Effects of EGR on Diesel Engine Operating Characteristics under Different Engine Conditions

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Abstract

The use of exhaust gas recirculation (EGR) has been widely recognized as an effective way to control NO_x emissions in internal combustion engines. The current study examines the effects of EGR on diesel engine operations under two different steady-state conditions: a low-load operation at 2000 rpm and a medium-load operation at 2500 rpm. All experiments were performed in a four-cylinder, turbocharged direct injection engine.

Data emphasized that the use of EGR could reduce NO_x emissions significantly. A greater amount of EGR reduced the combustion temperatures, thus reducing the rate of NO_x formation. An excessive rate of EGR led to lower engine efficiency, considerably higher CO and slightly higher HC emissions. Although regulating a high EGR rate can result in greater CO and HC emissions, the higher engine-out exhaust temperature under low-load operations is favorable for the conversion efficiency of the diesel oxidation catalyst operation. The EGR amounts should be tuned for high thermal efficiency and acceptable emissions at different engine conditions. The findings help facilitating further investigations into engine improvements and modifications.

Keywords: diesel, compression ignition, EGR, NO_x, emissions, direct injection, internal combustion engines.

1. Introduction

Increasingly stringent emission regulations, diminishing fossil energy supply and enhanced power requirements have provoked interest in promising powertrain technologies like hybrid cars and fuel cell vehicles. However, they are still far from being completed and internal combustion (IC) engines will continue to remain as the most efficient and cheapest transportation system available for the foreseeable future. A nature consequence of this would be to continue our efforts in making the readily available technologies in IC engines better.

Conventional compression-ignition or diesel engines are promising for a capability to achieve high thermal efficiency operations. Diesel engines utilize high compression ratios, compared to those of spark-ignition (SI) engines. The load in diesel engines is controlled by regulating the amount of fuel which is directly injected into the cylinder. As such, the engines operate on the excess of air. Together with a higher compression-ratio operation, a diesel engine has a higher efficiency than an SI engine. Under normal conditions, diesel engines emit less hydrocarbons (HC) and carbon monoxide (CO).

In contrast, diesel engines suffer from high soot and oxides of nitrogen (NO_x) emissions. With the diffusion-like combustion characteristics in diesel engines, soot is formed due to pyrolysis in the fuel-rich zones (core of the fuel spray) [1]. Although soot formation is an important issue in diesel engine operations, it is beyond the scope of the current study.

It is recognized that NO_x are generated from high-temperature zones during the combustion [1-5]. One practical way to control NO_x generation is by using exhaust gas recirculation (EGR). From this aspect, the use of EGR was brought into our attention. The focus of current work is then the effects of EGR on diesel-engine characteristics under different operating conditions. The findings from this study will provide a better understanding of the nature of combustion characteristics and engine operating control. This information will facilitate further investigations into engine improvements and modifications.

2. Experimental Setup

All experiments were performed on the PTT engine dynamometer test bed. The following sub-sections briefly describe the experimental preparation.

2.1 Engine Test Cell

The engine was a four-cylinder, four-stroke, turbocharged diesel engine equipped with a commonrail direct injection system. The engine specifications are shown in Table 1. The engine was mounted to the engine dynamometer system for engine load control and measurements. The engine parameters such as coolant temperature and oil temperature were measured and controlled. In order to explore a range of engine operating conditions, the original engine controller was replaced by the PTT engine controller. The open loop algorithm related to the engine crank angle position was used to control the injection timing. The simple PID close loop algorithm was used for EGR and throttle position control. Fig. 1 illustrates the schematic of the engine test cell setup.

Table 1. Engine specifications.

Number of cylinders:	4 (inline)
Number of valves:	16 (DOHC)
Manifold:	Cross-flow with turbocharger
Fuel system:	Commonrail direct injection
Displacement:	2,494 cc
Bore:	92 mm
Stroke:	93.8 mm
Connecting rod:	158.5 mm
Compression ratio:	18.5:1
Max power:	75 kW at 3,600 rpm
Max torque:	260 N·m at 1,600 – 2,400 rpm
Valve timings:	
IVO	718° CA[†]
IVC	211° CA
EVO	510° CA
EVC	0° CA
Firing order:	1-3-4-2

[†]In this study, crank angle degrees during an engine cycle will be counted after the exhaust TDC (i.e. 0° – 719°CA).

