperature reached 702º C, which is critical for engine thermal structural strength because exhaust port, valves, valve seat and manifold are designed to withstand temperature up to 720° C. Therefore DIT is advanced by 4°. Same in case of dual fuel mode considering the knocking, stability and temperature of exhaust gases, DIT is advanced by 6 to 8°.

Following are the curved plotted during experiments.

1. Torque curve:

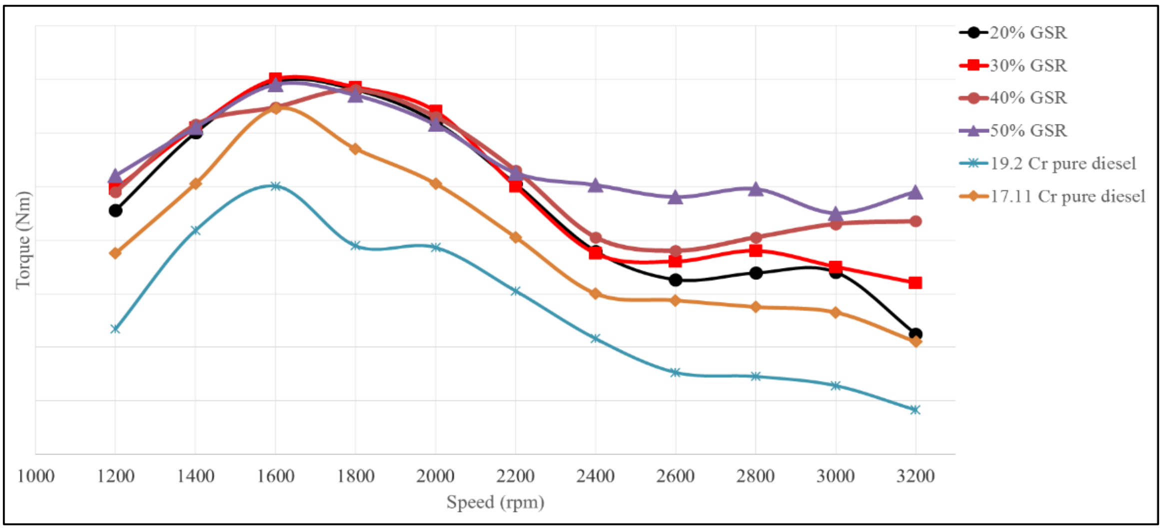


Fig. 66 Torque curves with different GSR

From the above graph, it is observed that torque curve of CR 17.11:1 is higher than that of CR 19.2:1 curve. Reason begin the DIT of lower CR is comparatively 4° advanced. Also, for dual fuel, DIT in Fig. 66 is advanced by 8º considering the knocking and stability of the engine. It may be considered that with change in heat release rate and advancing the engine dual fuel torque curved has raised. Torque of 17.11:1 CR has raised by average of 5 % relative to baseline curve. Whereas in case of 50 GSR, 15.4% of average torque raise was observed with 8° of DIT advancing. Keeping the same input fuel quantity for all the dual fuel GSR cases, it was observed that torque at the max torque RPM is same whereas torque at rated RPM increases as the GSR value was increased, but this also leads to high exhaust temperature.

