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THE EFFECTS OF EGR AND SPLIT FUEL INJECTION ON DIESEL ENGINE EMISSION

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ABSTRACT−An important goal in diesel engine research is the development of a means to reduce the emission of oxides of nitrogen (NO_x) and soot particulate. A phenomenological model based on the multizone concept is used in the current paper to analyze and compare the effects of exhaust gas recirculation (EGR) and split fuel injection on emission from a compression-ignited, direct-injection engine. The present results show that NO_x can be reduced with a minimum penalty of soot particle emission with cooled EGR. Compared with EGR, split fuel injection has a higher soot penalty at a given level of NO_x reduction.

KEY WORDS : Exhaust gas recirculation, Split fuel injection, Diesel engine emission

NOMENCLATURE

C_p : specific heat at constant pressure (kJ/kg-K)
EGR : exhaust gas recirculation
EGRR : exhaust gas recirculation rate
m : gas mass (kg)
Δm_{fci} : the gaseous fuel required to meet stoichiometry (kg)
Δm_{fi} : the burned fuel (kg)
Δm_{fui} : vaporized fuel (kg)
Q : the amount of heat release (kJ)
P : pressure (kPa)
T : temperature (K)
V : volume (m^3)
Y : mass fraction (%)

Greek symbols

ϕ_g : equivalence ratio of the vaporized fuel

Subscript

a : air
exh : exhaust gas
int : intake charge
reg : recirculated exhaust gas

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1. INTRODUCTION

From a conservation and environmental standpoint, the fuel-efficient compression-ignited, direct-injection (CIDI) diesel engine, is an attractive candidate as a prime mover for automobiles. Relative to the spark-ignited IC engine, however, the diesel emits large quantities of particulate and nitrogen oxide (NO_x), both of which have been legislated for closer regulation in the near future (EPA, 2000). A dilemma exists in modifying CIDI engine processes to reduce both of these pollutants: strategies to reduce either NO_x or particulate emission induce an increased emission of the other. For example, advancing the timing for the start of injection (SOI) reduces particulate emission but concurrently increases NO_x emissions. Retarding the SOI has the opposite effect on the relative emission rates of soot and NO_x.

Exhaust gas recirculation (EGR) is a process in which a portion of the exhaust gas is recirculated into the intake air for reducing nitrogen oxide (NO_x) emission from diesel engines. The CO_2 and H_2O in the exhaust gas can effect the combustion process in the CIDI engine in three different ways: chemical, thermal, and dilution. The chemical effect stems from the dissociation of CO_2 and H_2O to form free radicals and has only a minor impact on combustion and emissions. The thermal effect arises from the higher specific heat capacity of CO_2 and H_2O relative to the replaced oxygen and also has only an indirect effect on combustion and emission. The principle influence the EGR process has on the emission associated with combustion is the dilution of the oxygen fraction with CO_2 and H_2O in the inlet charge. As a consequence of the reduced oxygen fraction, the beginning of the combustion process is delayed until the expansion stroke has begun. This delay results in lower NO_x formation since the products of combustion are exposed to high temperature for shorter periods. Negating some of this NO_x reduction, however, is the thermal throttling arising from the the increase in the intake charge temperature due to the addition of hot EGR (Ladommatos *et al.*, 1998).

Split fuel injection is another method for reducing NO_x emissions (Tow *et al.*, 1994). In this method, the injected fuel pulse is split into two or more events, which can lead to a reduction in the ignition delay in the initial fuel pulse. The split fuel injection process causes a greater fraction of combustion to occur later in the expansion stroke. Since the majority of NO_x is formed in the premixed combution, the net amount of NO_x formed during the split fuel injection is lowered.

The objective of the current work is to use a phenomenological model (Gao and Schreiber, 2000) to compare the effects of EGR and split-fuel injection on soot and NO_x emission in a diesel engine. The study considers the effects of EGR temperature, EGR fraction, and the intake charge throttling effect and compares the effectiveness of EGR and split fuel injection.

2. MODEL FORMULATION

In practice, the EGR system includes valves, piping, a cooler, and a venturi in which EGR mixing occurs. In the current EGR analysis, the following two assumptions are made regarding EGR mixing:

(1) The exhaust gas mixes completely with air before entering the intake valve.

(2) The pressure of the intake charge within the throttle valve is constant.

When exhaust gas is introduced into the intake air, the EGR Rate (EGRR) is defined as:

$$\text{EGR Rate} = 100 \times \frac{\%\text{CO}_2(\text{intake})}{\%\text{CO}_2(\text{exhaust})}$$

The temperature of the intake charge is calculated using a simple heat balance.

$$T_{\text{int}} = \frac{(C_p mT)_a + (C_p mT)_{egr}}{(C_p m)_a + (C_p m)_{egr}}$$

The thermal throttling caused by the low density of the hot EGR can lead to reduced intake mass flow. A mass analysis can be derived using the perfect gas equation:

$$m_{\text{int}} = \left(\frac{PV}{RT}\right)_{\text{int}}$$

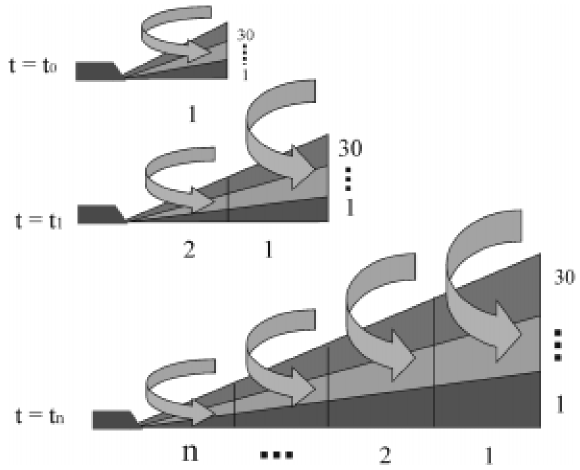

Figure 1. Packet distribution during the fuel injection in symmetric half of the spray jet.

