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Full Length Article

Combustion analysis of modified light duty diesel engine under high pressure split injections with cooled EGR

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ABSTRACT

The aim of the present research work is to study the combustion phenomenon in modified light duty diesel engine to run on Common rail direct injection (CRDI) system under cooled exhaust gas recirculation (EGR). The test engine is single cylinder 3.7 kW@1500 rpm direct injection diesel engine capable of injecting high pressure fuel for both retarded and split injections. The engine is fitted with separate electronic variable timing fuel injection kit instead of conventional fuel supply system. Retarded injection consists of single injection at 11° before top dead centre (BTDC) and split injection consists of both pilot injection at 54°BTDC of 10% mass share and main injection at 11°BTDC of 90% mass share. Diesel is injected directly in to the engine cylinder for both retarded and split injections at pressures of 200,230,250,300 and 350 bar respectively. Cooled EGR is circulated along with intake air for flow rates of 5% and 10% (wt/wt) basis. The experimental set up is capable of delivering precise control of fuel and EGR flow rates at all operating conditions. Test results show that there is trade-off exists between retarded and split injections at 350 bar injection pressure at full load conditions. Retarded injection has 34% brake thermal efficiency while split injection exhibits only 32.1% for 5% EGR flow rates. But higher EGR flow rates of 10% both retarded and split injection has nearly same brake thermal efficiency of 30.1%. Split injection reduced the combustion duration, ignition delay and exhaust gas temperatures for higher EGR flow rates compared to single retarded injection.

1. Introduction

Globally there is a large concern on environmental pollution due to increase in tremendous vehicular population all over the world. Engine emission mainly depends on fuel quality, injection strategy and engine operating conditions such as speed, load, EGR etc. Portable power generation sets and agriculture light duty diesel engine contributes significantly for deterioration of ambient air quality. EGR can be applied to engines with advanced combustion technology for diesel, biodiesel and gaseous fuels [1]. Earlier reports shown that cooled EGR reduces NO_x effectively compared to hot EGR but regarding brake thermal efficiency hot EGR is better than cold EGR. This is because of higher intake charge temperature [2]. Addition of n-pentanol in diesel fuel in unmodified diesel engine with 45%EGR reduced NO_x by 30% at full load with simultaneous decrease in thermal efficiency. Similarly combustion characteristics of lignin –derived cyclohexanol as a blend in diesel fuel with EGR resulted decrease of 61% NO_x and 14.2% smoke levels with 30%EGR but a further decrease in brake thermal efficiency was reported [3].

Studies conducted on turbocharged heavy duty DI diesel engine for the effect of cooled EGR temperatures for various flow rates revealed low temperature EGR reduced the bsfc and soot but its effect on NOx emissions is limited [4]. The effect of various EGR rates, equivalence ratio, flame behavior on thermal efficiency and emissions when EGR was applied resulted a decrease in equivalence ratio neared 1.00 [5]. Increase in EGR flow rates increased the combustion duration by 39.8% when the EGR rate is increased from 0 to 30%. This is due to the domination of heat release rate under high temperature and equivalence ratios [6]. Investigations on EGR on low combustion engine showed that there is no change in peak pressure with increase in EGR flow rate and further ignition delay increased with decrease in initial boiling point. The combination of EGR and early valve timing in medium speed turbo charged diesel modified diesel engine with two cam shaft configurations resulted a greatest decrease in NOx with trade-off between UHC and NO_x was greatly reduced. Also CO and PM emissions are much reduced over the entire load region due to higher volumetric

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Nomenclature

BTDC	Before Top dead centre
BDC	Bottom dead centre
B.P	Brake power (kW)
BSFC	brake specific consumption (kg/kWhr)
CAD	Crank angle in degrees
CRD	Common rail direct injection
DI	Direct injection
EGR	Exhaust gas recirculation
EGT	Exhaust gas temperature
HSU	Hatridge smoke units
MPFI	Split injections
PCCI	Premixed charge compression ignition
SI	Retarded Single Injection
SFC	Specific fuel consumption kg/hr
TDC	Top dead centre

efficiency [7]. EGR can also be applied to diesel Engine run on fuel prepared from reprocessed plastic waste. The effect of cooled EGR on single cylinder DI diesel engine run on waste plastic oil and reported that cooled EGR reduced the combustion peak temperatures with 20% EGR. Experimental investigations on HCCI with 100% gaseous fuel LPG and reported that brake thermal efficiency increases at part loads for all percentages of EGR but at full load higher flow rates of EGR affects the brake thermal efficiency [8]. The effect of early pilot injection with exhaust gas recirculation in HCCI direct injection heavy duty diesel engine investigations showed that NO_x emissions decreased with increase in pilot fuel quantity and suggested that low level of EGR is effective method in reducing NO_x emissions [9].Test runs conducted on a single cylinder medium speed research engine delivering power at 300 kW with high pressure multiple injections and cooled EGR on spray characteristics, combustion, and the extent of high pressure injection. The results concludes that soot emissions can be lowered effectively both at part and full loads with high pressure injections with sophisticated EGR system [10].Common rail direct injection (CRDI) with multiple injections with or without separation is widely used in passenger cars due to the increase in demand for higher torque, specific power output and reduce fuel consumption and emissions. This fuel injection equipment (FIP) was developed by Delphi. This fuel injection system can inject the fuel at high pressure and the time separation between two injections can be reduced to 0 micro seconds. With this injector diffused combustion was avoided with EGR. However early injection with EGR resulted higher HC and CO emissions due to lean combustion and lower combustion temperatures [11]. Effect of EGR on pilot injection with multi hole injector having 14 orifices under idling conditions showed that significant increase in power output with simultaneous reduction in PM and NO_x [12]. Low temperature exhaust gas recirculation will handle higher mass of exhaust gas due to the less volume occupied by it. This will create more space for fresh air so that the engine can digest more EGR compared to high temperature EGR. Low temperature exhaust gas in sports utility vehicle (SUV) showed that the engine can take higher amount of air so that more heat can be supplied by EGR process. The results shows that for the same amount of fuel with low temperature exhaust gas around 16.4% reduction in NO_x and 23.3% lower PM was achieved [13]. Similar results are also reported with ultra-cooled EGR with retarded injection increased thermal efficiency, even with low pressure injection [14]. The effect of premixed charge compression ignition (PCCI) combustion with EGR studies reported that advancing pilot injection resulted the in-cylinder peak pressure, ignition delay was shortened [15]. Investigations on 50% of pilot and main injection with high flow rate of EGR in single cylinder high speed DI diesel engine concluded that the combined effect of split injection and EGR requires less amount of EGR as compared to single injection with significant brake thermal efficiency [16]. High pressure split injections consisting of 6 injections per cycle in AVL single cylinder DI diesel engine with electronically controlled injections of pressure up to 200Mpa. Findings show that up to 100Mpa the combustion is faster due to which the in cylinder gas temperature is high compared to conventional fuel injection system [17]. Optimized split injection and EGR at various high-pressure injections resulted an increase in NOx with slight decrease in soot and bsfc was reported. However with split injection with minimum quantity of pilot injection in late injection improved the brake specific fuel consumption [18]. Numerical simulations showed that 5 split injections without EGR is more effective than 3 split and single injections [19]. Reactivity controlled compression ignition (RCCI) at high loads with EGR and boost pressure using computational fluid dynamics (CFD) showed that ignition delay is highly sensitive to EGR variation in late injections. Also Combustion and emissions are improved and the sensitiveness of EGR reduced with increase in injection pressure [20]. Experimental conducted on single cylinder diesel engine modified to CRDI and EGR analyzing the experimental results using grey-fuzzy taguchi approach concluded that EGR is the most influencing parameter on engine performance [21]. Numerical studies in Split injection strategy in Indirect injection (IDI) engine of 5.9 kW power at 730 rpm showed that for 25% of total fuel injected in second pulse with 20 deg crank angle dwell between successive injections reduced the total soot and NO $_x$ effectively [22]. At present limited research was conducted on the combined influence of ultra cooled EGR and split injections on single cylinder low duty diesel engine under high and low load operating conditions. Hence in the present work intensive experiments are conducted on modified single cylinder low duty high speed engine of 3.5 kW power at 1500 rpm running speed using CRDI system. The fuel injection system was modified to CRDI capable of injecting precise quantity and timing controlled by external programmable electronic controller. The injection system is capable of injecting the fuel at very high pressures up to 1000bar. The split injection consists of 10% of the fuel injected during pilot injection at −54degCAD and remaining 90% fuel injected at main injection at −11degCAD with two ultra cooled EGR flow rates of 5% and 10% on wt/wt basis.

