A COMPARATIVE EXPERIMENTAL STUDY ON ENGINE OPERATING ON PREMIXED CHARGE COMPRESSION IGNITION AND COMPRESSION IGNITION MODE

by

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New combustion concepts have been recently developed with the purpose to tackle the problem of high emissions level of traditional direct injection Diesel engines. A good example is the premixed charge compression ignition combustion. A strategy in which early injection is used causing a burning process in which the fuel burns in the premixed condition. In compression ignition engines, soot (particulate matter) and NO_x *emissions are an extremely unsolved issue. Premixed charge compression ignition is one of the most promising solutions that combine the advantages of both spark ignition and compression ignition combustion modes. It gives thermal efficiency close to the compression ignition engines and resolves the associated issues of high* NO_x *and particulate matter, simultaneously. Premixing of air and fuel preparation is the challenging part to achieve premixed charge compression ignition combustion. In the present experimental study a diesel vaporizer is used to achieve premixed charge compression ignition combustion. A vaporized diesel fuel was mixed with the air to form premixed charge and inducted into the cylinder during the intake stroke. Low diesel volatility remains the main obstacle in preparing premixed air-fuel mixture. Exhaust gas re-circulation can be used to control the rate of heat release. The objective of this study is to reduce exhaust emission levels with maintaining thermal efficiency close to compression ignition engine.*

Key words: *carbon monoxide, exhaust gas recirculation, hydrocarbons, oxides of nitrogen, premixed charge compression ignition engine, particulate matter*

Introduction

Recent research and development has made major reductions in engine emissions but a growing population and a greater number of automobiles mean that the problem will exists for many years to come. Two methods are being used to reduce harmful engine emissions. One method is after-treatment of exhaust gas. This is done by using thermal or catalytic convertors that promotes chemical reactions in exhaust flow. The other method is to improve the technology of the engine and fuel so that better combustion occurs and fewer emissions are generated [1, 2]. One such technology discussed in this paper is premixed charge compression ignition (PCCI) engine. Simescu *et al.* [3] conducted an experimental investigation of premixed charge compression ignition-direct injection (PCCI-DI) combustion coupled with cooled

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and uncooled exhaust gas re-circulation (EGR) in heavy duty Diesel engine with port fuel injector and the study showed significant NO_x reduction at light load conditions. The study however showed that early PCCI combustion could adversely affect NO_x emissions by increasing in-cylinder temperatures at the start of diffusion combustion. The PCCI-DI combustion also showed increased brake specific fuel consumption and HC, CO, and particulate matter (PM) emissions.

Midlam-Mohler *et al.* [4] has developed an atomizer for external mixture preparation and the authors have investigated the effect of uncooled EGR, boost pressure, air-fuel ratio, intake air temperature, swirl, and engine speed on homogenous charge compression ignition combustion. In PCCI combustion strategy, fuel can be introduced into the combustion chamber through port fuel injection, early DI and late DI [5]. Port fuel injection and early DI often suffers from incomplete fuel vaporization. However, early cycle charge densities are often low which can results in a very long liquid lengths and spray impingement on cylinder liner causes high level of HC and CO as well as fuel-oil dilution [6, 7]. Many researchers have addressed the issue of spray wall impingement associated with PCCI strategies. Kawaguchi [8] proposed an injector capable of changing the cone angle depending on a piston position in order to maintain optimal spray targeting over a wide range of injection timing. Sun and Reitz [9] showed the advantages of using low pressure injection for fuel injected early in the cycle and high pressure injection of fuel injected late in the cycle.