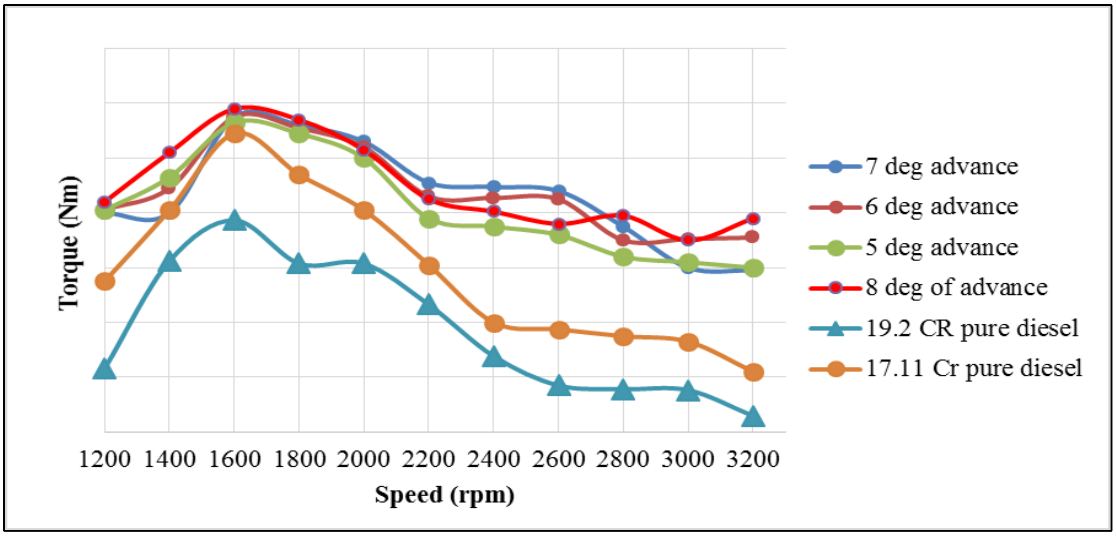


Fig. 67 Torque curves of 50% GSR with different DIT

Torque curve rises as the DIT is advanced for 50% GSR. Advance DIT is required as exhaust temperature has reached 700° C. At 1600 rpm with 6, 7 and 8º of advancement, it shows same torque whereas at rated point 8° of advance showed 10.78% higher torque than in baseline engine.

2. Exhaust temperature curve:

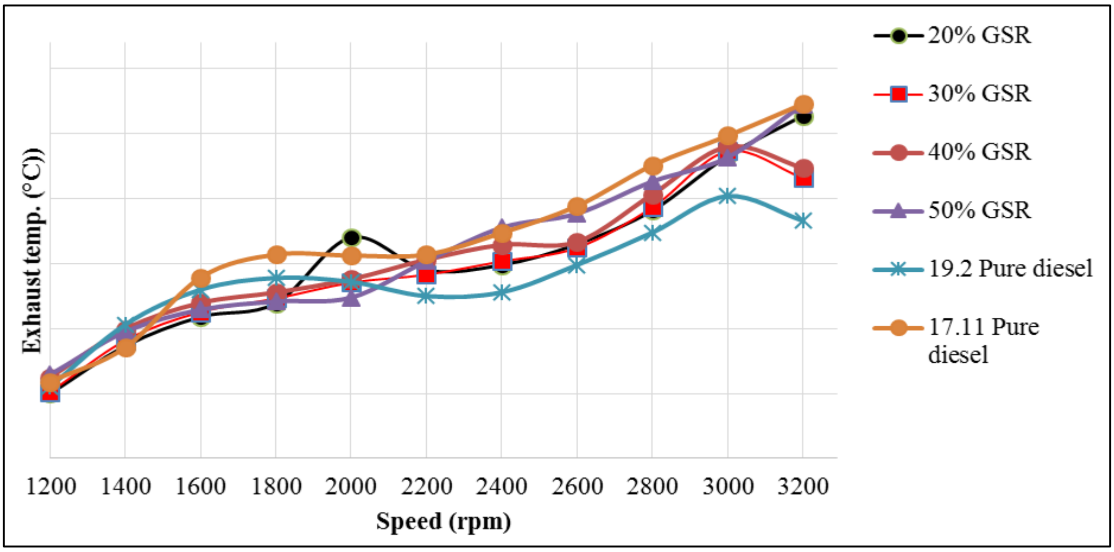


Fig. 68 Temperature curves with different GSR

As observed minimum temperature curve is of 19.2:1 CR under pure diesel operation. Advancing the engine helps in reducing the exhaust temperature but it also results in increase in in-cylinder temperature which results in higher NOx formation proportionately. Exhaust temperature at rated increased by 3.76% on average of initial value with 8º of advancing for 50% GSR.