1.1. Experiment set up and methodology

Crank angle Sensor (2) AC Dynamometer (3) Load-cell (4) Dynamometer (5) Load Control
(6) Fuel-pump (7) ECU (8) SMPS (9) High speed Daq (10) Encoder signal Daq (11) Sensor Daq
(12) Ethernet (13) Fuel Control Module (14) Diesel Fuel Tank (15) Indus Diesel smoke meter
(16) Exhaust gas analyser (17) Setting chamber (18) EGR (19) Furnace (20) EGR
ECU (21) control Valve (22) Mixing Chamber (23) Air-Flow meter (24) Laptop

The engine used in this experimentation was a Kirloskar make single cylinder direct injection diesel engine delivers 3.5 kW at

1500 rpm with specifications mentioned in the Table 1.The engine is coupled to dynamometer of capacity 3.5 kW supplied by Powermag India Ltd. The combustion pressure was measured by air cooled kistler make pressure sensor. It is peizo electric sensor for continuous cylinder pressure monitoring of the engine. A charge amplifier is connected to the sensor by high temperature cable. The good linearity and long term stability ensures reliable and repeatable measurements over long period of time. The incylinder combustion temperatures and exhaust gas, water temperature etc are precisely measured using thermo couples and the measured output are connected to data logger DASTEP8. The engine is fitted with variable injection timing kit which consists of all the components of CRDI and electronic injection controller is provided to control the injection timing and quantity. The fuel supply system consists of low and high-pressure circuits, which maintain constant and unvarying pressure. The high-pressure pump is run by 5H.P. electric motor. The fuel pressure is controlled by a control valve operated by a electromagnet solenoid valve. The fuel rail pressure is fitted with a pressure sensor and a pressure relief valve is provided for safety against high-pressure build up in the rail. The mechanical injector was replaced by electro magnet system in which the solenoid armature does not control the pintle but by the movement of a small rotating ball, which regulates the fuel flow from a valve control chamber within the injector. A crankshaft position sensor typically sends the data to the electronic control unit (ECU). This electronic control unit was a pre-programmed microcontroller used to signal start of injection and duration. This microcontroller was connected to the PC hardware by taking the input signal from the crankshaft position encoder. Thus the electronic fuel injector is controlled by ECU. Ultra cooled EGR is circulated in the engine by tapping the exhaust gases in the main exhaust line through a stepper motor control valve and passed through a cooler. To know the mass flow rate of EGR a digital manometer is provided at the inlet manifold of the engine.

2. Experiment method

Experimentation was conducted on a standard single cylinder, 4 S DI diesel engine with rated power of 3.7 kW. The specifications of the engine used for experimentation are detailed as tabulated in Table 1. Engine speed, intake air, cooling water flow rates and EGR temperature were maintained constant throughout the experimentation. The detailed experimental setup is as sketched in Figs. 1a and 1b. Eddy current dynamometer was connected to the engine to apply load on the engine in the steps of 0.5 kW. Initially, the engine was run on neat diesel, performance and emission parameters were recorded by varying load from 0.5 to 2.8 kW. During phase-I of experimentation, engine was run on neat diesel on retarded single injection at −11CAD without EGR and fuel injection pressures are varied from 200,250,300 and 350 bar respectively. During the phase –II of experimentation two EGR flow rates of 5% and 10% respectively was circulated for the same retarded injection angle for the above pressures respectively. Then during the phase -III of experimentation two split injections at angles of −54CAD and −11CAD was chosen whose mass share for pilot injection is 10% and main injection of 90% was chosen and experimentation was conducted as per the above mentioned injection pressures and EGR flow rates. Throughout the experimentation, Engine was run at constant speed of 1500 rpm. The engine was allowed to reach steady state condition before recording the parameters. All sets of reading were taken when engine is running on current experimentation fuel and necessary care has been taken that no fuel from previous experimentation was left out in the engine diesel oil filter. Every set of experimentation was repeated for three times and mean readings were recorded.

Table 1
Engine specifications of the modified single cylinder diesel high pressure fuel injection test rig.

Sl. No.	Parameter	Specification
1	Make	Kirloskar
2	Model	AV1
3	Type	Single cylinder, 4 Stroke, Direct Injection, Water cooled CI Engine
4	Rated power	3.7 kW@1500 rpm
5	Engine speed	1500 rpm
6	Fuel	Diesel
7	Bore & Stroke	80 × 110 mm
8	Displacement	$553E{-}6\ m^3$
9	Compression ratio	16.5:1, Range: 13.51–20
10	Injection pressure	200–350 bar
11	Cylinder pressure	Piezo Sensor, Range: 200 psi
12	Nozzle	1 hole, Ø 0.0020 m
13	Dynamometer	Eddy current dynamometer
14	Orifice diameter	20 mm

2.1. Uncertainty analysis

The accuracy of experiment results mostly depends on uncertainty analysis. Several factors contribute the uncertainty during experimentation such as observation, environment and instrument type. Before taking the reading for each load the engine is run for at least 30 min time and there is no much variation in cooling water temperature. Engine data is collected for each load at least 3 times and graphs are plotted by taking the average values. The measuring range and accuracies of parameters were represented in the Table 2.Uncertainty analysis was conducted for engine performance, emissions and smoke parameters based on analysis are presented in Tables 3 and 4 [23,24]. Detailed calculations are shown in Appendix A. It is found that the uncertainties measurable and computed parameters for experimentation with 95% confidence level at 350 bar injection pressure for split injection @10% EGR is range between 0.5 and 1.92%

3. Results and discussion

In exhaust recirculation with ultra cooled EGR the engine is run at constant speed varying fuel injection angle and pressures throughout the experiment and the results are compared with neat diesel engine operation without EGR. Combustion parameters like in cylinder gas temperature, rate of heat release, peak pressure rise rate, igntion delay are measured with respect to crank angle are measured and compared to that of neat diesel operation.

3.1. Brake thermal effeciency

The effect of exhaust gas recirculation on brake thermal effeciency for single and split injections for injection pressures ranging from 200 bar to 350 bar at different loads are shown in Fig. 2. Generally the brake thermal effeciency increases with increase in injection pressure. Generally higher EGR flow rates decreased the brake thermal efficiency marginally for both single retarded and split injections. This reduction in brake thermal effeciency is due to the decrease in availability of oxygen for combustion process because of oxygen replacement with higher percentages of EGR. EGR decreases excess air fuel ratio due to the replacement of air leading to deteriorated combustion and there by a decrease in brake thermal effeciency. The obtained results are in accordance with the earlier results reported [25,26]. At low EGR flow rates, single retarded injection has higher brake thermal effeciency compared to split injections at 0.75 & full load operating conditions at 350 bar injection pressure. However the experimental results