Naoya [10] attempted the control of ignition timing and suppression of rapid combustion in PCCI engine with in cylinder injection of water as a reaction suppressor. Expansion of viable operating range was achieved by optimization of timing and quantities of water injected. Light naphtha characterized by high velocity and sufficient self-ignitability was introduced in the intake manifold of PCCI engine, allowing sufficient time to form a quasi-homogeneous mixture absent of fuel rich zones. The compression ratio (CR) was set to the same level of ordinary compression ignition (CI) engines, to prevent misfiring and increase in CO and total HC as well as to maintain high thermal efficiency. The optimization of fuel injection conditions and combustion chamber geometry is one of the most important issue for realizing low CO and HC emissions as well as low NO_x and smoke emissions [11-13]. The PCCI shows lowest fuel consumption and NO_x emission but higher HC emission [14]. The PCCI potentially achieves ultra-lean burning under higher CR than possible in conventional spark ignition engines and elimination of NO_x and PM to an extent not possible in an ordinary CI engine [15, 16]. In the present experimental study, a premixed mixture of fuel and air was prepared by using diesel fuel vaporizer [17-23]. The diesel vapor provided by this device forms a very light and dispersed aerosol, where due to their sizes, the droplets lose their momentum a short distance downstream of the nozzle follow the air motion very well. They also have very fast evaporation due to very high surface to volume ratio, and disperse very uniformly in the surrounding air stream. For Diesel engines perfect homogeneous mixture of air and fuel is not possible due to poor volatile characteristics of diesel fuel. Hence the concept is named as PCCI has been introduced. In PCCI the fuel is dispersed into the intake manifold prior to intake. In this case, external heating is required to elevate the air or fuel temperature for partially homogeneous mixture formation inside the intake manifold.

Experimental procedure

The experimental set-up used in this investigation is shown in fig. 1. It consists of a two cylinders, 4-stroke, constant speed, water cooled, DICI engine with hemispherical open combustion chamber. For the engine to run on conventional CI mode of operation, the fuel

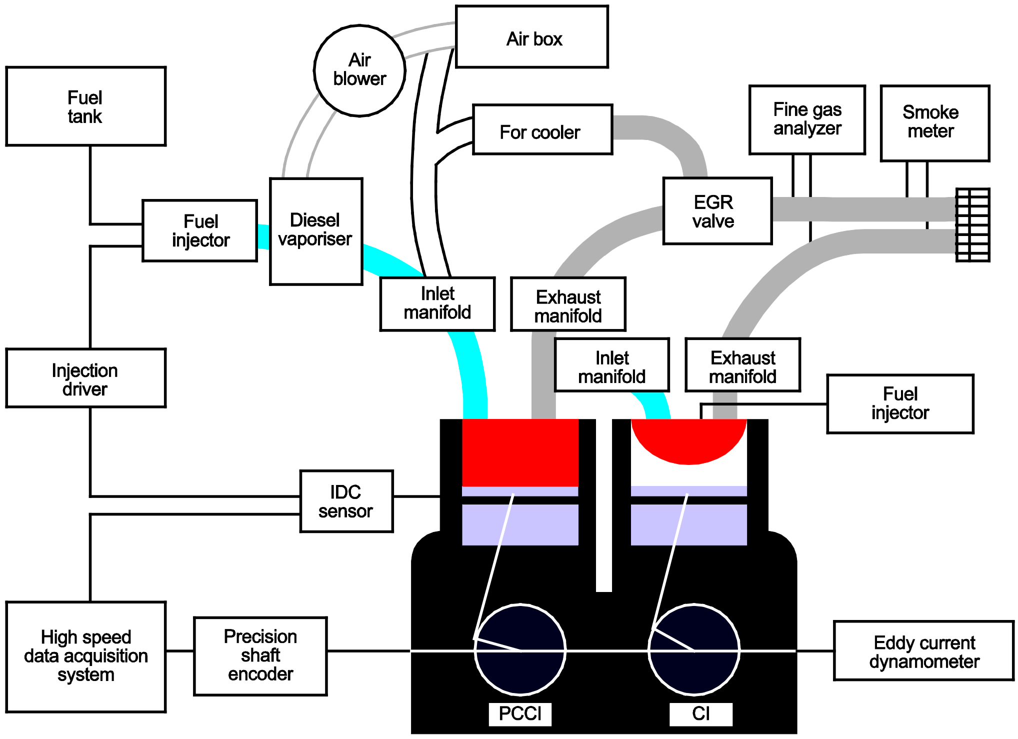

Figure 1. Schematic representation of PCCI engine

Table 1. Engine specification

Manufacturer	Kirloskar engines Ltd., Pune, India
Engine type	4-stroke, 2 cylinder, CI
Rated power	10.3 kW @1500 rpm
Bore	87.5 mm
Stroke	110 mm
Swept volume	1320 cm^3
Compression ratio	16.5
Mode of injection	Direct injection
Cooling system	Water
Dynamometer	Eddy current dynamometer