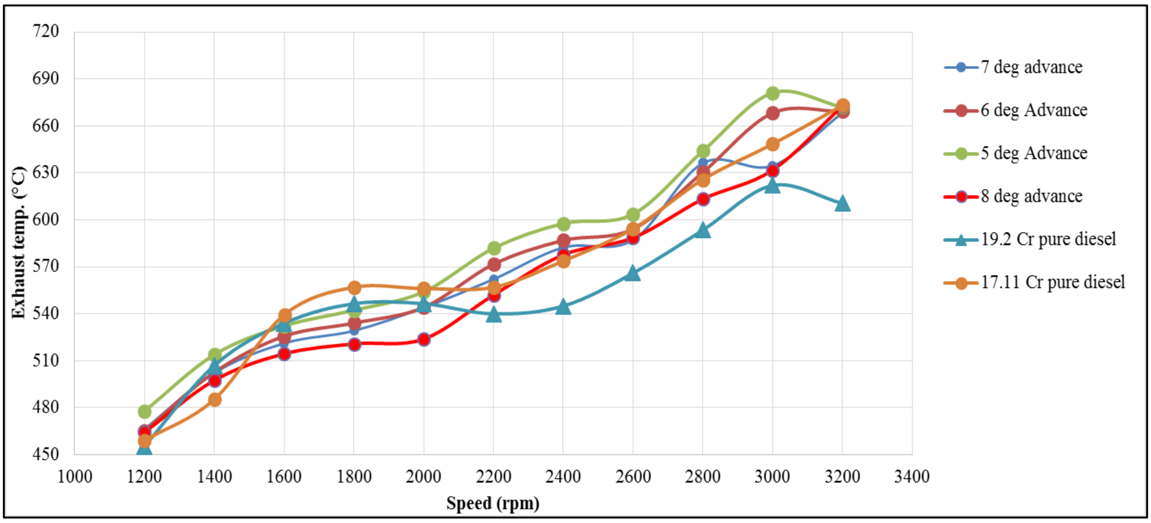


Fig. 69 Temperature curves for 50% GSR with different DIT

It is observed that with 5° of advancing the exhaust temperature has reached near 681° C at 3000 rpm which is 13% higher than baseline engine exhaust temperature. In fact, with 5° of advancing leads to 15.22% higher exhaust temperature at 3200 rpm. However with 8° of advancing at max. torque point 2.9% of fall in temperature is observed reason being at lower rpm exhaust burning is considerably reduced than at higher rpm.

3. BSFC curves:

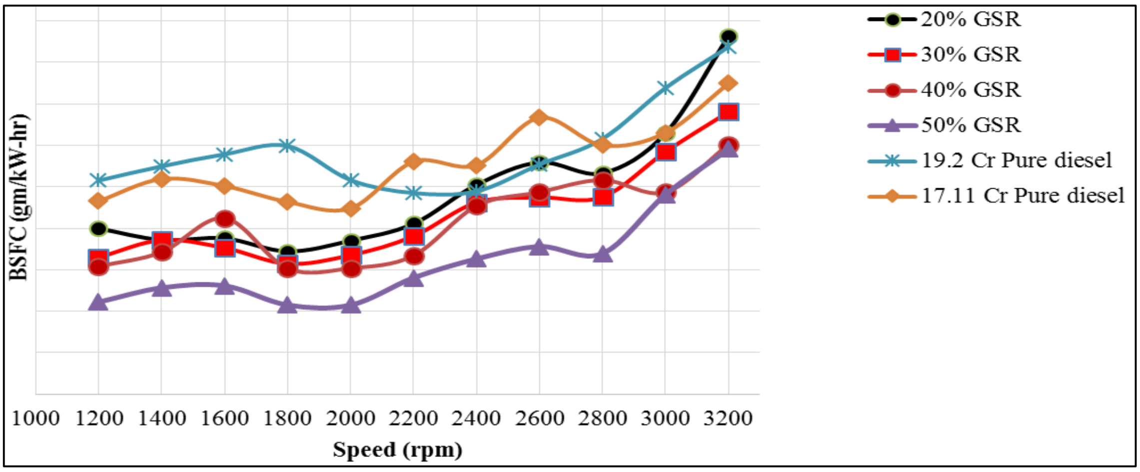


Fig. 70 BSFC curves with different GSR

Curves shows that the BSFC of 50% GSR is minimum for the same quantity of fuel input which means with same quantity of fuel input as in the baseline engine we can get higher power output. The main reason for increase in the power output is because of change in heat release rate of dual fuel engine i.e. heat release main duration has reduced, whereas, amount of heat release at peak point in case of dual fuel engine is higher than diesel engine. **From this curve an important conclusion is drawn that, we can decrease the quantity of fuel input to get baseline performance this will not only improve fuel economy but also reduce overall exhaust emission.**

BSFC at rated point has reduced by 17.1% whereas at max. torque it has reduced by 14.06% with 8° of advance w.r.t. its initial baseline engine. Overall curve came down by average of 14% with 50 GSR.