And the mass fraction of each species is calculated by:

$$Y_{\text{int}} = \frac{m_a Y_a + m_{exh} Y_{exh}}{m_{\text{int}}}$$

The basic concept of the phenomenological model (Gao and Schreiber, 2000) used for the current analysis divides the injected spray into small packages, or multiple zones, in the radial and axial directions. (See Figure 1) This multizone model consists of a spray model, an entrainment and evaporation model, a heat release model, and models for NO_x formation and soot formation and oxidation. At each crank angle of injection, a new set of packets, each containing equal masses of fuel, is injected. The mass of fuel in each packet is determined by dividing the total fuel mass to be injected by the total number of packets injected. As the injection process progresses, liquid fuel in packages injected earlier begins vaporizing and gaining entrained air. To predict the transient and heterogeneous process of diesel combustion, the fuel-air mixture is modeled in a multizone fashion. Each packet has its own temperature and composition history. The time-dependent pressure in the engine cylinder varies uniformly in all the packets. The first law of thermodynamics and the conservation equations for mass and momentum are used to derive a differential equation describing the thermodynamic state of each packet. Differential equations also describe the chemical reactions in the packets. Solution of the packets differential equations yields the local temperature, the concentrations of oxygen and carbon dioxide, and the soot and NO_x concentration for each packet.

The number of zones needed to depict the length of the fuels jet's axis is determined by the duration of the fuel injection period and the calculation time step used, while the number of radial zones is determined by the accuracy requirement. In the current work 30 radial zones were used. The time step used is 0.2 deg of crank angle; consequently, the total number of packets tracked over 20 CA degrees in the computation is more than 3000. In order to simplify the problem, each package is assumed to be an isolated system, and mass and energy transport with respect to the packet is determined empirically for each package as a function of that package's location and local conditions only. Heat and mass transfer between packets is not modeled directly.

The heat release submodel of Hiroyasu's phenomenological model (Hiroyasu, 1983) considers only the effect of stoichiometric heat release in diesel combustion. In actuality, however, the combustion process in the typical diesel engine consists of two stages: premixed combustion and mixing-controlled combustion.

In the current model the heat release in a packet during a time step consists of two steps.

In the first step, premixed combustion is described using a two equation reaction scheme (Emerson and Rutland, 1999):

$$a\text{Fuel} + bO_2 \Leftrightarrow c\text{CO} + dH_2O \tag{1}$$

$$CO + \frac{1}{2}O_2 \Leftrightarrow CO_2 \tag{2}$$

Premixed combustion is kinetically limited as described in the reaction rate equation (1):

$$\frac{d}{dt}[\text{Fuel}]_p = -3.8 \times 10^{10}[\text{Fuel}]^{0.25}[O_2]^{1.5}\ \exp(-9250/T)$$

The rate coefficients for reaction equation 2 are taken from:

$$\frac{d}{dt}[CO] = -10^{14.6}[CO][O_2]^{0.25}[H_2O]^{0.5}\ \exp(-40.0/RT) + 5 \times 10^{8}[CO_2]\ \exp(-20000/T)$$

The diffusion-controlled combustion process is described by the following one equation reaction scheme:

$$\frac{d}{dt}[\text{Fuel}] = -1.5 \times 10^{11}[\text{Fuel}]^{0.25}[O_2]^{1.5}\ \exp(-15078/T)\times f(T)$$

where $f(T) = 1/(T/1000)^{8.0}$

The transition from premixed combustion to mixing controlled combustion is dependent on the critical Dahmkohler number, which is set equal to 100 in the current work. The Dahmkohler number, in the units of cm-sec-mole-kcal-K, is defined by the equation:

$$Da = \frac{A[\text{Fuel}][O_2]\exp(-Ea/RT)}{\min\left([\text{Fuel}], \left(\frac{[\text{Fuel}]}{O_2}\right)_{stoich}[O_2]\right)}$$

Table 1. Engine specifications for the Caterpillar 3406 engine.

Bore (mm)	137.19
Stroke (mm)	165.1
Connecting rod length (mm)	263.62
Compression ratio	15:1
Engine speed (rpm)	1600
Intake pressure (KPa)	201
Intake temperature (K)	310
Fuel injection (g/cycle)	0.1622
Swirl ratio	1.0
Wall temperature (K)	500
Exhaust pressure	105 Kpa
Injection duration (CA)	21
Injection pressure (Mpa)	90
Nozzle orifice diameter (mm)	0.259
Number of nozzle hole	6
Engine load fraction	75%
Initial fuel temperature (K)	312

In the current work, the heat release submodel includes the effect of the stoichiometric reaction in diesel combustion. Heat release from a single packet during a time step is given by:

$$\Delta Q = H_u \Delta m_{fi}$$

where Δm_{fi} can be expressed as:

$$\Delta m_{fi} = \begin{cases} \Delta m_{fui} & \phi_g < 1 \\ \Delta m_{fci} & \phi_g > 1 \end{cases}$$

The total heat release is the summation of heat release from all the packets.

3. RESULTS AND DISCUSSION

The Caterpillar 3406 diesel engine is chosen in the study.

Figure 2 illustrates a comparison of the current model with a model using Hiroyasu's heat release model and with experimental data (Patterson *et al.*, 1994).

The soot – NO_x tradeoff as a function of injection timing for the baseline engine is illustrated in Figure 3. The trend predicted by the current model using a new heat release submodel is reasonably close to experimental results.

4. ANALYSIS OF EXHAUST GAS RECIRCULATION (EGR)

Since the addition of exhaust gas raises the intake temper-

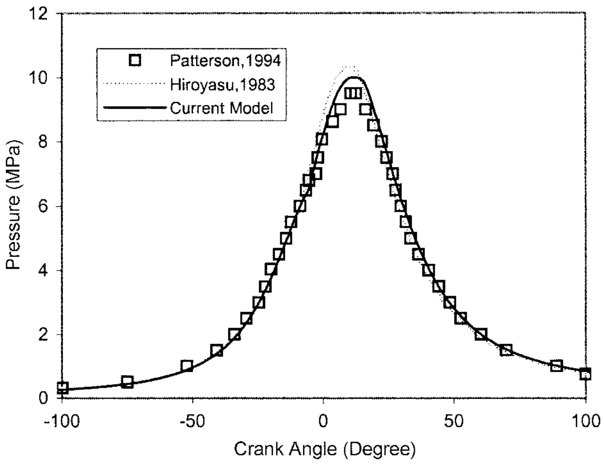

Figure 2. Comparison of the predictions of two models with experimental measurements of engine cylinder pressure as a function of crank angle.