injection system of the engine comprised of a plunger type pump with an injector having three spray holes, each 0.28 mm diameter. The injector needle lift pressure and fuel injection timing of the engine are 210 bar and 27° before top dead center (bTDC), respectively. The specifications of the engine are shown in tab. 1. A twin cylinder 4-stroke water cooled Diesel engine was coupled to an eddy current dynamometer with a load cell. The in-cylinder pressure was measured by piezoelectric pressure transducer fitted on the engine cylinder head. A crank angle encoder was used to sense the crank position. Exhaust gas analysis was performed using five gas exhaust analyzer. A Hartridge smoke meter was attached to exhaust pipe to measure smoke levels.

The engine was operated in conventional CI mode and PCCI mode with diesel as fuel. In both modes, experiments were conducted at variable load at rated speed 1500 rpm. Initially the engine was started at no load condition in CI mode. The engine can directly not be operated on PCCI mode because of low charge temperature (low volatility of diesel may remain the main obstacle). Once engine achieved warm-up condition in CI mode, the engine governor cuts the fuel supply to the conventional fuel injector (210 bar pressure) thereon the engine operated completely in premixed diesel vapor-air mixture. To inject the fuel into the vaporizer, the cut-off temperature of the vaporizer maintained at 140 ±50 °C which provides the diesel vapor into the intake manifold. In the intake manifold the diesel vapor mixed with air to form a premixed mixture. This premixed mixture was inducted in the engine cylinder during the suction stroke. Thus ensures that engine switched to diesel PCCI mode [24]. A diesel vaporizer consists of a heating element, M. S. pipe of 20 mm diameter. The length of the diesel vaporizer was 165 mm. The port fuel injector (up to 4 bar) was mounted on the top of the diesel vaporizer to supply the correct quantity of fuel to the vaporizer. The port fuel injector was controlled by an electronic control unit (ECU). The ECU controls both the timing and quantity of the fuel. The ECU controlled low pressure fuel injector that injects the pre-determined quantity of diesel into the diesel vaporizer. The fuel vaporizer was heated through a power supply. A temperature controller was used to maintain the temperature to provide diesel vapor at all the loads. A low pressure electrical pump was used to supply diesel to the low pressure injector. Both the pump and the injector were controlled by ECU. At each load air flow rate, fuel flow rate, exhaust gas temperature, HC, CO, NO_x, and smoke emissions were recorded [25].

Experimental set-up

The experimental engine is designed for 60 Nm torque at 1500 rpm and 10.3 kW power for both cylinder in operation. The one of the cylinders operate on CI mode of operation generates maximum 5 kW power at 30 Nm torque when fuel supply of the cylinder modified to PCCI mode is cut. The same procedure is followed when the engine is operating on PCCI mode. A water cooled pressure transducer is used to measure in-cylinder pressure of the engine. Provision is made to measure in-cylinder pressure of the both cylinders. The pressure transducer is used to measure cylinder pressure in either of the engine cylinders which is working. Both cylinders are not active at a time.

Figure 1 representation of PCCI engine. Engine experiments were performed in a two cylinders, 4-stroke, constant-speed, water-cooled, DI Diesel engine with eddy current dynamometer. In this engine, one cylinder is modified to operate in PCCI combustion mode, while the other cylinder operates in conventional diesel combustion mode.