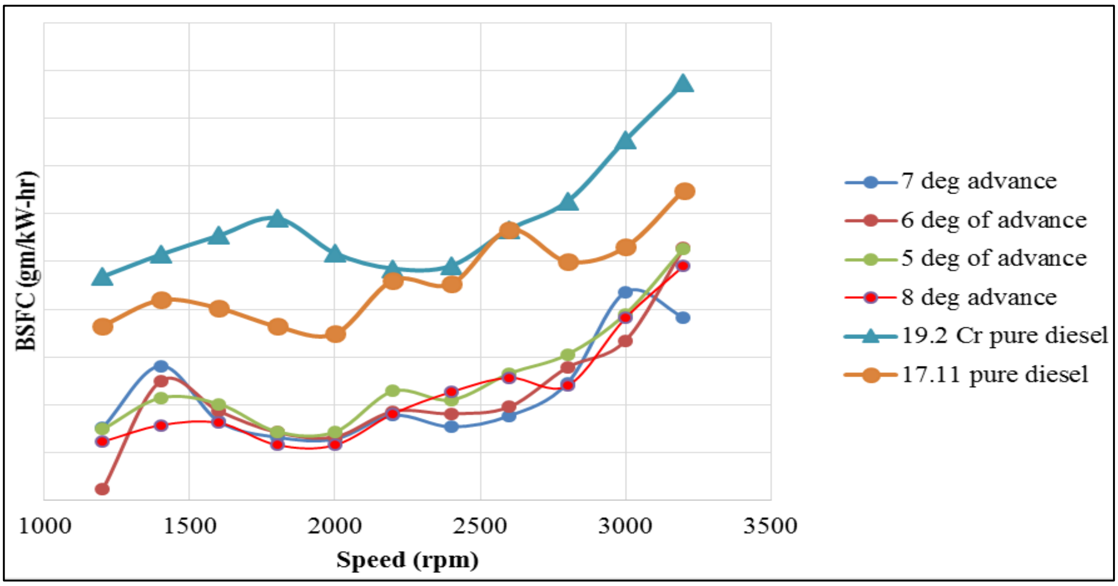


Fig. 71 BSFC curves for different DIT at 50% GSR

Trend of BSFC at 50% GSR is mostly same at all points whereas 8° of advancing shows slightly less BSFC as compared with other curves. At rated, 8.88% lower BSFC was observed with 6 and 7° of advancement w.r.t. baseline engine BSFC. Whereas 15.38% lower BSFC is observed with 8° of advance at max. torque point.