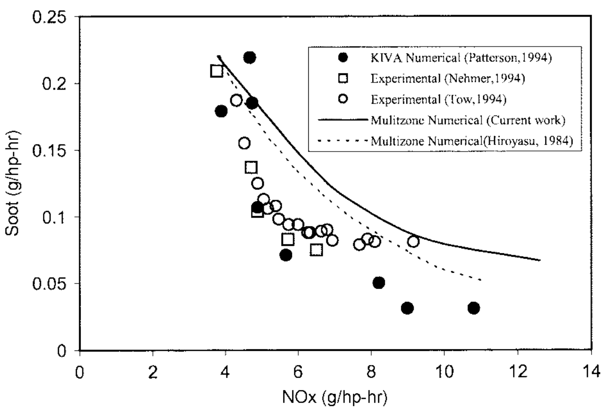

Figure 3. Comparison of the two models, multidimensional model predictions, and experimental measurements of the soot and NO_x tradeoff curve over the range of injection timings from 5 to 15.

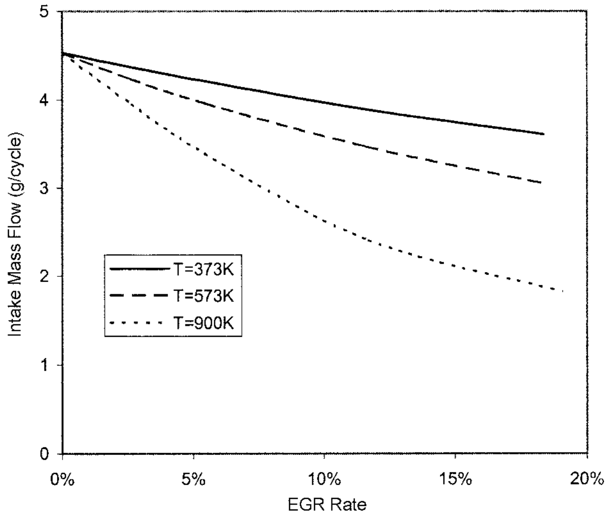

Figure 4. The effect of EGRR on intake mass flow at the different temperature of recirculated exhaust gas for EGR.

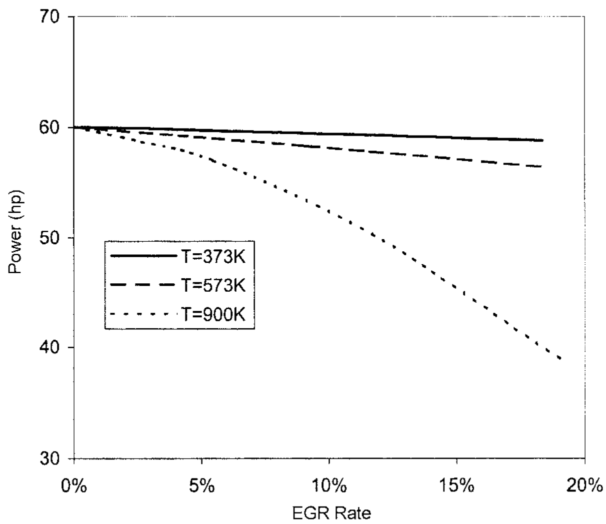

Figure 5. The effect of EGRR on intake mass flow at different temperatures of recirculated exhaust gas for EGR.

ature, EGR can cause thermal throttling at the intake; therefore, the effect of the recirculated gas temperature on thermal throttling of the intake gas mass flow is an important consideration. Figure 4 illustrates the dependency of the intake gas flow rates on the EGR rate and temperature. For example, with a 6% EGR rate, recirculated gas at 900K reduces the flow rate of the inlet charge by 25% whereas 573K recirculated gas reduces the mass flow by 13%. Recirculated gas at 373K reduces the mass flow by 7%. As a consequence of thermal throttling the power output from a diesel engine is lowered in proportion to a combination of recirculated gas temperature and EGRR as illustrated in Figures 5 and 6. Figure 5 illustrates that at 373K recirculated gas accounts for very little power degradation regardless of the EGRR. The close correlation between recirculated gas temperature and power degradation emphasizes the importance of cooling the recirculated gas used in the EGR process.

Figure 6 shows the effect of EGR rate on inlet charge temperature. Since the exhaust gas temperature is higher than that of the intake air, EGR raises the inlet charge temperature and reduces the inlet charge mass due to thermal throttling. Increasing the inlet charge has a

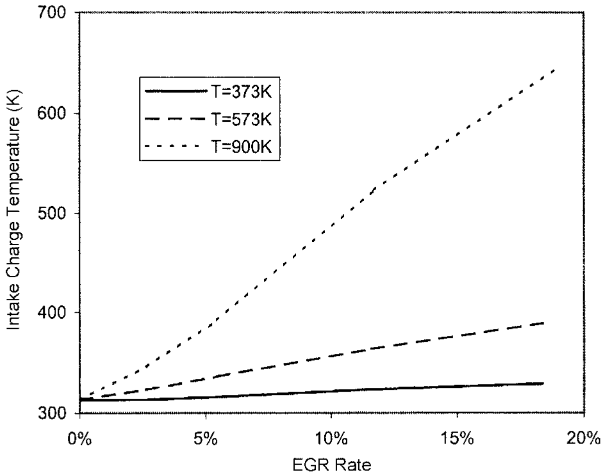

Figure 6. The effect of EGRR on intake charge temperature at various recirculated exhaust gas temperatures.