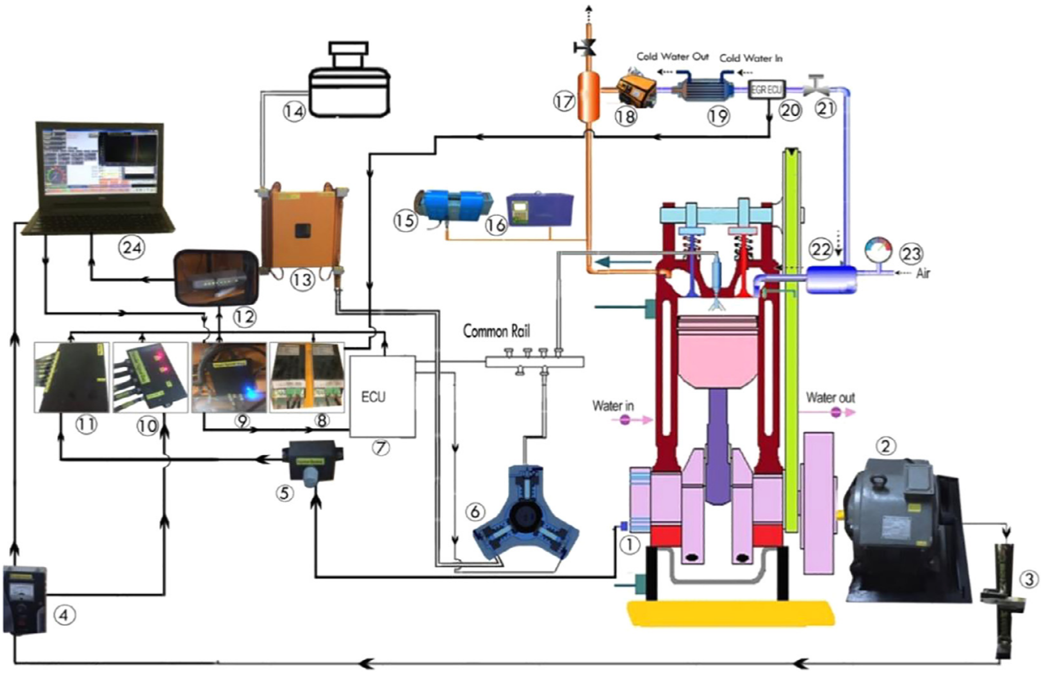

Fig. 1a. Schematic diagram of high-pressure multiple split injection engine test rig with exhaust gas recirculation facility.

Fig. 1b. Experiment Test Rig.

shows that without EGR split injection has higher brake thermal efficiency than single retarded injection for all injection pressures. At moderate loads for higher EGR flow rates retarted single injection is exhibiting better brake thermal effeciency compared to split injections as can be seen in Fig. 2. At full load operating condition 10% EGR flow rate there is a marginal variation of 0.01% brake thermal efficiency for both SI and MPFI injection at 350 bar. This shows that for higher EGR flow rates there is no much variation in brake thermal effeciency either for retarded and split injections. The graph shows that at 350 bar injection pressure split injection with 5% EGR exhibits 32.1% brake thermal efficiency at full load condition. While SI system exhibits 34%brake thermal efficiency at 350 bar injection pressure. This shows that at low EGR flow rates of 5% split injection is exhibiting better brake thermal efficiency while its brake thermal efficiency deteriorates at higher EGR flow rates of 10%. At higher EGR flow rates brake thermal efficiency of split injection decreases as compared to single injection. Hence the present work reveals that lower EGR flow rates is not performing better for multiple injections than single retarded injection for higher pressure injection system.

3.2. Brake specific fuel consumption

Fig. 3 shows the effect of EGR on bsfc at different injection pressures under retarded and split injection strategies as a function of engine load. Generally the Fig. 3 shows that increase in injection pressure from 200 to 350 bar decreases the specific fuel consumption for both single and split injections either with EGR or without

Table 2
The measuring range and accuracy of measurable parameters.

Measured qty.	Measuring range	Accuracy
Speed	0–2000 rpm	±1 rpm
Specific fuel consumption	0–3 kg/h	0.001 kg/h
Exhaust gas temperature	30–1000 °C	±1 °C
Load	0 kW-2.8 kW	±0.01 kW
Torque	0–30 Nm	0.01 Nm
In -cylinder gas temperature	20–2000 °C	0.01 °C
Ignition delay	0–20 deg CA	0.1 deg CA
Air-fuel ratio	0–100	0.01
Peak pressure	1–200 bar	0.01 bar
Average combustion duration (deg CA)	0–100 deg CA	0.1 deg CA
Averaged maximum rate of cylinder pressure rise	0–10 bar/deg	0.01 bar/deg
Averaged maximum rate of heat release	0–100 J/deg	0.01 J/deg
Crank angle	0–360 deg	0.1 deg

EGR. This is due to the improved spray atomisation and deep penetration of the fuel particles in to the combustion chamber. The results shows that higher EGR flow rates increased the bsfc for all injection pressures when the EGR flow rate increased from 5% to 10% for MPFI except for 200 bar at 0.5Full load conditions. This may be due to improper spray formation at part loads under low injection pressure. When the EGR flow rate is increased from 5% to 10% there is an increase in bsfc from 0.273 kg/kWhr to 0.281 kg/kWhr at 350 bar for MPFI system. An increase in 2.852% bsfc was noticed for MPFI when the EGR percentage is increased. This increase in bsfc is more when the engine is operated under no load conditions. However, the results shows that the maximum increase in bsfc was noted when the injection pressure is 200 bar less than that of design pressure of 210 bar. There is almost 12.83% increase in bsfc was noted for the injection pressure 200 bar for MPFI. This shows that MPFI is suitable for lower percentage of EGR circulation for better brake thermal efficiency for all injection pressures. Similar results are noticed for retarded SI injection with EGR circulation. Both MPFI and retarded SI injection noticed almost similar variation in bsfc with EGR circulation for all injection pressures when the engine is operated under varying load conditions.

3.3. Exhaust gas temperature

Fig. 4 shows that the variation of exhaust gas temperature with injection pressure for various EGR flow rates. Exhaust gas temperature increases with increase in load for different injection pressures. It can be observed that exhaust gas temperature decreases with increase in injection pressure from 200 to 350 bar for both retarded single injection and split injections with EGR. At full load it is found that the exhaust gas temperature decreases from 463 °C to 409 °C for MPFI when 5% flow rate. However the results shows that when EGR flow rate is increased from 5% to 10% it is noted that the Exhaust gas temperature starts increasing from 409 °C to 453 °C for the load ranging from 0% full load to 100% full load conditions at constant speed. Similar results are also noted for single retarded injection with EGR as seen in Fig. 4. Hence it can be concluded that lower EGR flow rates decreased the exhaust gas temperature and when EGR flow rate is increased to 10% it is noted that the exhaust gas temperature starts increasing more than that of without EGR. The result shows that 5% EGR flow rate has shown minimum exhaust gas temperature for all injection pressures. This shows that higher EGR flow rates promote ignition delay resulting high combustion temperatures. Compared to single retarded injection MPFI has shown higher exhaust gas temperature without EGR. MPFI is successful in reducing the exhaust gas temperature compared to retarded SI injection. 10% EGR flow rate recorded 367 °C for MPFI at full load whereas retarded SI has shown 377 °C. Similarly at ¾ full load operation 10% EGR flow rate for SI operation at 350 bar pressure shown 304 °C while MPFI has recorded only 289 °C. This is due to the decrease in peak combustion pressure and better improvement in combustion [27].This big variation in diesel MPFI is due to the better fuel spray formation and improvement in diffused combustion. The early pilot injection promotes better fuel vaporization for the secondary main injection due to which the MPFI performance was enhanced. Hence it can be concluded that 5%EGR for MPFI injection is successful in reducing the exhaust gas temperature and it can be observed that higher flow EGR rates is unsuccessful in reducing the EGT.

3.4. Peak pressure

The variation of peak pressure with respect to crank angle at full load conditons are shown in Fig. 5. The experimental results shows that increase in EGR flow rates decreased the peak pressure during the combusiton. Simultaneously peak pressure occurence also advanced with increase in injection pressure. The results shows that as the injection pressures are increased MPFI has developed more peak pressure compared to SI injection for all injection pressures as seen in Fig. 5. Results shows that for SI injection peak pressures increases with increase in loads as shown in Fig. 6. For higher

Table 3
Sample calculation of total uncertainty (%) in measurement of various measurable parameters at 2.8 kW load for 350 bar injection pressure for split injection @10%EGR.