4. Smoke (HSU) curves:

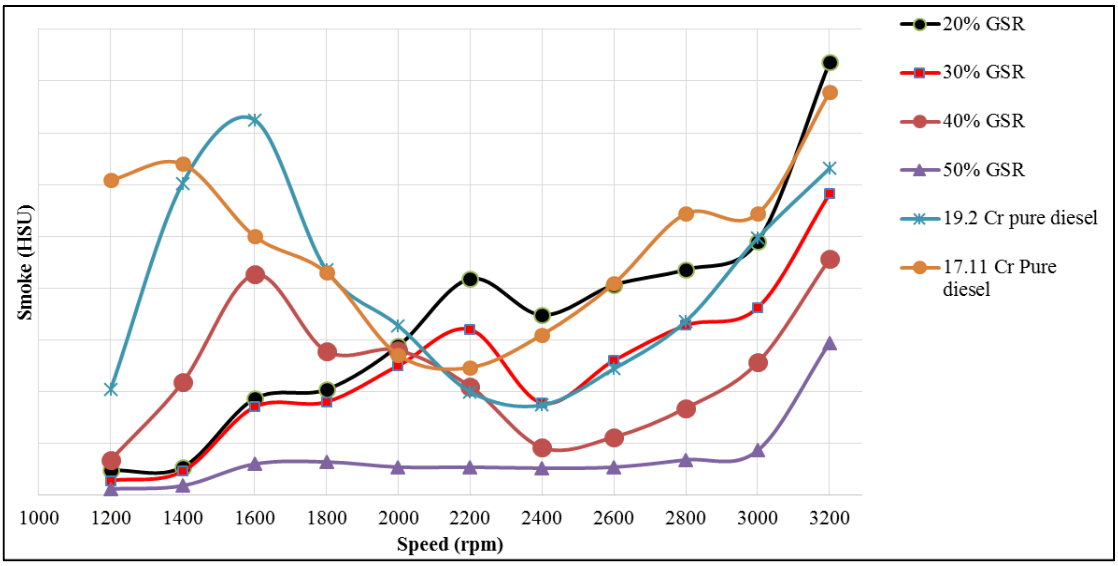


Fig. 72 Smoke curves with different GSR

From the above graph it is observed that smoke formation for 17.11 CR pure diesel operation produces more smoke for same fuel quantity as 19.2 CR pure diesel. This is due to reduction in compression ratio, combustion completion in lower CR diesel engine has reduced which is believed to increase the smoke formation. However, since the combustion of CNG is more complete increase in the GSR value has decreased the smoke formation.

Smoke at rated RPM has reduced by 53.4% and at max. torque it has reduced by 80.7% on average with 8° advancing w.r.t baseline engine for 50% GSR.

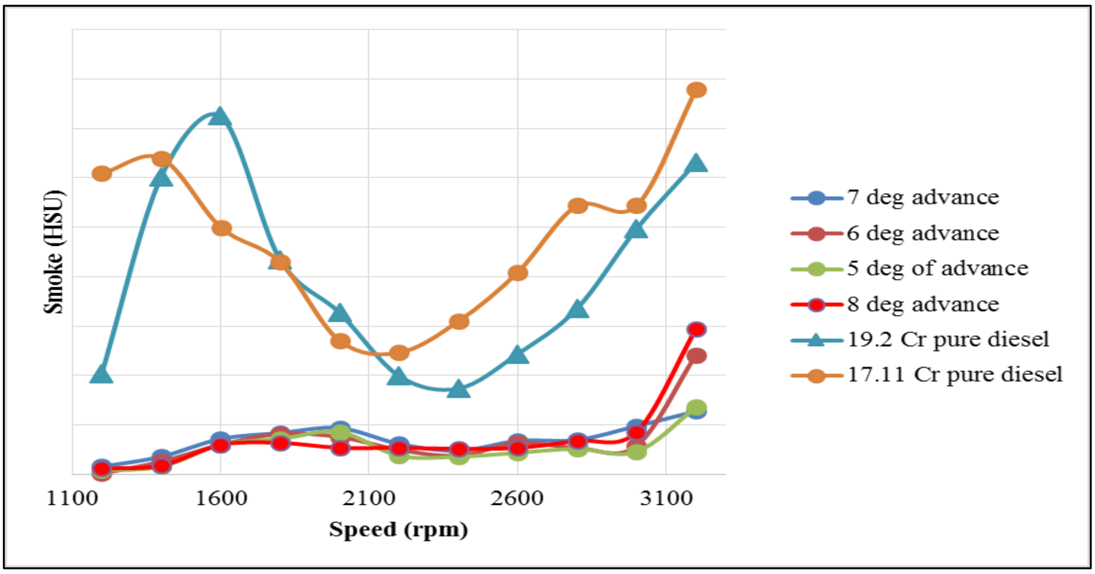


Fig. 73 Smoke curves with different DIT at 50% GSR

With variation in DIT for 50% GSR, very little difference in trend is observed. However, all the curves of 50% GSR are below 15 HSU. With 7º advancing system shows 79.72% lower smoke at rated compared to baseline engine. Whereas at max. torque 91.72% lower smoke for all DIT cases is observed.