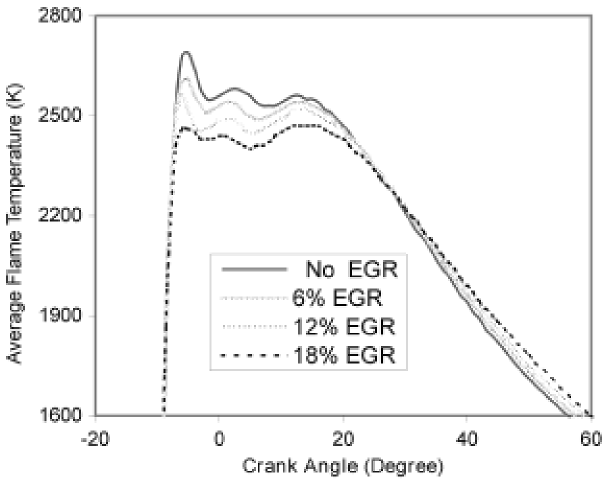

Figure 7. The average flame temperature at different EGR rates of 373 K recirculated exhaust gas.

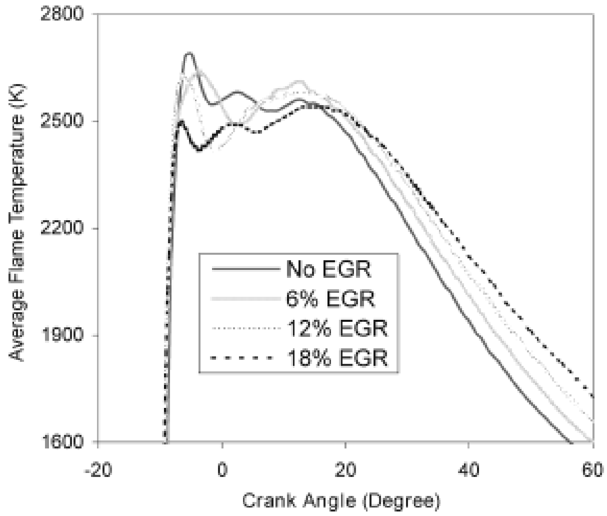

Figure 8. The average flame temperature at different EGR rates of 573 K recirculated exhaust gas.

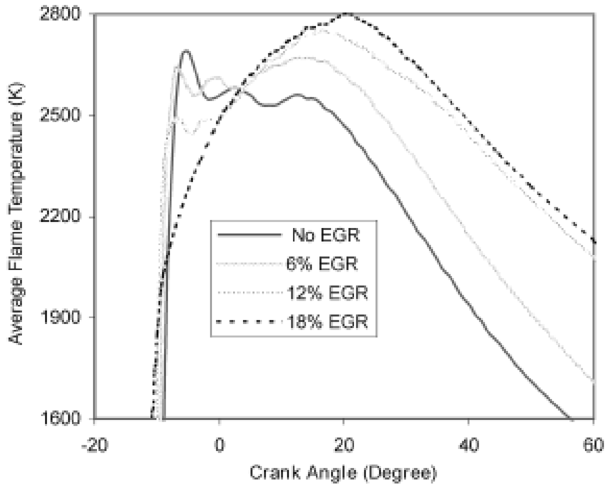

Figure 9. The average flame temperature at different EGR rates, when hot exhaust gas without cooling (T = 900 K) is directly used for EGR.

complex effect on combustion process, as is reflected by average flame temperature shown in Figures 7-9. The flame temperature is inversely proportional to the EGR rate of the cooler recirculated exhaust gas as illustrated in Figure 7. Higher temperature EGR rates, on the other hand, can lead to increases in the average cylinder gas temperature throughout the combustion cycle, as shown in Figures 8 and 9. The higher charge temperature due to EGR enhances fuel evaporation and air-fuel mixing during the ignition delay period and the combustion process. This higher temperature enhances soot oxidation but also increases NO_x formation.

When EGR is employed, the oxygen concentration of the intake air is diluted with between two to three percent CO_2 and H_2O as illustrated in Figures 10, 11, and 12. Although the concentrations of CO_2 and H_2O are small, the effect on combustion and emissions is significant.

Particulate and NO_x emissions are dependent on the effects of thermal throttling, recirculated gas temperature, and species concentration in the EGR system. Figure 13 illustrates the importance of cooling recirculated gas used in the EGR in order to reduce NO_x emission. As the temperature of the recirculated gas increases, the effectiveness of the EGR system to lower NO_x emission decreases. If the recirculated gas can be cooled to 373 K, NO_x can be reduced by 31%, 56%, and 77% for EGR rate

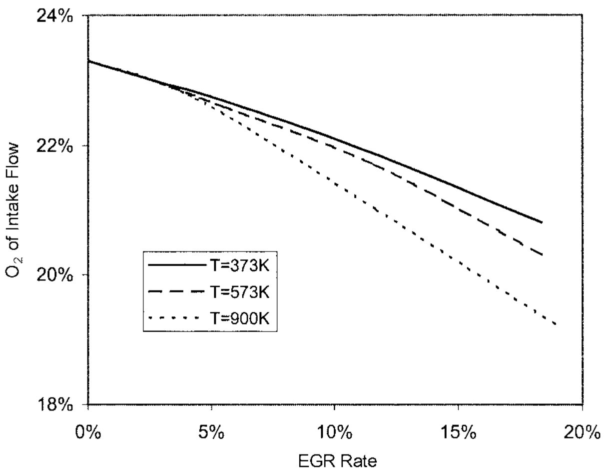

Figure 10. The effect of EGR on the O_2 concentration of intake flow at the different temperature of recirculated exhaust gas.

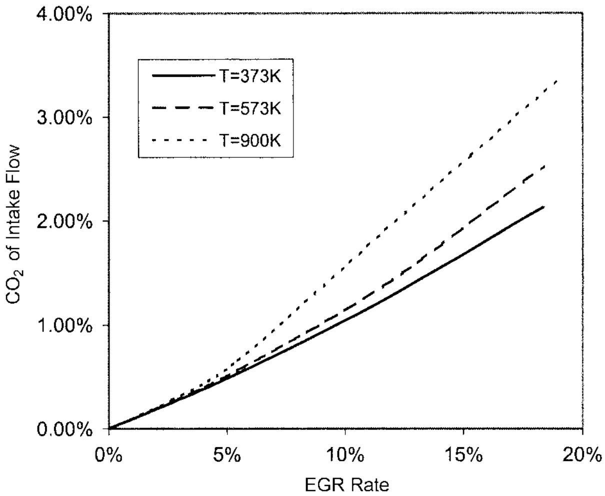

Figure 11. The effect of EGR on the CO_2 concentration of intake flow at the different temperature of recirculated exhaust gas.