Measured Parameter	Test-1	Test-2	Test-3	Mean = $\bar{X}$	Variable Error with 95% confidence level = 2??	$\%U_{.95} = \frac{2\sigma*100}{\bar{X}}$	% Fixed error of Instrument (FEI)	$\%TotalUncertainty\ inmeasurement\ (TotalUncertainity) = \sqrt{U_{.95}{}^2 + FEI^2}$
SFC	0.268	0.269	0.27	0.27	0.002	0.607	0.5	0.79
Torque	17.95	18.33	18.14	18.14	0.310	1.710	0.1	1.71
In -cylinder Gas Temperature	1601.5	1606.3	1600.1	1602.63	5.310	0.331	0.5	0.60
Ignition Delay	5.25	5.37	5.34	5.32	0.102	1.917	0.1	1.92
Air-fuel ratio	24.9	25.4	24.9	25.07	0.471	1.881	0.1	1.88
Peak Pressure	48.5	49.5	49	49.00	0.816	1.666	0.1	1.67
Average combustion Duration (deg CA)	44	44.7	44.9	44.53	0.772	1.733	0.1	1.74
Averaged maximum rate of cylinder pressure rise	1.41	1.4	1.4	1.40	0.009	0.672	0.3	0.74
Averaged maximum rate of heat release	20.2	19.99	20.05	20.08	0.177	0.880	0.2	0.90
EGT (°C)	33.25	33.7	33.5	33.48	0.368	1.100	0.1	1.10

Table 4
Total uncertainty (%) in performance parameters and their relevant measurable parameters at 2.8 kW load for 350 bar injection pressure for split injection @10%EGR.

Parameter	Variable	Uncertainty in measurement or computation
Brake power	Torque, Speed	1.68%
BSFC	Brake Power, SFC	0.79%
BTE	Brake Power, SFC	0.57%

EGR flow rates SI injection reduced the peak pressures confirming that higher EGR flow rates are efficient in reducing the peak pressures for both SI and MPFI systems. However higher EGR flow rates could deteriorate the lub oil condition and increase in soot formation. Fig. 6 shows that there is an increase in peak pressure when the engine load increases at different injection pressures for both SI and Multiple injections. Increase in injection pressures also increases the peak pressures for both SI and multiple injections for all EGR flow rates. However MPFI injection without EGR circulation recorded higher peak pressures as compared to SI injection. EGR circulation reduced the peak pressures for both SI and multiple injections. Results show that for 10% EGR flow rate for multiple injections at full load conditions reduced the peak pressure by an amount of 14.2% as compared to without EGR. This may be attributed due to decrease in combustion temperature in presence of cooled EGR with multiple injections. Fig. 6 show that SI injection with 10% EGR flow rates decreased the peak pressure to 48 bar compared to 51 bar with 5%EGR flow rates at higher injection pressures. Higher EGR flow rates was able to reduce the combustion peak pressures by 5.8%.Hence It can be concluded that Multiple injection strategy with high EGR flow rates is effective in reducing the combustion peak pressure for all injection pressures compared to SI injections.

3.5. Rate of pressure rise

The effects of EGR on Incylinder pressure rise rate is shown in Fig. 7. Premixed dominated combustion is noted for MPFI for all injection pressures ranging from 200 bar to 350 bar. For example the maximum rate of peak pressure rise is observed for MPFI is 1.75 bar/deg as seen in Fig. 7. This improvement in premixed combustion increased the cylinder pressure reduces the ignition dealy. This is mainly due to the earlier pilot injection at −54°BTDC. The results are similar to the earlier results reported in fumigation method [28].Thus MPFI is a good technique which can be adopted by modifying the common rail injection system to single cylinder diesel engine. Also the MPFI technique for single cylinder diesel engine reduces the structure born noise due to the improvement in controlled combustion. Fig. 7 shows that high pressure injection of 350 bar for 10% EGR flow rate decreased the peak pressure in premixed combustion. This decrease in peak pressure is due to the decrease in heat release rate due to the cool flame combustion [29]. At the same it can be seen that SI combustion has improved the pemixed combustion at all injection pressures with EGR flow rates. The results show that higher EGR flow rates has adverse effect on premixed combustion for SI combustion. It can be seen that higher peak pressures in the premixed combustion is observed for 10%EGR where as 5% EGR flow rates has better premixed and

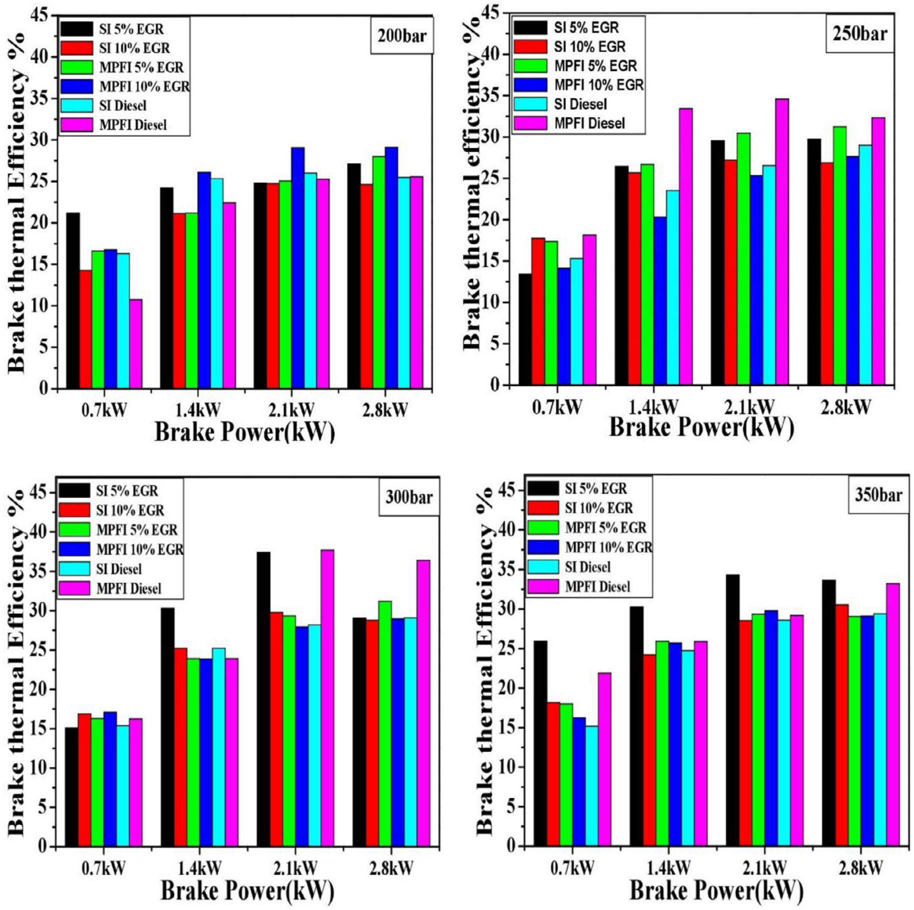

Fig. 2. Brake thermal Efficiency of SI and MPFI diesel combustion at varying injection presssures for at different EGR conditions.