5. Power curve:

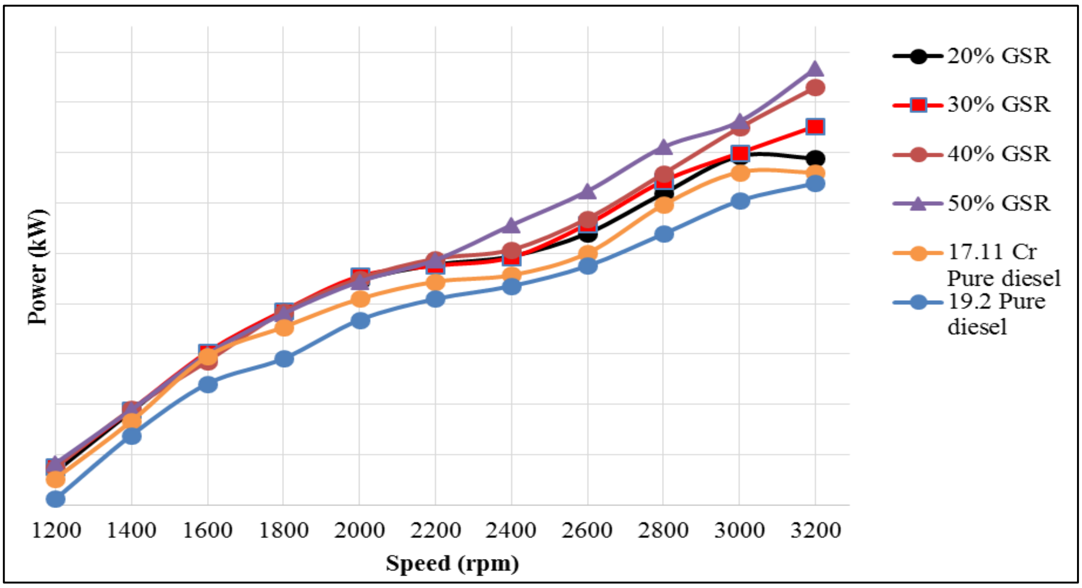


Fig. 74 Power curve with different GSR

As observed in torque curve with increase in GSR power increases. Power was increased by 14.17 % average with 8° advancement and 50% GSR.

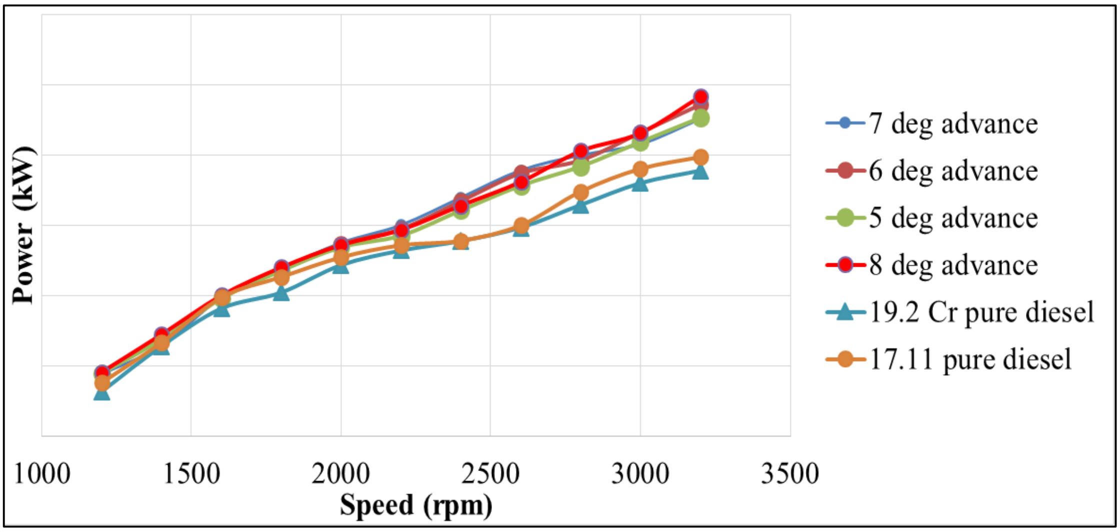


Fig. 75 Power curves with different DIT at 50% GSR

Power curve of 8° advance is higher, as is in torque curve, power has increased by 21.87% of baseline engine at rated point. At max. torque point, there is rise about 9.26% of power with same advance DIT. Maximum average power rise of 14.57% is observed.