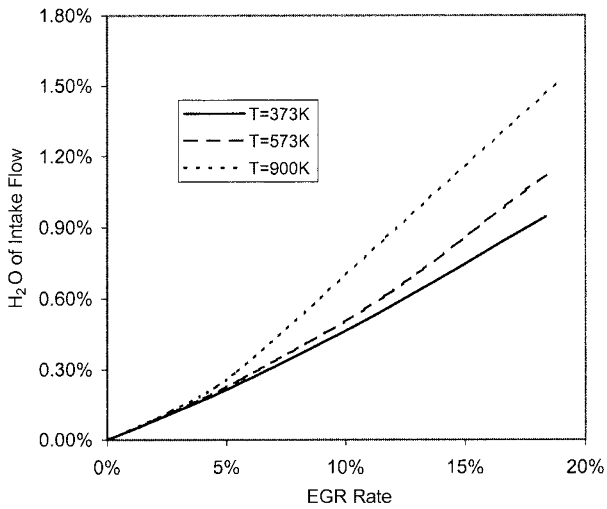

Figure 12. The effect of EGR on the H_2O concentration of intake flow at the different temperature of recirculated exhaust gas.

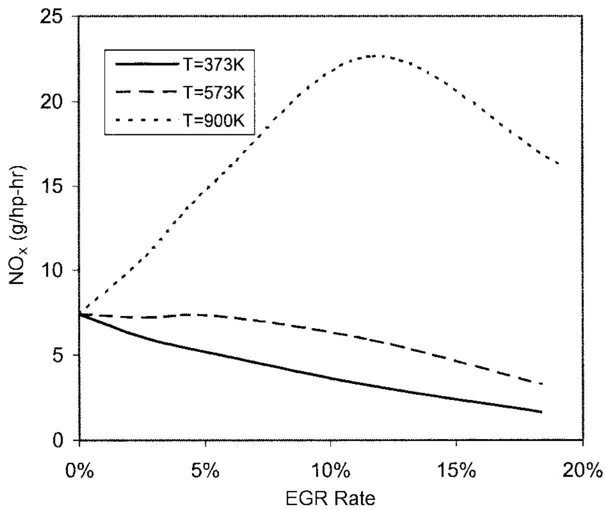

Figure 13. The effect of EGR on the NO_x emission at the different temperature of recirculated exhaust gas.

of 6%, 12%, and 18% respectively, and EGR can be a very effective way to reduce NO_x emission.

Particulate emission can likewise be reduced using EGR as illustrated in Figure 14. Even though the recirculated exhaust gas reduces some of the intake oxygen content, the heat added to the intake air enhances the soot oxidation to some extent. As seen in Figure 14, soot emission increases dramatically with EGRR for high temperature exhaust gas. The combination of high temperature recirculated gas and a large EGR rate creates a large throttling effect at the intake, which enhances particulate production and hinders soot oxidation.

The multizone model, with an EGR rate of 6% and an EGR temperature of 373K is used for a more detailed study of the effect of EGR on the soot and NO_x formation during the combustion cycle. For this study, the start of injection (SOI) is 11CA BTDC.

In Figure 15, the effect of the EGR process on the combustion process is seen. The heat release in premixed combustion phase is reduced due to the reduction in O_2 availability associated with the EGR diluents. Moreover, the retardation of the combustion process to a time later

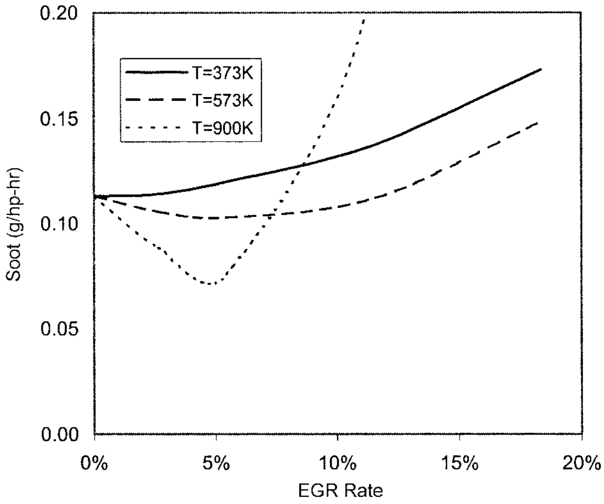

Figure 14. The effect of EGR on the soot emission at the different temperature of recirculated exhaust gas.

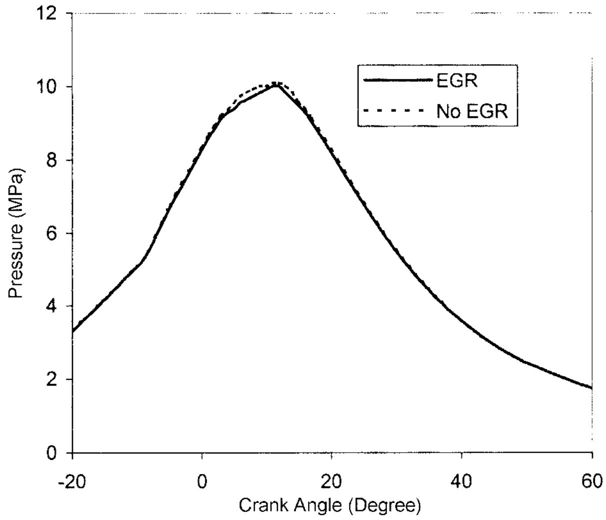

Figure 16. The comparison of pressure between baseline and EGR case, in which EGR rate = 6% and T = 373 K.