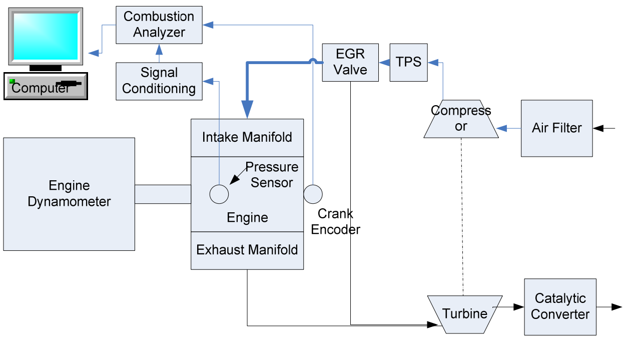

Figure 1: Schematic of the engine test cell setup.

2.2 Emission Measurement

AVL AMA1800 emission analyzer was installed to measure the concentrations of engine-out emissions at the location before the diesel catalytic converter. Measured emission species included total hydrocarbons (THC), CH_4, CO, CO_2 and NO_x. The CO_2 concentrations in the ambient and in the intake manifold were measured for the calculation of the EGR ratio.

2.3 Combustion Measurement

The combustion process is normally analyzed from the in-cylinder pressure data. As such, a Kistler pressure transducer was installed through the grow plug hole in the first cylinder, which was the closest cylinder to the EGR line. The pressure data was recorded for 50 engine cycles by the data acquisition system with a resolution of 0.1 degree of crank angle. Individual pressure data and their averages were filtered by using the Fourier series method. Mass-averaged temperature was calculated from pressure data by using an ideal gas assumption from IVC to EVO. The apparent energy release rates (AERR) were calculated by Eq. (1), where we considered the ideal gas system undergoing an isentropic process [1]:

$$\mathrm{AERR} \equiv \frac{dQ_{net}}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} \qquad (1)$$

where p is the in-cylinder pressure, V is a cylinder volume, γ is a specific heat ratio, and θ is a crank angle degree. For all experimental data presented in this paper, we used γ = 1.33.

3. EGR Calculations

The EGR line was connected from the exhaust runner of the first cylinder (upstream from the exhaust manifold) to the intake line (right after the throttle valve). The EGR flow rate was controlled by adjusting the opening position (EGR set point) of the solenoid valve. The EGR set point could be varied from 0% (fully closed) to 100% (fully opened). One popular technique to estimate the EGR amount is using CO_2 concentrations. It is the fact that CO_2 is one of the major components in the exhaust that is stable and measurable. The EGR ratio was calculated from measured CO_2 concentrations in the intake manifold and in the exhaust line (before the turbine) by using Eq. (2) [6]:

$$\%\mathrm{EGR} = \frac{[\mathrm{CO_2}]_{\mathrm{int}}}{[\mathrm{CO_2}]_{\mathrm{exh}}} \times 100 \qquad (2)$$

where $[CO_2]_{int}$ is the measured CO_2 concentration in the intake system, and $[CO_2]_{exh}$ is the measured CO_2 concentration in the exhaust system.

Notice that as the actual EGR flow rate approaches zero, %EGR calculated by Eq. (2) does not go to zero. To adjust this offset for low EGR cases (say less than 20%), we use the following equation:

$$\%\mathrm{EGR} = \frac{[\mathrm{CO_2}]_{\mathrm{int}} - [\mathrm{CO_2}]_{\mathrm{ref}}}{[\mathrm{CO_2}]_{\mathrm{exh}} - [\mathrm{CO_2}]_{\mathrm{ref}}} \times 100 \qquad (3)$$

Where $[CO_2]_{ref}$ is the measured CO_2 concentration at the reference point, which is taken as the ambient CO_2 concentration in the laboratory room.