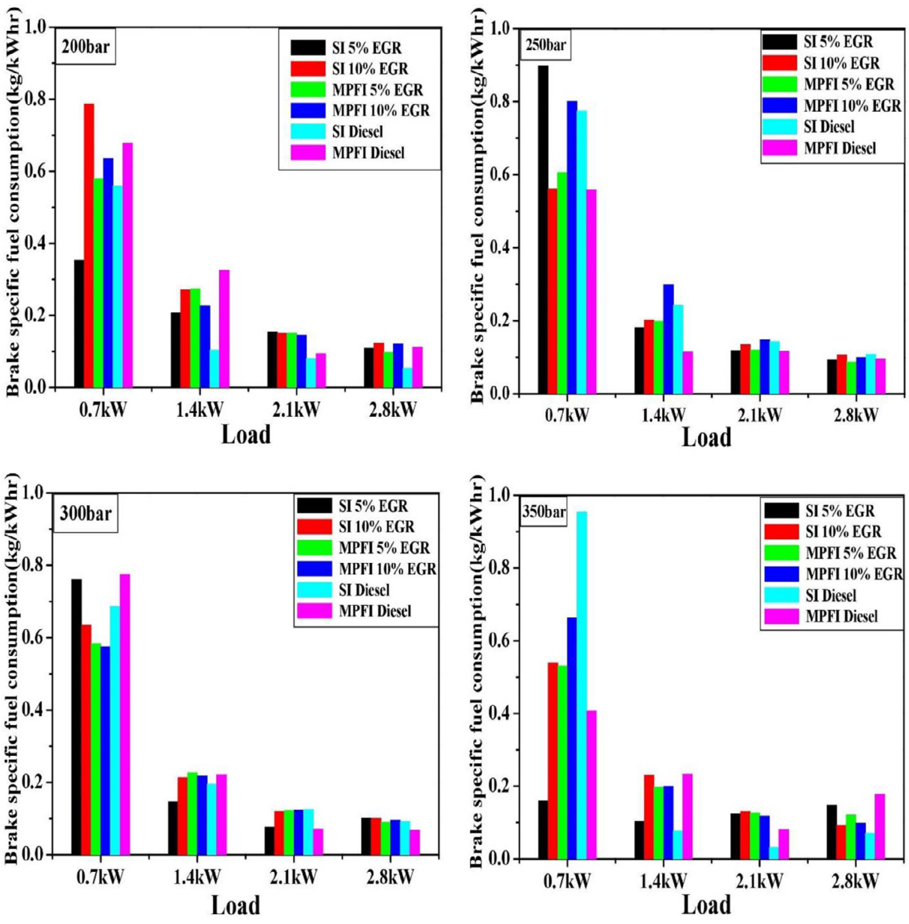

Fig. 3. Brake specific fuel consumption of SI and MPFI diesel combustion at varying injection presssures for at different EGR conditions.

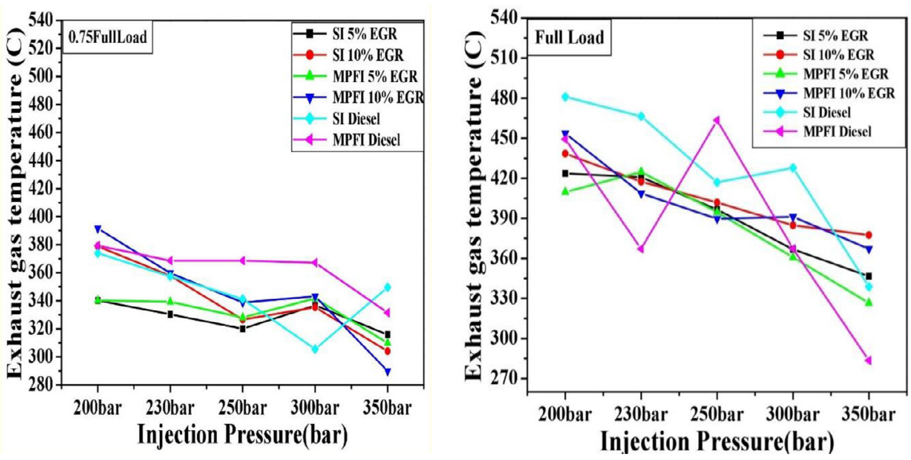

Fig. 4. Exhaust gas temperature of SI and MPFI diesel combustion at varying injection presssures for at different EGR conditions.

improved controlled combustion. Thus it can be concluded that SI combustion for higher pressure combustion is suitable for low EGR flow rates.

3.6. Ignition delay

The data presented for ignition delay relative to engine load at various different injection pressures are shown in Fig. 8. The variation of ignition delay for retarded SI at $-11°$BTDC and multiple injections at $-54°$ & $-11°$ BTDC for both 5% &10%EGR are presented in these Fig. 8. It can be seen that increase in injection pressures has marginal effect on ignition delay for both SI and multiple injections without EGR. Increase in injection pressures does not change the ignition delay significantly for all engine loads for both SI and multiple injections without EGR circulation. The results are in accordance with the earlier reports [30]. Experiment results show that increase in injection pressure decreased the ignition delay for SI injection up to 300 bar without EGR. However when the injection pressure increased to higher values of 350 bar there is an increase in ignition delay as shown in Fig. 8. Higher EGR flow

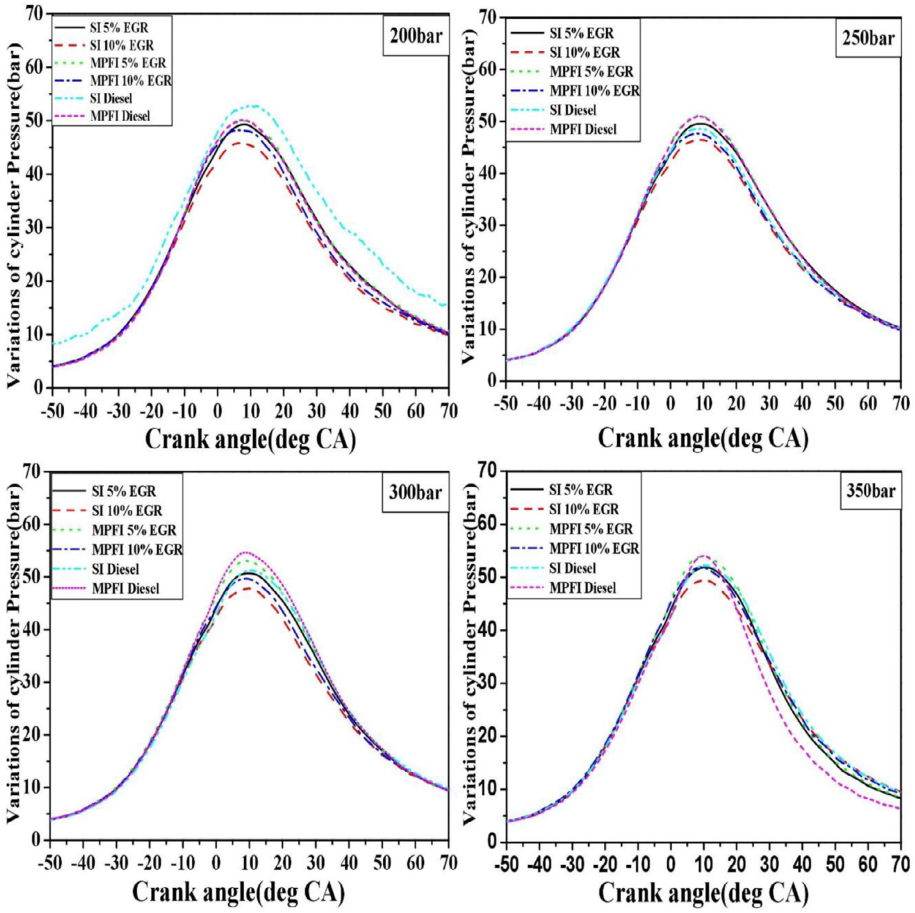

Fig. 5. In cylinder Pressure variation of SI and MPFI diesel combustion at varying injection presssures for at different EGR conditions.