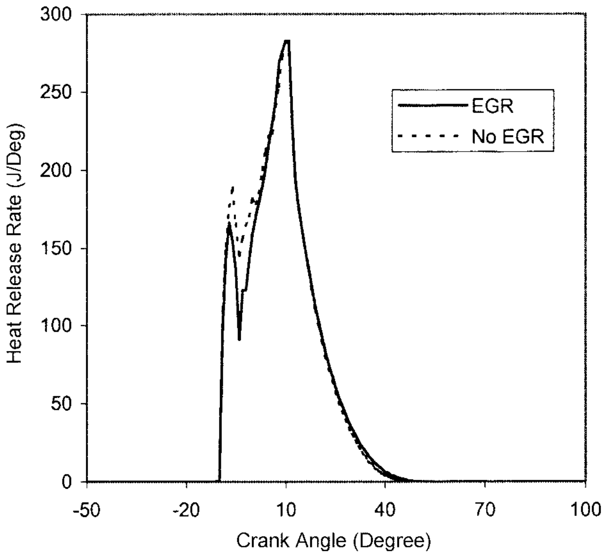

Figure 15. The comparison of heat release rate between baseline and EGR case, in which EGR rate = 6% and T = 373 K.

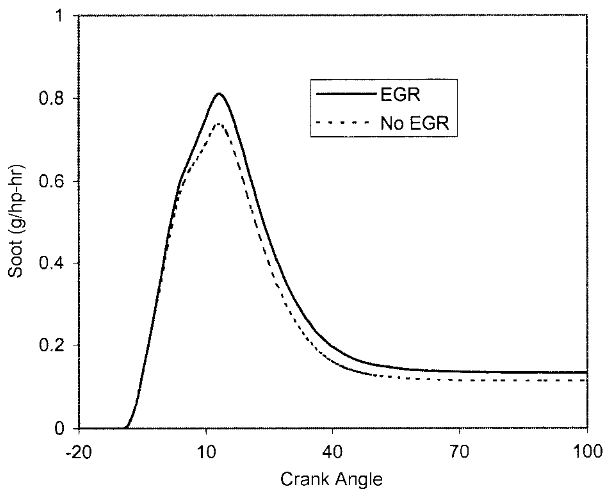

Figure 17. The comparison of soot between baseline and EGR case, in which EGR rate = 6% and T = 373 K.

in the expansion stroke results in earlier quenching of the combustion process, which leads to a shorter duration of high-temperature combustion so that NO_x has less time to form.

Figure 16 compares the cylinder pressure histories with and without EGR. The heat releases are similar except that the presence of EGR has lowered the heat release peak. Figures 17 and 18 illustrate the histories of particulate formation and destruction and NO_x formation, respectively. Employing recirculated gas, with the given EGR rate and temperature, can effectively reduce NO_x emission with only a small penalty in soot emission.

The soot – NO_x tradeoff with EGR is illustrated in Figure 19. The curve is found by varying the start of fuel injection between 11CA BTDC and 10CA ATDC. Lines connect the two curves at locations of similar SOIs. As can be seen, soot emission remains almost constant at a given injection timing, while EGR significantly reduces NO_x emission. Using EGR, it is seen that advancing the fuel injection to a point where NO_x emission is still acceptable can result in reduced soot emission.

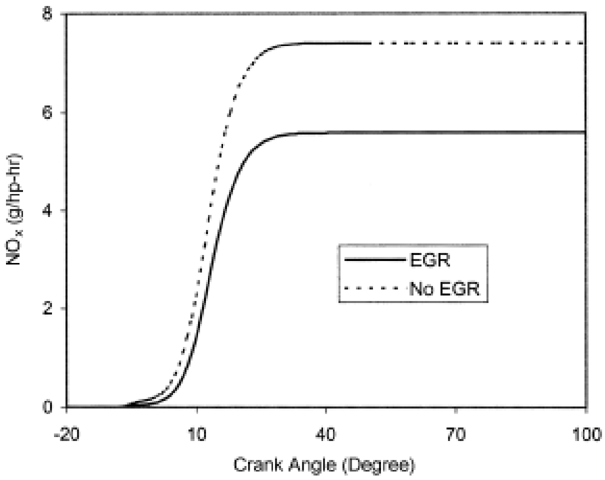

Figure 18. The comparison of NO_x between baseline and EGR case, in which EGR rate = 6% and T = 373 K.

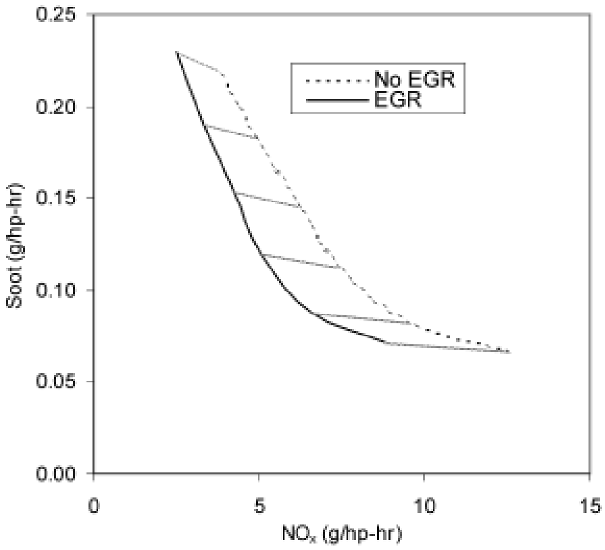

Figure 19. The comparison of soot and NO_x tradeoff between baseline and EGR case, in which EGR rate is 6% and T = 373 K.

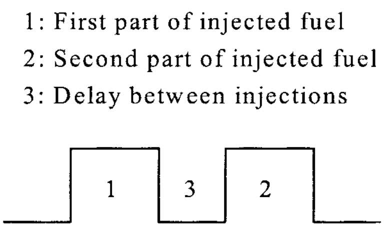

Figure 20. Description of split injection nomenclature.

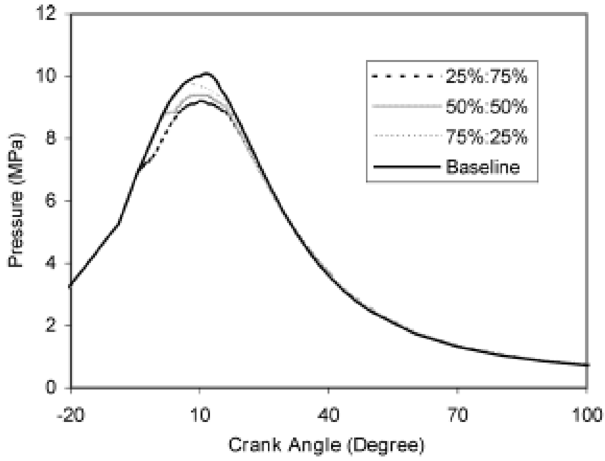

Figure 21. Effect of split fuel injection on pressure history when delay between injections is 3 CA.