Note that Eq. (2) and Eq. (3) are approximations and should be used with caution. Theoretically, the EGR rate is defined by the ratio between the mass of the EGR and the total mass of the charge mixture entering the cylinder [1]. However, the mass flow rate of EGR was not measured. Using Eq. (2) and Eq. (3) offered a simplified approach to estimate our EGR amount. To gain a better understanding into our EGR system characteristics, a relation between the EGR set point (%EGRSP) and the estimated EGR rate (%EGR) was monitored at two different engine conditions: a low-load operation at 2000 rpm (Fig. 2) and a medium-load operation at 2500 rpm (Fig. 3). The error bars on these figures indicate data variations calculated from the repeatability test conditions (to be discussed in the next section).

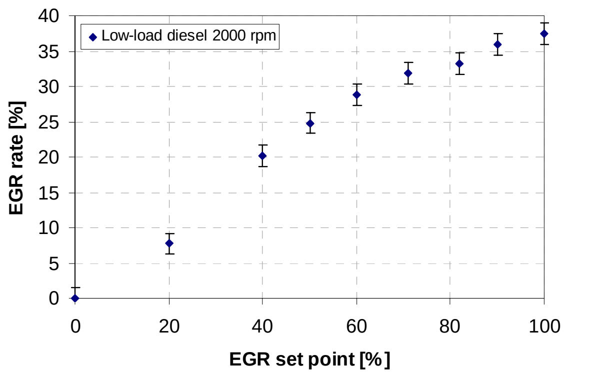

Figure 2: Estimated EGR rates at different EGR set points at low-load (IMEP ~ 3.3 bar) diesel operations at 2000 rpm.

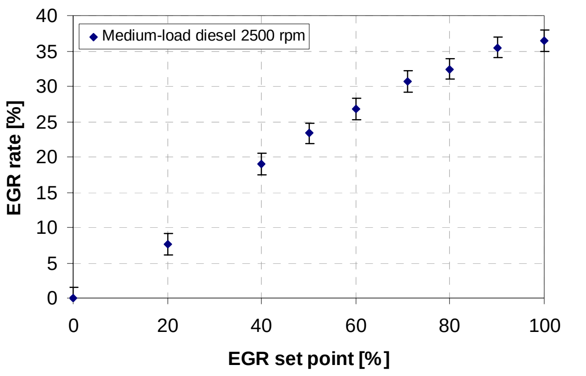

Figure 3: Estimated EGR rates at different EGR set points at medium-load (IMEP ~ 5.8 bar) diesel operations at 2500 rpm.

4. Experimental Matrices

Experiments were designed to investigate the effects of EGR on diesel engine operations at 2000 rpm and 2500 rpm. The low and medium loads were selected as their thermal efficiencies were lower than those of high-load operations. Table 2 lists all controlled operating parameters for the two engine conditions.

Table 2. Controlled operating parameters for all engine conditions.

Engine condition:	Low-load 2000 rpm	Medium-load 2500 rpm
Coolant temperature:	80°C	80°C
Oil temperature:	90°C	90°C
Throttle position:	Fully open	Fully open
EGR set point:	vary	vary
Rail pressure:	355 bar	510 bar
1st-pulse inj. signal		
Start of injection:	343.8°	349.2°
Injection duration:	4.2° CAD	12.9° CAD
2nd-pulse inj. signal		
Start of injection:	358.0°	–
Injection duration:	9.55° CAD	–

The engine condition for repeatability test was the baseline diesel operation at 2000 rpm, where the EGR set point was at 82%. For all experiments, we performed this engine condition as the first data point with combustion after the coolant and the oil reached their operating temperatures. As a routine check point, we recorded data under this condition during the day and at the last data point of the day. Repeatability data under the baseline diesel operation at 2000 rpm and the properties of our diesel fuel are provided in the appendix.

5. Results and Discussion

All experimental data were recorded at steady-state operations. Results of engine performance and engine-out emissions are presented. Then, the analysis on in-cylinder pressure data is discussed.

5.1 Engine Performance Data

Data of brake power and net thermal efficiency under low-load operation at 2000 rpm are presented in Fig. 4. Varying EGR rate did not significantly change the brake power and the efficiency of the engine under this operation. On the contrary, regulating high EGR rates under the medium-load operation at 2500 rpm caused the brake power and the efficiency to decrease (see Fig. 5).