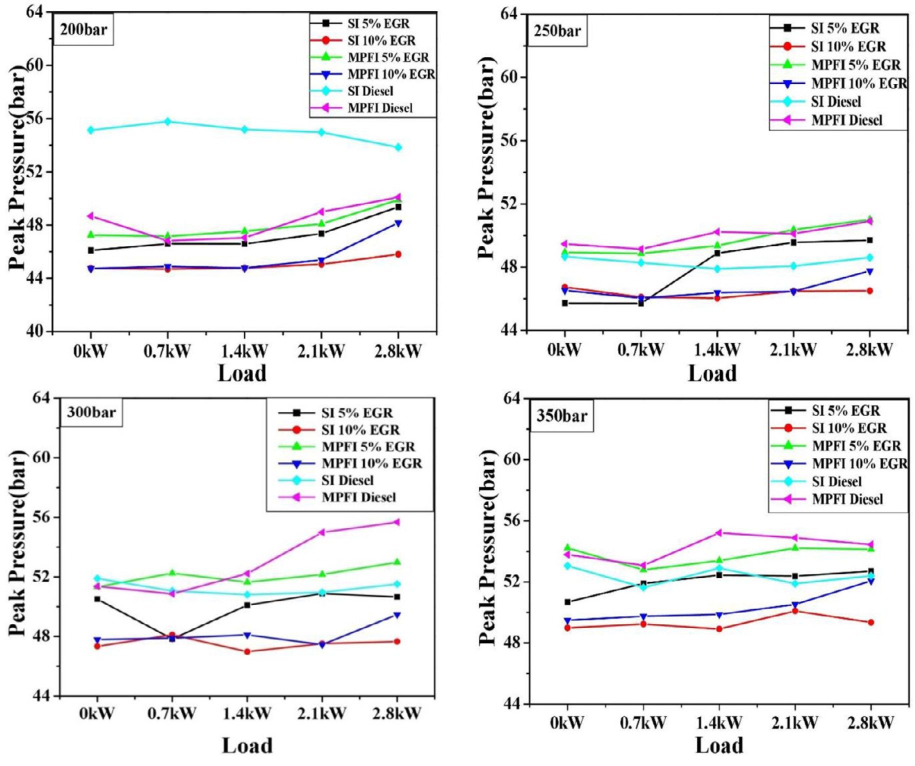

Fig. 6. Variation of peak pressure of SI and MPFI diesel combustion at varying injection presssures for at different EGR conditions.

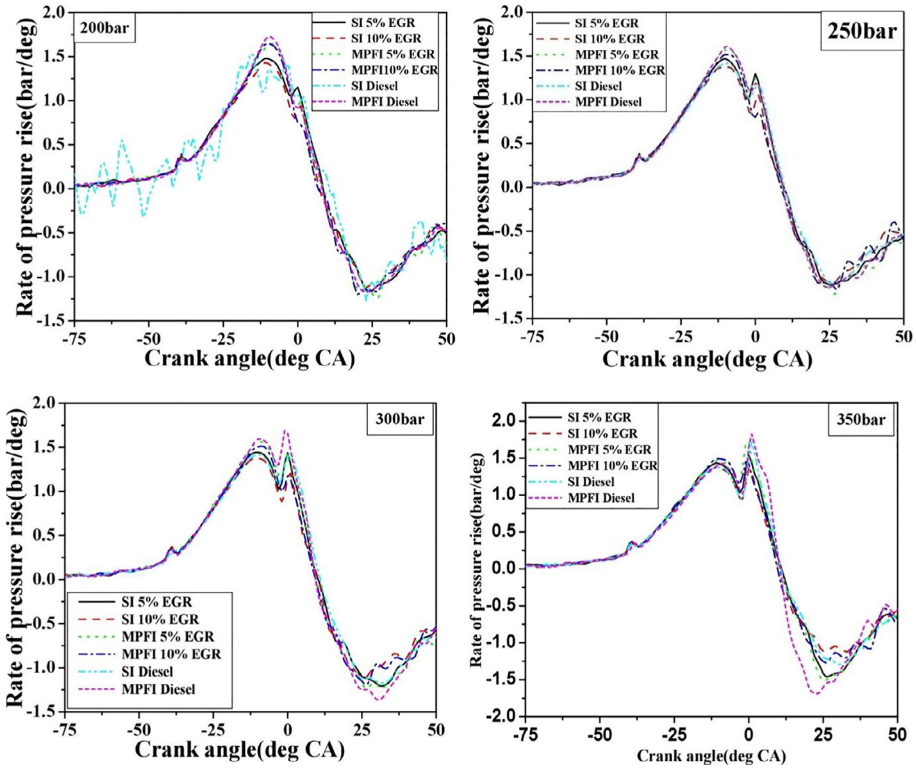

Fig. 7. Rate of pressure rise of SI and MPFI diesel combustion at varying injection presssures for at different EGR conditions.

rates increased the ignition delay for SI injection for all injection pressures. This is due to the less availability of air and decrease in temperature at SOI due to the higher percentage of cooled EGR. It can be seen that multiple injections with EGR reduced the ignition delay compared to SI injection with EGR for all injection pressures ranging from 200 bar to 350 bar. This can be attributed to the improvement in combustion process and equivalence ratios. Multiple injections without EGR have different effect on ignition delay when compared to SI injection. Increase in injection pressures from 200 bar to 350 bar for multiple injections without EGR shows that there is an increase in ignition delay as seen in Fig. 8. This behavior is quite opposite to the earlier results reported with SI injection. EGR has positive effect on multiple injections for increase in injection pressures as one can see that the ignition delay decreased with EGR circulation. This is due to the decrease in chemical delay since the SOI injection is fixed and also the engine speed is kept constant at 1500 rpm. Similar results are reported earlier that chemical delay is close to the ignition delay and physical delay is almost very short due to the rapid evaporation in high pressure split injection [31].

3.7. Combustion duration

The variation of combustion duration with increase in fuel injection pressure with various percentages of EGR for SI and multiple injections are shown in Fig. 9 at 0.75 load and full load engine operating conditions. In this study the duration between start of combustion and end of combustion is considered as combustion duration. MPFI injection exhibits lower combustion duration at high injection pressures also combustion duration decreases with increase in injection pressures as seen in Fig. 9. This is due to the better atomization of fuel particles at high injection pressures. At the same time in multiple injections 10% of fuel is injection at −54°CBTDC and remaining 90% fuel at retarded angle of −10°BTDC. Hence the pilot fuel will ignite first and the heat released during this period will enhance the vaporization of secondary fuel at −10°BTDC. Hence multiple injections reduce the combustion duration. It can been seen that increase in EGR rates increases the combustion duration due to the longer ignition delay periods. This is mainly due to the decrease in oxygen content and specific heat capacity of intake charge due to the change in molecular composition of intake mixture because of EGR [32]. It is observed that maximum decrease in combustion duration takes place at high injection pressures of 350 bar at 0.75 full load engine operating conditions. However slight increase in combustion duration is noted at full load conditions as compared to 0.75 full load. This is may be attributed to late ignition and decrease in brake thermal efficiency.

3.8. Heat release rate

Heat release rate is one of the important characteristic to study the combustion process in high pressure multiple injections with EGR circulation. Several peaks can be seen in HRR graphs in Fig. 10. EGR is used to reduce the higher peak values of HRR in high pressure multiple injection with advanced fuel injection strategies. MPFI heat release rate curves has 4 peaks showing the small peaks at −38° and −11° fuel injections while the remaining two high secondary peaks shows the low temperature and high temperature zones.

In multiple injection the pilot fuel is injected at 54° BTDC but the SOC is initiated at 38^0 at high injection pressure of 350 bar. This retarded SOC is due to the EGR circlation. Similar results are also reported in HCCI combustion using biodiesel with external mixture formation technique [8]. This delay can be attributed to chemical delay. Retarted injection without EGR has highest HRR peak as seen in all Fig. 10.

But EGR circulation reduced the peak HRR value and also the occurrence of HRR is shifted towards BTDC. This is because of decrease in cylinder temperature and decrease in combustion rate.