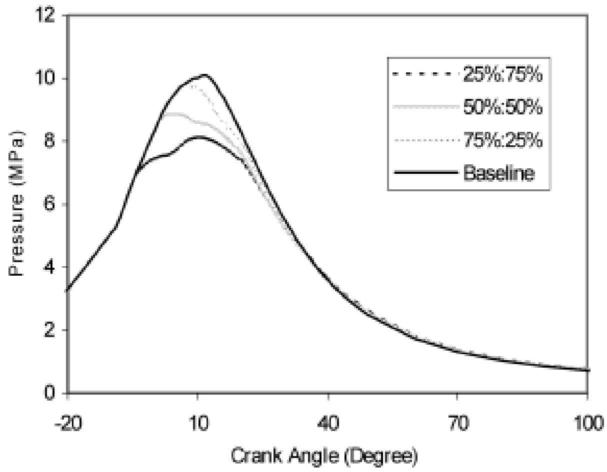

Figure 22. Effect of split fuel injection on pressure history when delay between injections is 8 CA.

5. SPLIT FUEL INJECTION

The use of split fuel injection in diesel combustion (Montgomery and Reitz., 1996) and (Patterson *et al.*, 1994) is another strategy for reducing NO_x. Split fuel injection reduces NO_x formation by retarding the injection timing of a fraction of the fuel.

Figure 20 illustrates the split injection strategy. In the current analysis of the effect of split fuel injection on soot and NO_x emission, three split injection ratios (25%:75%, 50%:50%, and 75%:25%) and two injection time delays, 3CA and 8 CA, are considered.

Figures 21 and 22 illustrate the effect of the injection ratios and the split injection delay times on the pressure histories. Combustion with split fuel injection affects the

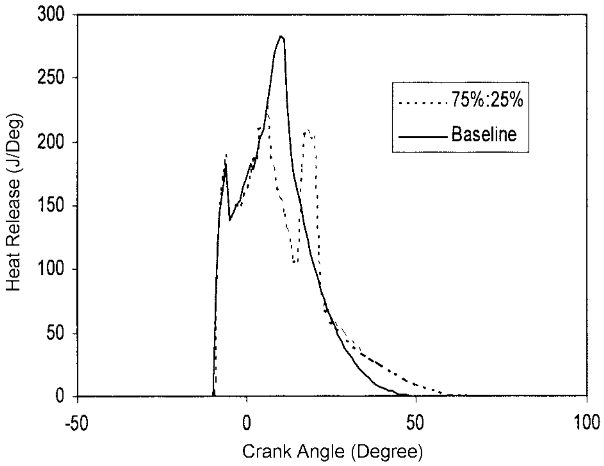

Figure 23. Comparison of Heat release curve between baseline and split injection (75%:25%) when delay between injection is 8 CA.

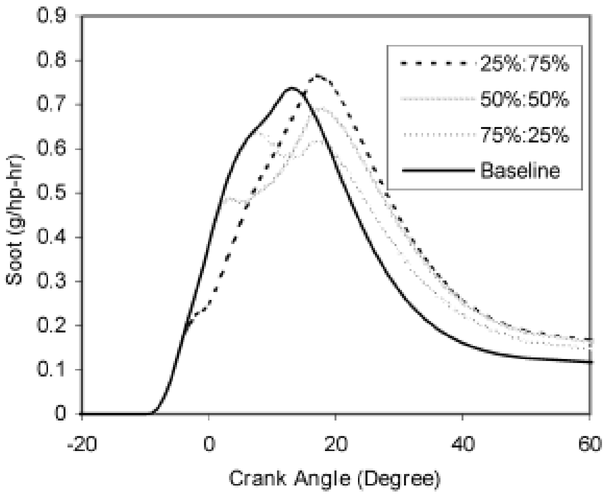

Figure 24. Soot history curve at different split fuel injection ratios when delay between injection is 3 CA.

heat release profiles seen in Figure 23 for the combustion with 75%:25% fuel split and an 8 CA delay time.

The altered shape of heat release caused by split injection can affect soot and NO_x emission significantly. Figures 24 and 25 illustrate that two peaks in soot production can be associated with split injection. The size of the split injection ratio and the delay between injections strongly affect the timing and magnitude of the second peak. The second peak is significantly moved toward the expansion stroke for the case of 8CA delay between injections. The presence of split fuel injection causes soot emission to increase. Soot oxidation is reduced since more particulate is formed later in the

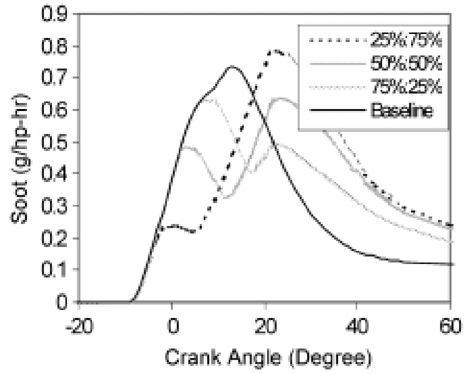

Figure 25. Soot history curve at different split fuel injection ratios when delay between injection is 8 CA; here solid curve is no split injection.

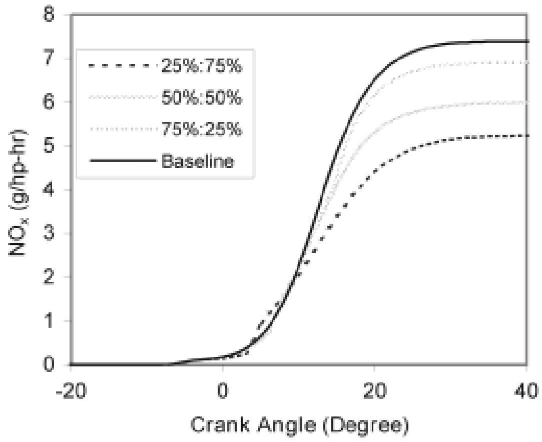

Figure 26. NO_x history curve at different split fuel injection ratios when delay between injection is 3 CA; here solid curve is no split injection.

