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A Simulation Study on Reducing the NOx Emission of Turbo Diesel Bus Engines

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Abstract

This research presents a solution to reduce nitrogen oxide emissions from bus vehicles in Vietnam by using exhaust gas recirculation (EGR) system. The bus engine and its EGR system were built in the AVL-BOOST software based on the engine parameters of the most popular diesel engine bus in Hanoi city, the capital of Vietnam. The simulation results showed that the EGR system contributed for dramatically reducing NOx emission. In addition, the engine power reduction can be minimized when the EGR rate is optimized corresponding to the engine operating conditions. Especially, at 75% load, the reduction of NOx emissions was 60% when EGR valve was fully opened. However, soot emission was also dramatically increased. Consequently, when using EGR technology to reduce NOx emissions, it is recommended that the solution to control soot emissions should be considered.

Keywords: **Exhaust gas recirculation, Environmental emissions, Diesel engine, engine characteristic, Nitrogen oxides, Soot.**

1. Introduction

Transportation vehicles play an important role in the development of countries. However, the exhaust emissions from vehicles become more and more serious and directly affect to the environmental quality and human well-being, especially in developing countries such as Vietnam. Currently, diesel and gasoline engines are still popular and are the major power sources retrofitted for transportation vehicles. Diesel engines normally operate at a higher temperature and pressure in comparison with those of gasoline engines. As a result, these conditions result in the increase of NOx emissions, which are advantageously formed at the high temperature and pressure. Consequently, many studies investigated and proposed the solution to reduce the NOx emissions of diesel engines [1-5]. EGR technology is a good and suitable method to control NOx. As described be Ibrahim et al. [6], NO emissions could be reduced up to 50% by adding EGR to the intake charge. In another research, a stronger effect in reducing NOx emission of EGR technology is described by the results of Schöffler et al. [7], in which a reduction of 80–90% in NOx emissions was obtained when applying EGR system for the test engine. Yang Shuai et al. [8] also performed a simulation of the turbocharged diesel engine retrofitted EGR system, the results showed the EGR system contributed for the reduction of air fuel ratio, cylinder pressure, cylinder temperature and exhaust temperature; as a result, the NOx emissions were dramatically decreased.

Based on the EGR technology, this research aims to find out the solution to reduce NOx emissions of bus vehicles in Vietnam. Consequently, the simulation of bus engine and its EGR system were modeled in AVL_BOOST software based on the engine characteristics of real engines. The simulation model was also verified by comparing its engine performance curves with that of the experiment. Corresponding to the each operating engine point, the EGR rate was evaluated and optimized.

2. Simulation of the test engine retrofitted EGR system

2.1. Theoretical basis and governing equation

2.1.1. Basic conservation equations

The calculation of the thermodynamic state of the cylinder is based on the first law of thermodynamics. The first law of thermodynamics for high pressure cycle states that the change of the internal energy in the cylinder is equal to the sum of piston work, fuel heat input, wall heat losses and the enthalpy flow due to blow-by, Eq. 1 [9,10]:

$$\frac{d(m_c.u)}{d\alpha} = -p_c \cdot \frac{dV}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_W}{d\alpha} - h_{BB} \cdot \frac{dm_{BB}}{d\alpha} \quad (1)$$

Where m_c - mass in the cylinder, u - specific internal energy, p_c cylinder pressure, V - cylinder volume, Q_F - fuel energy, Q_W – wall heat loss, α - crank angle, h_{BB} - enthalpy of blow-by, m_{BB} - blow-by mass flow

2.1.2. Combustion Model

BOOST_SOFTWARE uses the Mixing Controlled Combustion (MCC) model for the prediction of the combustion characteristics in direct injection compression ignition engines [9,10]. The heat release is a function of the fuel quantity available (f_1) and the turbulent kinetic energy density (f_2), Eq. 2.

$$\frac{dQ}{d\phi} = C_{Comb} \cdot f_1(M_F, Q) \cdot f_2(k, V) \quad (2)$$

Where $f_1(M_F, Q) = M_F - \frac{Q}{LVC}$, $f_2(k, V) = \exp(C_{rate} \cdot \frac{\sqrt{k}}{\sqrt[3]{V}})$, C_{Comb} - combustion constant [kJ/kg.deg CA], C_{rate} - mixing rate constant [s], k - local density of turbulent kinetic energy [m^2/s^2], M_F - vapourized fuel mass [kg], LCV - lower heating value [kJ/kg], Q - cumulative heat release for the mixture controlled combustion [kJ], V - cylinder volume [m^3], α - crank angle [deg CA].