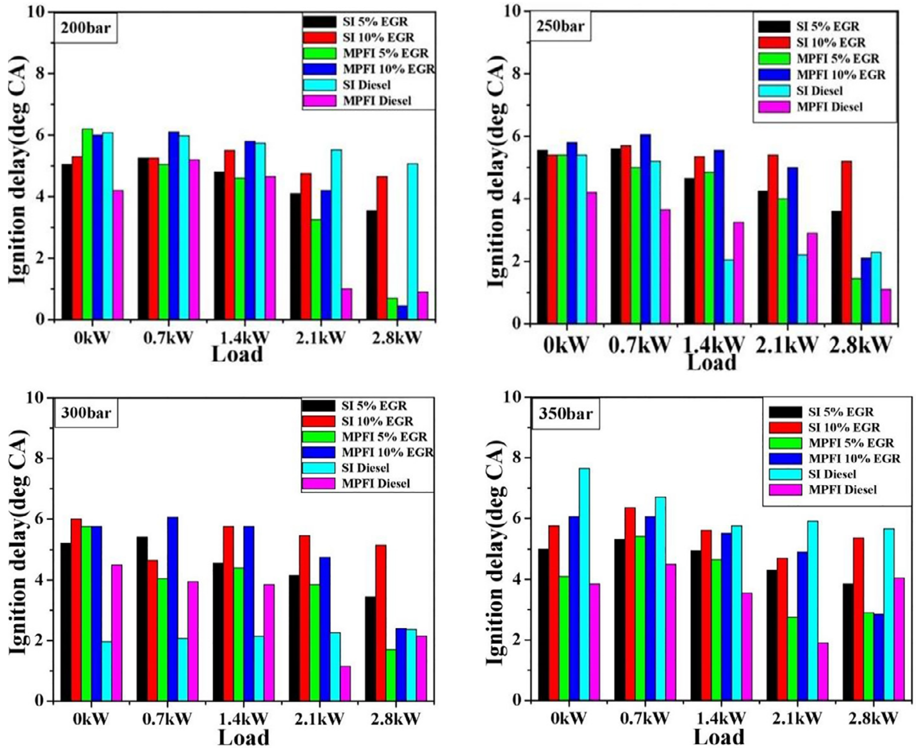

Fig. 8. Variation of Ignition delay of SI and MPFI diesel combustion at varying injection presssures for at different EGR conditions.

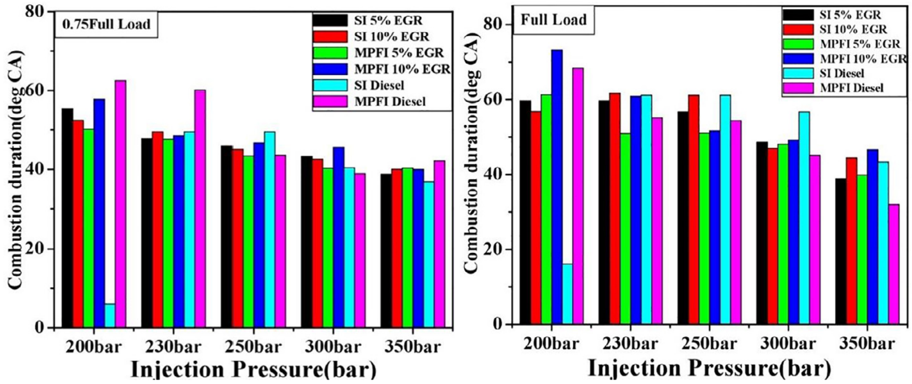

Fig. 9. Combustion duration of SI and MPFI diesel combustion at varying injection presssures for at different EGR conditions.

MPFI fuel injection reduced the peak value of HRR compared to single retarded injection showing that MPFI technique at high pressure injection is an effective tool in reducing the peak value of HRR. Thus High pressure MPFI technique is an effective tool to control the combustion rate and improves the combustion process and safety of the engine structure when operated at high pressure fuel injection. High EGR flow rates reduced the HRR for both SI and MPFI techniques. This is because of absorbing large amount of heat released during combustion by the non reactive mixture components in the cold EGR [33]. Higher EGR flow rates improved the premixed combustion and decreased the diffusion combustion as seen in above figs. Similar results are also reported in earlier discussion in combustion duration as seen in Fig. 9. Hence the combustion duration decreases with increase in EGR percentages for high pressure fuel injections. As a result 10% EGR has different HRR compared to 5%EGR for both retarted and multiple injections.

3.9. Cylinder gas temperature

In Fig. 11 shows the variation of cylinder gas temperature for no load and full load engine operating conditions. It can be seen that cylinder gas temperature increases with increase in engine load due to the increase in fuel quantity. EGR percentage and fuel injection pressures are the two important parameters which effect the cylinder gas temperature [34]. It can be seen that EGR increased the incylinder gas temperature for both SI and MPFI injections for all injection pressures ranging from 200 bar to 350 bars. This is due to the increase in igntion delay due to the EGR circulation. However the cylinder gas temperature can be reduced by increasing the EGR percentages as seen in Fig. 11. This is due to the charge dilution and less intense cylinder conditons such as less oxygen availability, decrease in chemical reaction rate, increase in specific heat of reactant mixture. Similar results are also reported in HCCI engine fueled with natural gas and diesel [35].

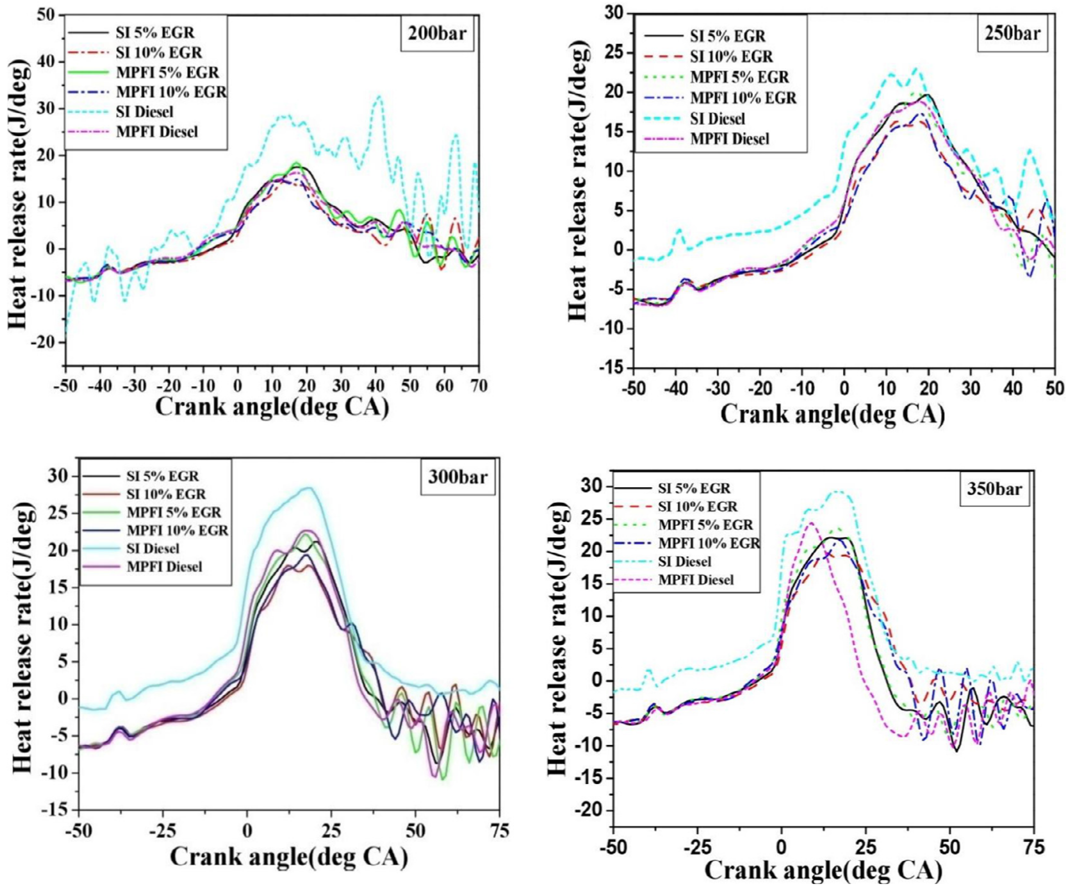

Fig. 10. Heat release rate of SI and MPFI at different injection presssures.