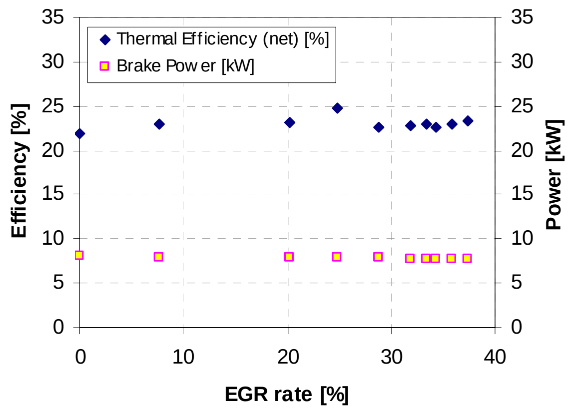

Figure 4: Net thermal efficiency and brake power at different EGR rates under low-load at 2000 rpm.

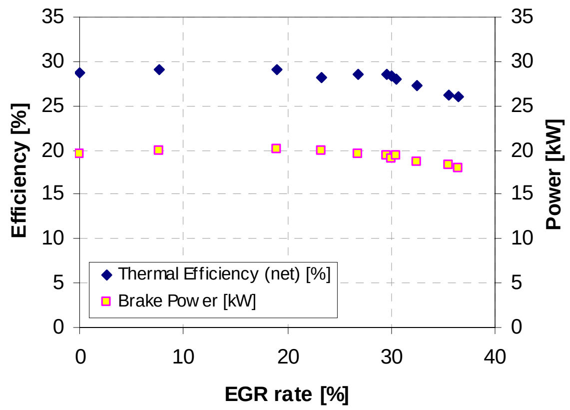

Figure 5: Net thermal efficiency and brake power at different EGR rates under medium-load at 2500 rpm.

5.2 Engine-out Emission Data

As shown in Fig. 6, NO_x were fairly decreased with a greater amount of EGR ratio at 2000-rpm operation. As the EGR rate was increased, THC tended to increase gradually and CO was steadily increased. At 2500 rpm (Fig. 7), NO_x were rapidly dropped with increasing %EGR. Meanwhile, THC was slightly increased and CO rapidly grew, especially at EGR more than 30%.

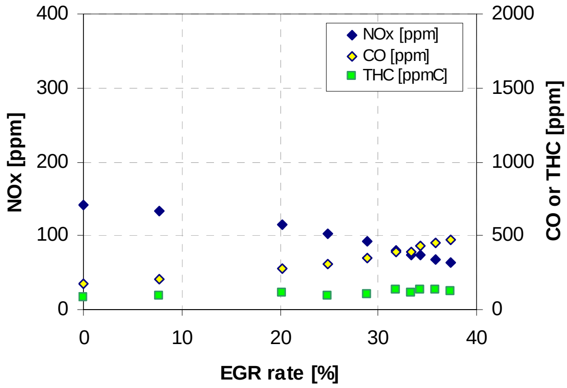

Figure 6: Engine-out emissions at different EGR rates under low-load at 2000 rpm.

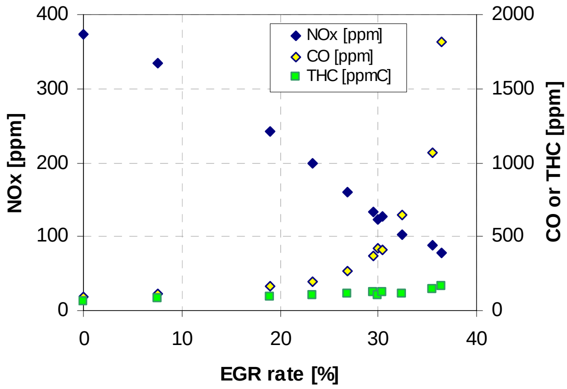

Figure 7: Engine-out emissions at different EGR rates under medium-load at 2500 rpm.