The main hurdle behind the formation of premixed mixture in the intake port was the low volatility of diesel. Therefore, diesel vaporizer is a device used to tackle out the problem. The fuel injector sprays atomized fuel into heated diesel vaporizer chamber. The fuel droplets absorb latent heat of vaporization. High velocity air was supplied from the blower to forces these fuels vapor containing tiny fuel droplets to mix homogeneously. The air supplied from the air box serves two purposes. One is to push diesel vapors from the vaporizer using the air blast and another purpose is to supply the air to the engine intake manifold. Figure 2 shows the diesel vaporizer with modified intake manifold.

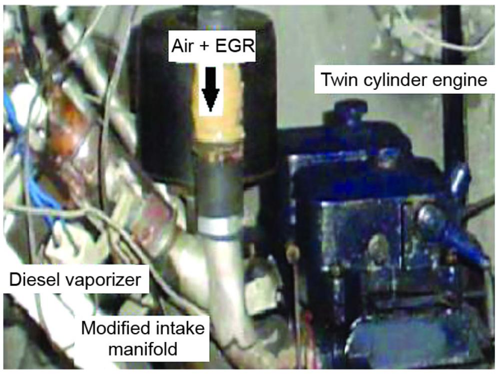

Figure 1. Diesel vaporizer

The EGR [26-28] is possible way to reduce the in-cylinder temperature. The EGR reduces the concentration of oxygen, so that for each mole of oxygen consumed, there are more moles nitrogen, water vapors, and CO_2 to be heated by the heat of reaction released from an equivalent amount of fuel. The temperature rises with EGR is also limited by the higher specific heat of water vapor and CO_2 which absorbs more heat than NO_2. The lower flame temperature with EGR decreases the amount of NO_x production. However, the amount of oxygen entrainment in the fuel jet will decreases as the oxygen concentration decreases with EGR. This decrease entrainment of oxygen could lead to production of more carbonaceous particles. In this study EGR was varied (10, 20, and 30%) and its effects on the engine performance, emission, and combustion were studied. The EGR mixes with air in the intake manifold. The maximum 30% of EGR is used during the engine operation. The air-fuel mixture was supplied to the combustion chamber through the intake valve located in inlet manifold. The fuel-air mixture used in PCCI engine is lean mixture which operates between λ = 4.8, above which combustion deteriorates and λ = 2.6, below which engine knocks. Fuel injection system of PCCI mode Diesel engine consists of a fuel injector, fuel pump, fuel tank, fuel accumulator, and an injector control circuit. A 12V DC electrical fuel pump installed inside the fuel tank to supply diesel from the fuel tank to the fuel accumulator. The fuel injection control circuit takes input from the TDC sensor to determine the *start of fuel injection* and to control injection quantity.

Results and discussion

The variation of brake thermal efficiency with load is shown in fig. 3 with 0, 10, 20, and 30% EGR. It is observed from the figure that the brake thermal efficiency decreases with increase in EGR percentage for PCCI mode compared to DICI mode of operation. The decrease in brake thermal efficiency is about 1.9, 5.8, 9, and 12.6%, respectively, for 0, 10, 20, and 30 EGR compared to that of DICI mode of operation. As the mixture is formed externally, combustion timing can only be influenced by diluting the cylinder charge with exhaust gas. Premixed combustion with its steeply rising pressure produces high combustion noise to avoid this and the charge must be highly diluted using EGR. As the proportion of cooled EGR increases, decrease in brake thermal efficiency is observed due to energy loss of the vaporizers and the losses due to the unburned fuel. Other reasons may be due to increase in HC/CO emission.