10.0 Conclusion and future work

Experimental evaluation of dual fuel engine has certainly shown benefits over diesel and CNG engine in terms of BSFC and Torque/Power. Reduction in BSFC with minimum 10% increase in power (with 50% GSR); and price of natural gas being lower than diesel, would certainly reduce running cost of the vehicle. This is even more beneficial when engine is running with higher GSR. Also, increase in GSR has increased the peak pressure rise rate which directly effects the engine noise level. Pressure rise rate can however be limited by using pilot along with main injection, this results in smooth pressure rise rate. This also helps in further NOx reduction as in-cylinder pressure is reduced.

According to the literature survey, following are the two main objectives that have been successfully achieved during the project period:

1. To convert the diesel baseline engine to RCCI engine with minimum design changes so as to not to disturb the engine working on pure diesel operation.
2. To improve the brake thermal efficiency. As proved by simulation and experiment, if the fuel input for baseline engine is kept same as for dual fuel mode, power output is increased by 7% at rated with minimum of 20% GSR (also considering the effect of DIT advancing). This is due to change in the heat release curve for dual fuel burning.

Additional target that was set initially was of emission testing. However, due to time constraint and difficulty in availability of emission test cell, this activity could not be completed.

It was observed that with use of 50% GSR, engine out smoke is below 15 HSU, which is quite invisible to naked eye, and thus use of Diesel particle filter can be omitted from the final exhaust after treatment system. This will certainly reduce the exhaust after treatment cost.

10.1 Research recommendation

Present performance is taken only at FTP points. Further research in this project involves dual fuel engine testing on respective mode cycles. In present work two ECU's, one for diesel and other for CNG, are used in order to reduce overall dual fuel engine concept development cost. For further work, it is recommended to use single ECU in order to have good communication between two systems or to have conventional mechanical fuel injection with modified governor and CNG system controlled by ECU. This will also help in reducing engine overall development cost but mechanical fuel injection carries lack of ability of varying injection timing with speed and load.

Further, for vehicle running on dual fuel mode will require its proper optimisation done on both FTP and PTP points. Optimization is done by setting DIT, fuel quantity, EGR flow control, varying GSR value at different FTP and PTP points considering the effect on the engine out emission and its performance.

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Abbreviations:

η_{Comb} - Combustion efficiency

ATDC - After top dead centre

BSFC - Brake Specific fuel consumption

BTE - Brake thermal efficiency

CA50 - Crank angle at which 50% of fuel has burned

CDC - conventional diesel combustion

CFD – Computational fluid dynamics

CI - Compression ignition

CNG - Compressed natural gas

CR - Compression ratio

CRDI - Common rail diesel injection

DIT – Dynamic injection timing

DOE - Design of experiment

DPF - Diesel particle filter

EGR - Exhaust gas recirculation

EISFC - Equivalent indicated specific fuel consumption

ESR - Energy Substitution Ratio

ETC - European Transient cycle

FMEP - Friction mean effective pressure

FTP - Full throttle performance

GSR - Gas supplement ratio

GTE - Gross thermal efficiency

HCCI - Homogenised charged compression ignition

IMEP - Indicated mean effective pressure

ISFC - Indicated specific fuel consumption

LTC - Low temperature combustion

m_{Diesel} - Mass flow rate of diesel

m_f - Total mass of fuel injected

m_{NG} - Mass flow rate of natural gas

N - Rotational speed

NTE - Net thermal efficiency

PCCI - Pre-mixed charge compression ignition

PM - Particulate matter

PTP - Part throttle performance

P_{Max} - Maximum in-cylinder pressure

P_{Max} - Maximum in-cylinder pressure

PMEP - Pump mean effective pressure

R - Ideal gas constant

RCCI - Reactivity controlled compression ignition

RI - Ringing intensity

RoHR - Rate of heat release

Rs - Swirl ratio

SCR - Selective catalytic reduction

T- Maximum in-cylinder temperature

TMAP - Temperature manifold absolute pressure

UHC - Unburnt hydrocarbon

W_g - Output work