We hypothesized that the rapid reduction in NO_x with a greater EGR rate at higher loads was due to the reduced amount of O_2 concentration in the combustion chamber and a more significant decrease in the gas temperature during the combustion. The rapid rise in CO at EGR more than 30% under the 2500-rpm operation is likely to be involved with two factors. At higher loads, the O_2 concentration becomes lower. As the engine speed increases, there is less time (in milliseconds) available for the mixing process. As such, these two factors suppress the rate of CO-CO_2 oxidation.

5.3 Analysis of In-cylinder Pressure Data

Fig. 8 and Fig. 9 show histories of pressure, AERR, and mass-averaged temperature at different EGR rates under the low-load, 2000-rpm operation and the medium-load, 2500-rpm operation, respectively. It is obvious that increasing %EGR causes the pressure to decrease.

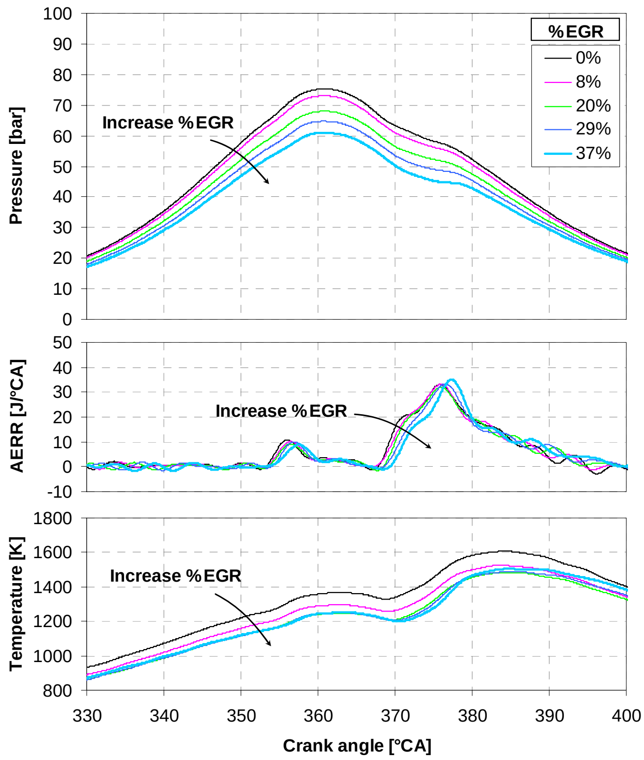

Figure 8: Histories of pressure, apparent energy release rates, and mass-averaged temperature at different EGR rates under low-load at 2000 rpm.

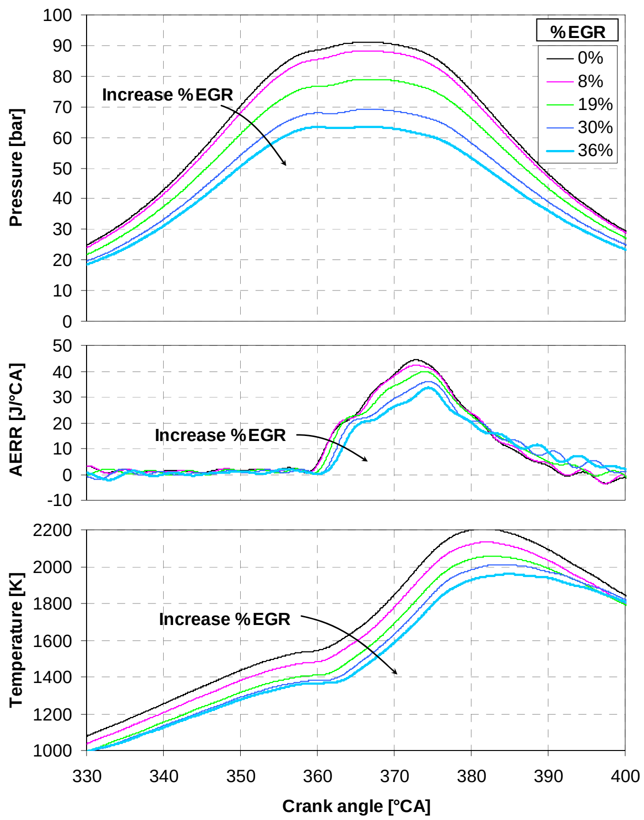

Figure 9: Histories of pressure, apparent energy release rates, and mass-averaged temperature at different EGR rates under medium-load at 2500 rpm.