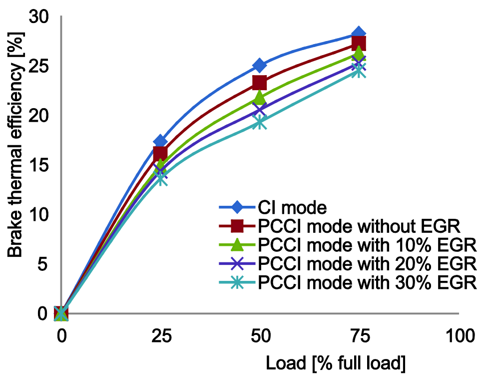

Figure 3. Variation in load *vs.* brake thermal efficiency

The parameter responsible for higher HC and CO emissions was low temperature combustion due to lean mixture and high EGR necessary to achieve PCCI operation. The EGR has two primary effects on HC emission: one is that the intake of some unburnt HC with exhausted gas into the next cycle leads to decrease in HC emissions, the other one is that the decrease of combustion temperature in the cylinder leads to an increase in HC emissions. Figures 4 and 5 shows the variation of HC and CO with load. In PCCI mode of operation, the HC and CO emissions are 30 times higher than the conventional CI mode of operation [29, 30]. The HC emissions are 0.51, 0.59, 0.69, and 0.81 g/kWh for PCCI mode of operation with 0, 10, 20, and 30% EGR rates. For CI mode of operations the HC emission is measures as 0.25 g/kWh.

With increase in EGR rate, the combustion reaction rate is reduced, the mean temperature in the cylinder is decreased, and the combustion reaction becomes more incomplete, the reason is that more and more mid product CO can not be oxidized completely into CO_2 because of the decrease in temperature. Figure 5 shows the variation of CO emission with load. The CO emissions are 4.32, 5.06, 6.3, and 7.5 g/kWh for PCCI mode with 0, 10, 20, and 30% EGR, whereas in CI mode it is about 1.28 g/kWh.

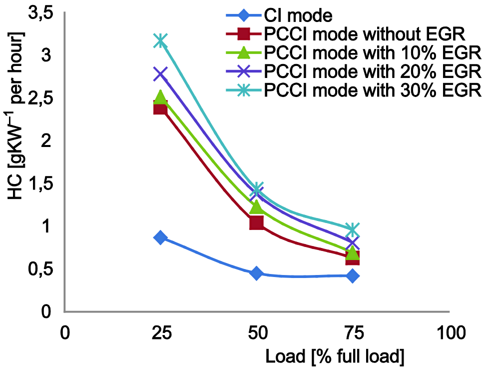

Figure 4. Variation of unburned HC *vs.* load

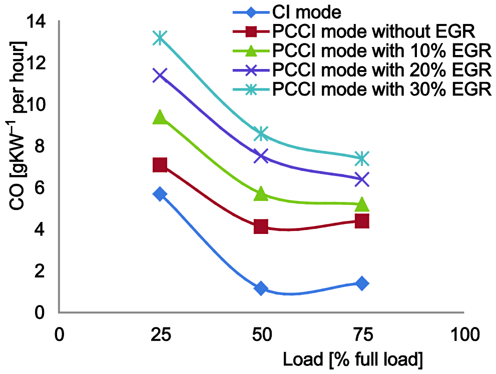

Figure 5. Variation of CO *vs.* load

Figure 6 shows variation of NO_x with load in both modes of operation using diesel as fuel. In the CI engine, the NO is formed in very hot zones closer to stoichiometric conditions. The in-cylinder average air-fuel ratio is globally always lean but locally the combustion process is not. This means that there is a large potential to reduce emissions of NO_x by simply mixing fuel and air before combustion. The PCCI combustion can reduce NO_x emissions up to 90-98% as compared to CI mode of operation. When cooled EGR was inducted with premixed diesel vapor-air mixture NO_x emissions further reduces due to lower combustion temperature and pressure. The combustion products in EGR such as CO_2 and H_2O have a higher specific heat per unit mass than air. Hence it reduces the combustion temperature and chemical kinetic reaction rate. The EGR role is very crucial in controlling combustion phase as well as the rapid rise in cylinder pressure in PCCI combustion. The figure clearly shows that the NO_x emission reduces by 45, 80, 86, and 95%, respectively, for PCCI mode with 0, 10, 20, and 30% EGR with that of the conventional CI mode of combustion.