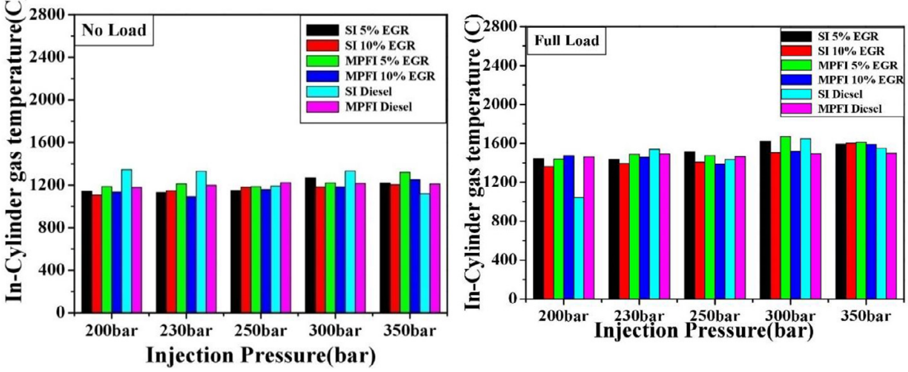

Fig. 11. Incylinder gas temperature of SI and MPFI diesel combustion at varying injection presssures for at different EGR conditions.

3.10. Air – Fuel ratio

The effect of air-fuel ratio on EGR in high pressure SI and MPFI strategy at various loads on single cylinder engine running at constant speed of 1500 rpm is shown in Fig. 12. The variation of A/F is almost linear with engine load for different injection pressures and EGR flow rates for both SI and MPFI. The graphs show that increase in engine load decreases the Air – Fuel ratio for all injection pressures for both SI and MPFI engines. It is desired that the engine opertes at lean mixture at full load operating conditons. The fuel injection pressure, EGR percentages and fuel injection stratagies and engine load plays a significant role in controlling the emissions and improve the engne performance. Increase in EGR rate from 5% 10% shows that air fuel ratio decreases by 20% as seen in Fig. 12 at full load condition for MPFI at 350 bar injection pressure. Hence MPFI is so sensitive for EGR circulation such that a small change in EGR rate has pronounced effect on engine emissions, while at constant EGR rate SI injection has noticed higher Air – Fuel ratio compared to MPFI injection at 0.75 and 1.0 full load condtions. Thus it can be concluded that NO_xemissions released by SI could be higher compared to MPFI injection at all loads under high pressure fuel injections. Higher EGR flow rates decreased the Air-fuel ratio for all operating engine load condtions. This is due to the replacement of fresh air with EGR [36,37]. Since at full load operating conditions the intake charge dilution should be minimised to keep NO_x levels at minimum. However smoke levels in the

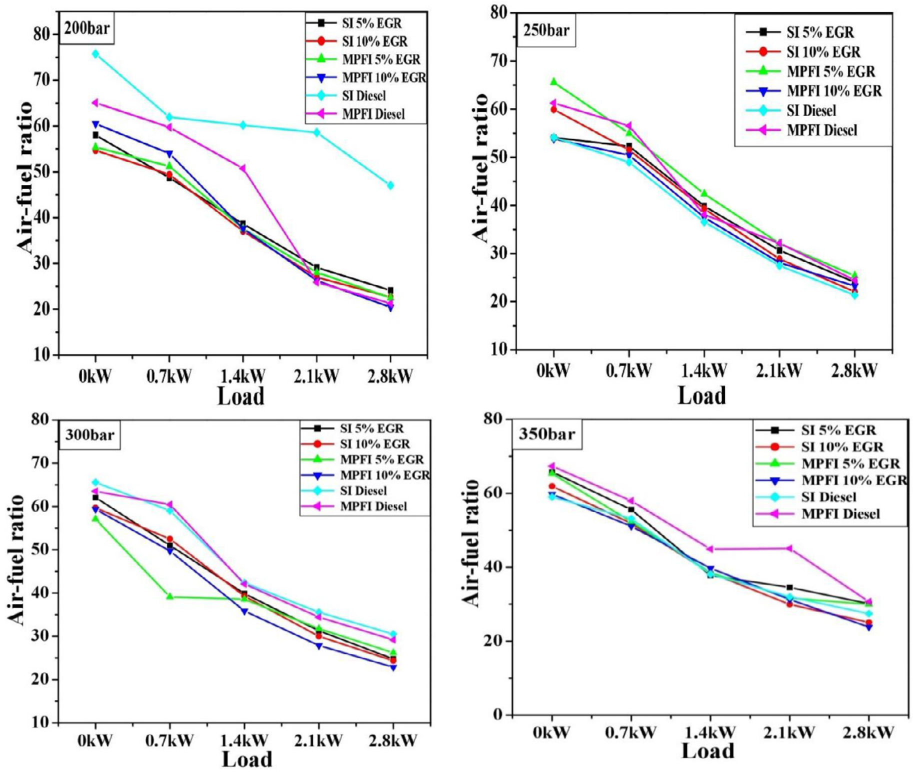

Fig. 12. Air-fuel ratio of SI and MPFI diesel combustion at varying injection presssures for at different EGR conditions.

engine can be kept minimum by diluting the incylinder charge at full load conditons with EGR. This requires a balance between air fuel ratio, fuel injection pressure and combustion peak pressure. To achieve this retarted injection, high pressure injection, multiple split injection is tested in the present work. The results shows that MPFI technique with 10%EGR rate is effective in comparting to retarded SI combustion.

4. Conclusions

The conventional direct injection single cylinder diesel engine is modified to run on high pressure fuel injection system in which fuel is injected by two strategies retarded injection (SI) and split injections (MPFI). The ultra-cooled EGR is circulated with different flow rates with high pressure fuel injected by modified common rail direct injection (CRDI) system with electronic timing injection kit. The results show that MPFI system is most suitable for engine running on design pressure and high injection pressures of 350 bar. Simultaneously for moderate and medium high pressures of 50% higher than the design pressures retarded injection SI system is having better performance than MPFI system. The following conclusions are obtained:

1) There is no much variation in brake thermal efficiency for higher EGR flow rates for all injection strategies. 5% EGR flow rate is exhibiting better brake thermal efficiency compared to MPFI. Split injection with 5%EGR flow rate the maximum thermal efficiency is 32.1% while SI system exhibits 34% brake thermal efficiency at 350 bar injection pressure. This shows that MPFI system exhibits marginal decrease in brake thermal efficiency compared to SI injection.
2) Bsfc increased from 0.273 kg/kWhr to 0.281 kg/kWhr at full load condition for 350 bar injection pressure under MPFI condition when the EGR percentage increased from 5 to 10%. There is almost 2.852% increase in bsfc is observed for MPFI when the EGR percentage is increased. Hence it is concluded that MPFI is suitable for lower percentage of EGR circulation for better brake thermal efficiency for all injection pressures.
3) MPFI is successful in reducing the exhaust gas temperature compared to retarded SI injection. 10% EGR flow rate recorded 367 °C for MPFI at full load whereas retarded SI has shown 377 °C. Combustion analysis show that higher EGR flow rates was able to reduce the combustion peak pressures by 5.8%.Hence It can be concluded that Multiple injection strategy with high EGR flow rates is effective in reducing the combustion peak pressure for all injection pressures compared to SI injections.
4) EGR has positive effect on multiple injections for increase in injection pressures as one can see that the ignition delay decreased with EGR circulation. MPFI injection exhibits lower combustion duration at high injection pressures also combustion duration decreases with increase in injection pressures.
5) MPFI heat release rate curves has 4 peaks showing the small peaks at −38° and −11° fuel injections while the remaining two high secondary peaks shows the low temperature and high temperature zones. Regarding HRR MPFI fuel injection reduced the peak value of HRR compared to single retarded injection showing that MPFI technique at high pressure injection is an effective tool in reducing the peak value of HRR.
6) Thus High pressure MPFI technique is an effective tool to control the combustion rate and improves the combustion process and safety of the engine structure when operated at high pressure fuel injection. EGR has effect on Air – Fuel ratio. It was noticed that Increase in EGR rate from 5% 10% shows that air fuel ratio decreases by 20%.

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Appendix A. Supplementary data

Supplementary data to this article can be found online at https://doi.org/10.1016/j.jestch.2019.01.013.

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