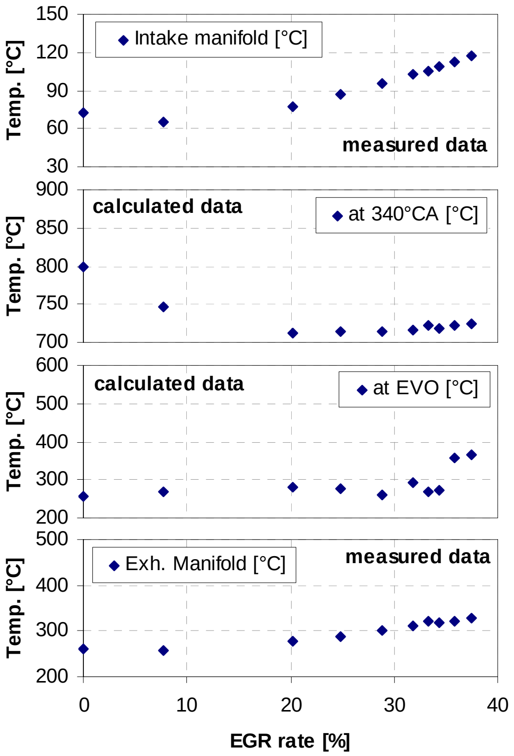

Figure 10: Measured temperatures at intake manifold and exhaust manifold, and calculated gas temperatures at 340°CA and at 510°CA (EVO) at different EGR rates under low-load at 2000 rpm.

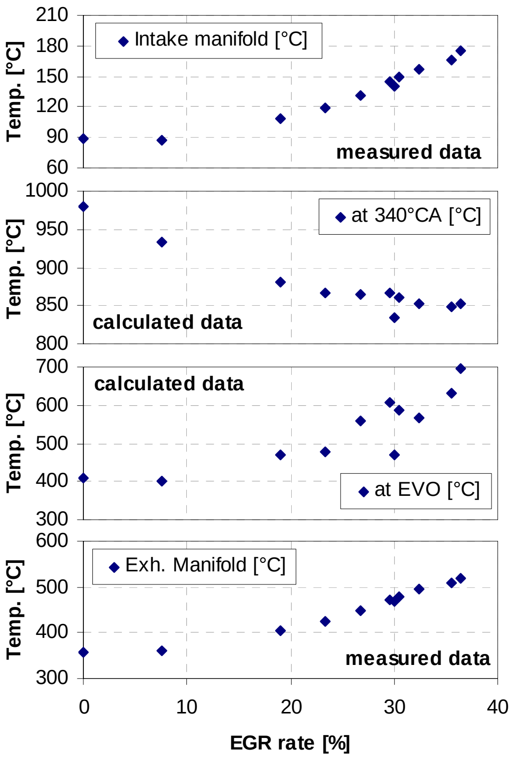

Figure 11: Measured temperatures at intake manifold and exhaust manifold, and calculated gas temperatures at 340°CA and at 510°CA (EVO) at different EGR rates under medium-load at 2500 rpm.

The combustion phasing was retarded with higher %EGR. At 2000 rpm under low-load operation, varying EGR from 0% to 37% caused the peak value of the mass-averaged temperature to reduce from 1608 K to 1505 K. For medium-load at 2500 rpm, the peak temperature was reduced from 2208 K to 1958 K by raising the EGR from zero to its maximum value. A larger drop in the peak temperature at a higher load was in agreement with our hypothesis on the rapid reduction of NO_x, as discussed above. By looking at the energy release rates, the combustion phasing was retarded with higher EGR ratios. A change in the combustion phasing leads to a change in engine operation, and its performance and emissions.