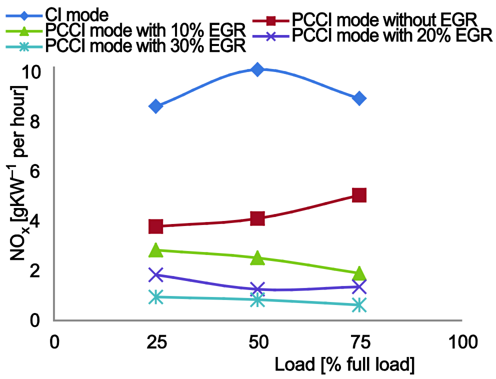

Figure 6. Variation of NO_x *vs.* load

Figure 7 shows the smoke emission behavior of the engine operated with PCCI mode compared to conventional CI mode of operation. Smoke emission is another attraction of PCCI combustion. However, Diesel PCCI engine smoke reduction is dependent on efficient control of EGR. From the figure it can be observed that the engine operated with PCCI mode exhibits a significant reduction in smoke at all loads. Smoke and NO_x emission rise significantly as charge heterogeneity increases. However, mixture is available inside the cylinder as a homogeneous mixture. Hence there is an absolutely free from liquid fraction of fuel pockets, unlike heterogeneous combustion that takes place in a conventional DI Diesel engine. The reduction in smoke emission is observed to be about 39, 56, 78, and 83%, respectively, for PCCI mode of operation with 0, 10, 20, and 30% EGR compared to that of conventional CI mode of operation.

In the present experimentation EGR was used to reduce the combustion temperature and pressure. As the mixture is formed externally, combustion timing can only be influenced by diluting the cylinder charge with exhaust gas or by altering the temperature of the mixture.

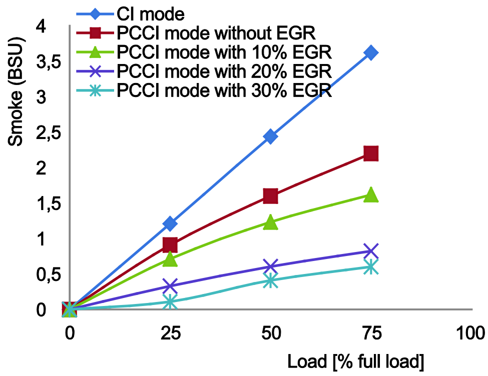

Figure 7. Variation of load *vs.* smoke

Comparison of peak pressure and heat release rate for PCCI mode with 0, 10, 20, and 30% EGR with conventional CI mode of operation is shown in figs. 8 and 9. The peak cylinder pressure occurs just after TDC and shifts away from TDC with increasing EGR rate. It happens because of delayed combustion due to mixture dilution by recirculated exhaust gas. As EGR rate increases, level of dilution also increases and hence delay period increases which shifts pressure curve away from TDC. Figure 9 shows that heat release rate decreases with increasing EGR rate due to reduction in rate of combustion. Both cool temperature reaction and hot temperature reactions get suppressed with increasing EGR rate. It happens due to reduction in peak pressure rise and cylinder mean temperature with increasing EGR rate. Net level of oxygen concentration decreases at high level of EGR that decelerate the rate of combustion.