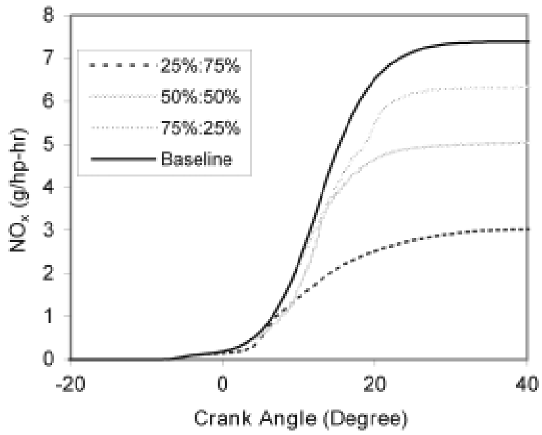

Figure 27. Soot history curve at different split fuel injection ratios when delay between injection is 3 CA; here solid curve is no split injection.

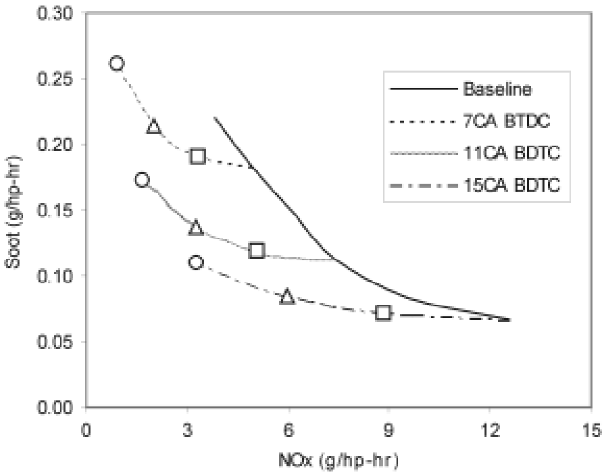

Figure 28. Plot of soot emission and NO_x emission as a function of EGR fraction (□ : 6%EGR; □ : 12% EGR; □ : 18%EGR) and fuel injection timing. Baseline has no split fuel injection.

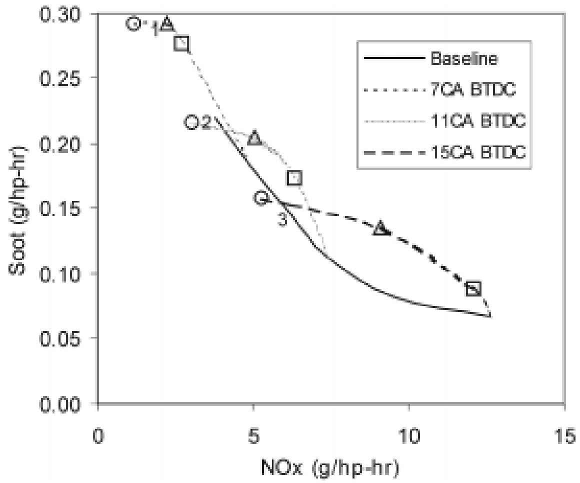

Figure 29. Plot of soot emission and NO_x emission as a function of split injection ratio (□ : 25%:75%; □ : 50%:50%; □ : 75%:25%) and fuel injection timing.

expansion stoke. On the other hand, split injection is found to reduce NO_x emission significantly since it reduces the magnitude of the combustion peak as seen in Figures 26 and 27.

While split fuel injection is seen to reduce NO_x emission, the adverse affect on soot oxidation seems significant.

6. COMPARISON BETWEEN EGR AND SPLIT FUEL INJECTION

At a given SOI, both EGR and split injection reduce NO_x emission effectively; yet both increase soot emission in diesel combustion. Since a greater portion of the fuel is injected at a later time with split injection, it has some of the characteristics of simply retarding the fuel injection of the non-split fuel pulse.

In Figure 28, the solid curve illustrates soot and NO_x tradeoff with no EGR when injection timing is varied from 15 CA BTDC to 5 CA BTDC; dashed curve 1 shows the effect of EGR at 7CA BTDC injection timing with a range of EGR rate from 0 to 18% at 373 K; Dashed curve 2 portrays the effect of EGR at 11CA BTDC injection timing; dashed curve 3 is for EGR at 15CA BTDC injection timing. In the range tested, the higher the rate of exhaust gas recirculation, the more attractive the trade-off curve.

Figure 29 illustrates the effect of split injection for optimizing the soot – NO_x trade-off curve. Dashed curve 1 shows the effect of split injection at 7CA BTDC injection timing; dashed curve 2 portrays the effect of split injection at 11CA BTDC injection timing; dashed curve 3 is for split injection at 15CA BTDC injection timing. At each injection timing, a 8CA delay exists between injection pulses, and split fuel injection rates of 25%:75%, 50%:50%, and 75%:25% are plotted. In general, a comparison of Figures 28 and 29 illustrate that the soot penalty of split fuel injection is higher than that for EGR at a given NO_x emission. Only the 50%:50% split fuel ratio adds any advantage to the baseline case for improving the soot – NO_x trade-off curve.

7. CONCLUSIONS

The purpose of this investigation is to study and compare the strategies of EGR and split-injection for NO_x emission reduction with a minimized increase in soot emission.

The EGR affects diesel combustion through the dilution of the inlet charge oxygen concentration with CO_2 and H_2O, the increase in the inlet charge temperature, and the thermal throttling arising from the use of hot EGR. Cooled EGR shows promise for significantly reducing NO_x emission with no or small penalty of soot particle emission. Using EGR, it is seen to be possible to reduce soot emission by advancing the fuel injection to a point where NO_x emission is still acceptable.

Split fuel injection can be used to reduce NO_x emission; however, the reduction in soot oxidation is significant resulting in higher soot emission.

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