Regulating a greater EGR ratio raised the intake manifold temperature (see Fig. 10 and Fig. 11). However, a greater amount of EGR caused the specific heat ratio of the mixture to reduce. As a result, the gas temperature during the late compression (e.g. at 340°CA as shown in Fig. 10 and Fig. 11) was decreased. The decrease in the compression temperature resulted in a delayed phasing of the combustion process. As the start of combustion was retarded, the combustion duration became slightly longer (see AERR in Fig. 8 and Fig. 9). This resulted in hotter gas temperatures at EVO, as shown in Fig. 10 and Fig. 11. The exhaust manifold temperature was also increased, as a result of a higher gas temperature at EVO with a greater EGR amount.

The use of EGR should be tuned to achieve an optimum point for high engine efficiency and acceptable engine-out emissions. In some operating conditions where the mass-averaged temperature during combustion is low (e.g. low-load operations), high %EGR helps increasing the exhaust temperature. Conventional diesel oxidation catalytic converter systems require the exhaust gas temperatures to be sufficiently greater than the light-off temperatures. This implies that different EGR rates should be applied at different engine conditions. Further investigations can be examining the soot emission and performing experiments at different rail pressures. A higher injection pressure will enhance the droplet break-up and atomization, thus improving the mixing process. As such, CO and soot emissions can be reduced.

Results and findings from the current study provide a frame of reference on compression-ignition engine operations. It is interesting to explore other alternative types of combustion such as the homogeneous charge compression ignition (HCCI) [2-5] and the dual fuel (natural gas and diesel fuel) operation [7] in this engine. These two types of engine operations are attractive for a capability to achieve diesel-like efficiency together with nearly zero NO_x and soot engine-out emissions.

6. Conclusion

Data have been presented for diesel operations with a range of EGR rates in a four-cylinder, turbocharged direct injection engine. Two engine conditions explored were the low-load operation at 2000 rpm and the medium-load operation at 2500 rpm.

Results showed that the use of EGR helped reducing NO_x emissions. An excessive rate of EGR resulted in a delayed combustion phasing, lower engine efficiency, slightly greater THC, and rapidly increased CO emissions. There appeared to be an optimum amount of EGR for a certain engine condition. Compared to low-load operations, varying %EGR under operations at higher loads were likely to produce a greater impact on the combustion characteristics.

Under low load operations, high amounts of EGR should be regulated to raise the exhaust temperature at exhaust valve open. The ability to achieve higher exhaust temperatures under low load operations is favorable for diesel oxidation catalysts.

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Appendix

Table A.1. Repeatability data under baseline diesel operation (82% EGR set point) at 2000 rpm.

Parameter	Unit	Average	Max	Min	Std. Dev.
%EGR	%	35	44	31	3
Diesel mass flow	kg/h	2.68	2.74	2.62	0.03
Lambda (λ)	-	2.19	2.31	2.07	0.06
Net thermal efficiency	%	22.3	23.5	20.9	0.7
Brake power	kW	7.54	7.90	6.95	0.30
Intake temperature[1]	°C	107	112	102	3
Exhaust temperature[2]	°C	317	326	303	7
IMEP	bar	3.35	3.44	3.20	0.05
COV of IMEP	%	2.26	3.57	1.39	0.58
CA10 (10% mass burn)	°CA	362.58	370.60	360.10	2.24
CA50 (50% mass burn)	°CA	377.38	378.00	376.90	0.32
B1090 (10% to 90% burn duration)	°CAD	24.62	26.10	17.70	1.83
Total energy release[3]	J	428.87	440.33	415.15	5.94
Peak pressure	bar	62.62	64.15	61.43	0.88
Rate of pressure rise	bar/°CA	1.95	2.00	1.89	0.03
NO_x	ppm	73	90	56	9
THC	ppmC1	212	436	106	85
CO	ppm	434	576	338	59

[1] Measured at the intake manifold (after leave the compressor).
[2] Measured at the exhaust manifold (before enter the turbine).
[3] Total apparent energy release during the combustion calculated from in-cylinder pressure data.

Table A.2. Diesel fuel properties.

Specific gravity at 15.6/15.6°C	0.827
Calculated Cetane Index	57.8
Viscosity at 40°C [cSt]	3.096
Sulfur content [%wt.]	0.043
Net heating value [J/g]	42,636