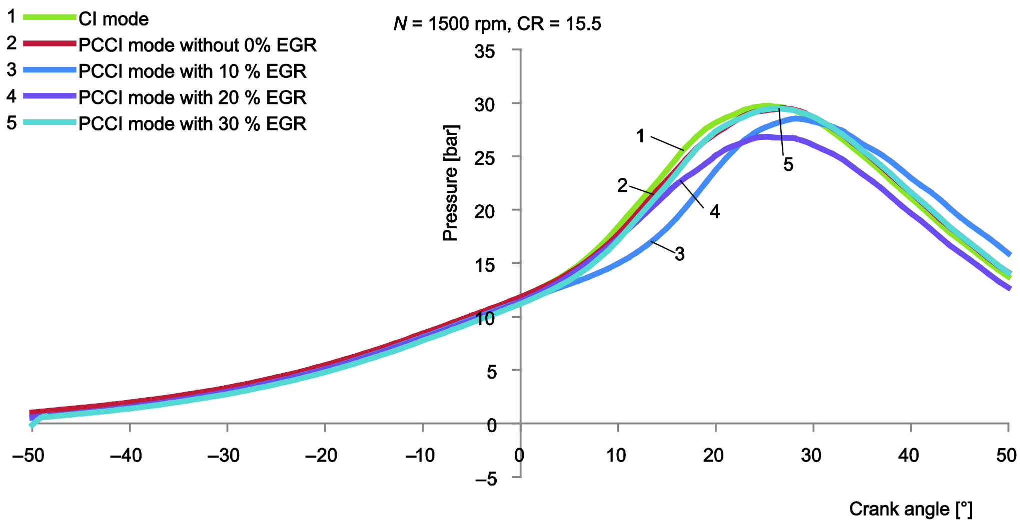

Figure 8. Variation of cylinder pressure *vs.* crank angle

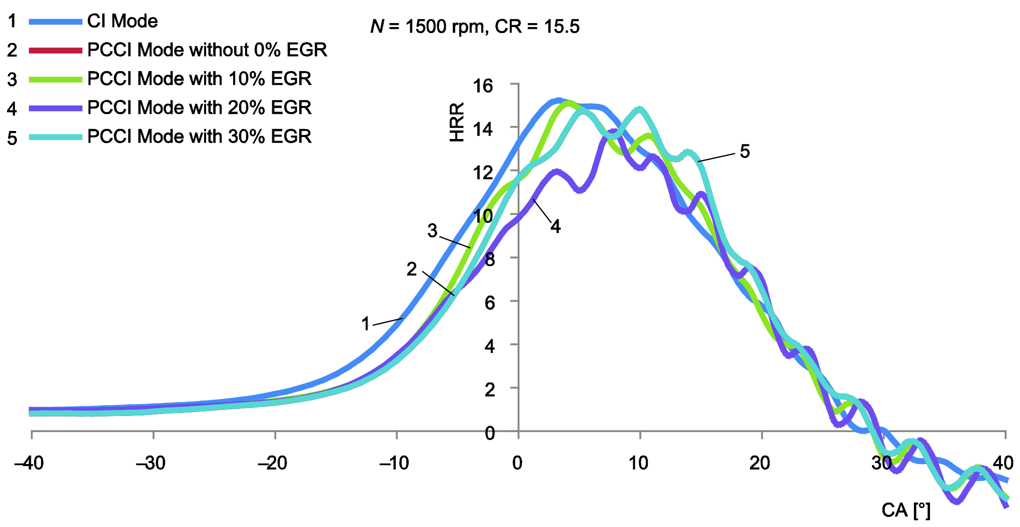

Figure 9. Heat release rate *vs.* crank angle

Conclusions

The PCCI is a promising concept for achieving low emissions at part load operations. This technique can be successfully applied to conventional DICI engines with low costs and fewer medications. However, high HC and CO emissions are major disadvantage. The performance and emissions characteristic of DICI engine were analyzed and compared in conventional CI mode and PCCI mode. There was found a reduction in brake thermal efficiency for PCCI mode by 8% compared to CI mode at part load with 0% EGR condition. It was observed that the HC and CO emissions were higher in PCCI mode than CI mode, because of the premixed charge and fuel spray wall impingement. There was found a reduction in NO_x emission by 13% for PCCI mode of operation with 30% EGR compared to CI mode because of low combustion temperature generated inside the cylinder.

In present experimental study, to control and delay the combustion in PCCI mode, EGR was used up to 30%. Using EGR the combustion was controlled and delayed. Further increase in EGR would lead to higher HC and CO emission and combustion instability. The conventional CI engine is converted into PCCI mode and it was observed from the experiments that NO_x and smoke emissions were very less when compared with conventional CI mode for limited operational area.

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