2.1.3. Heat Transfer Model

The heat transfer to the walls of the combustion cham-ber, i.e. the cylinder head, the piston, and the cylinder liner, is calculated from Eq. (3) [2].

$$Q_{wi} = A_i \cdot \alpha_i \cdot (T_c - T_{wi}) \quad (3)$$

where Q_{wi} - wall heat flow, A_i - surface area, α_i - heat transfer coefficient, T_c gas temperature in the cylinder, T_{wi} - wall temperature.

Heat transfer coefficient (α_i) is usually calculated by WOSCHNI Model, The Woschni model published in 1978 for the high pressure cycle is summarized as follows: [3]

$$\alpha_w = 130.D^{-0,2}.p_c^{0,8}.T_c^{-0,53}.[C_1.c_m + C_2 \cdot \frac{v_D.T_{c1}}{p_{c,1}.V_{c,1}}.(p_c - p_{c,0})]^{0,8} \quad (4)$$

Where $C_1 = 2,28 + 0,308.c_u/c_m$, $C_2 = 0,00324$ for DI engines, D - cylinder bore, c_m - mean piston speed, c_u - circumferential velocity, $c_u = \pi.D.nd/60$, V_D - displace-ment per cylinder, $p_{c,o}$ - cylinder pressure of the motored engine [bar], $T_{c,1}$ - temperature in the cylinder at intake valve closing (IVC) , $p_{c,1}$ - pressure in the cylinder at IVC [bar].

2.1.4. Turbocharger simulation

For the simulation of a turbine, the performance characteristics along a line of constant turbine are required. The power provided by the turbine is determined by the turbine mass flow rate and the enthalpy difference over the turbine.

$$P_T = \dot{m}.\eta_m.(h_3 - h_4) \quad (5)$$

where P_T - turbin power, $\dot{m}$ - turbin mass flow, η_m - mechanical efficiency of the turbocharger, h_3 - enthalpy at the turbine inlet, h_4 - enthalpy at the turbine outlet

$$h_3 - h_4 = \eta_{s,T}.c_p.T_3.\left[1-\left(\frac{p_4}{p_3}\right)^{\frac{K-1}{K}}\right] \quad (6)$$

where $\eta_{s,T}$ - isentropic turbine efficiency, Cp - mean specific heat at constant pressure between turbine inlet and outlet, T_3 - turbine inlet temperature, p_4/p_3 - turbine expansion ratio, η_{tot} - total efficiency of the turbine = $\eta_m.\ \eta_{s,T}$.

The power consumption of the turbo compressor depends on the mass flow rates in the compressor and the enthalpy difference over the compressor. The latter is influenced by the pressure ratio, the inlet air temperature, and the isentropic efficiency of the compressor.

$$P_c = \dot{m}.(h_2 - h_1) \quad (7)$$

where P_c - compressor power consumption, $\dot{m}_c$ - mass flow rate in the compressor, h_2 - enthalpy at the outlet of the compressor, h_1 - enthalpy at the inlet to the compressor

$$h_2 - h_1 = \frac{1}{\eta_{s,c}}.c_p.T_1.\left[\left(\frac{p_2}{p_1}\right)^{\frac{K-1}{K}} - 1\right] \quad (8)$$

Where $\eta_{s,c}$ - isentropic efficiency of the compressor, c_p - mean value of the specific heat at constant pressure between compressor inlet and outlet, T_1 - compressor inlet temperature; p_2/p_1 - compressor pressure ratio.

For steady state engine operation the performance of the turbocharger is determined by the energy balance or the first law of thermodynamics. The mean power consumption of the compressor must be equal to the mean power provided by the turbine.

2.2. Engine simulation model in AVL_BOOST

The test engine is a four-stroke diesel engine, rated output 140 kW/2200 rpm. The engine specification and engine modelled by AVL Boost software are shown in Table 1 and Fig. 1, respectively.

Table 1. The parameter characteristics of the test engine

No	Parameter	Value	Unit
1	Firing order	1-3-5-6-2-4	-
3	Diameter	139	mm
4	Stroke piston	111	mm
5	Connecting rob	230	mm
6	Compression Ratio	16.4	-
7	Rated power	140	kW
8	Maximum torque	740	N.m
9	Injection timing	16	Degree (BTDC)

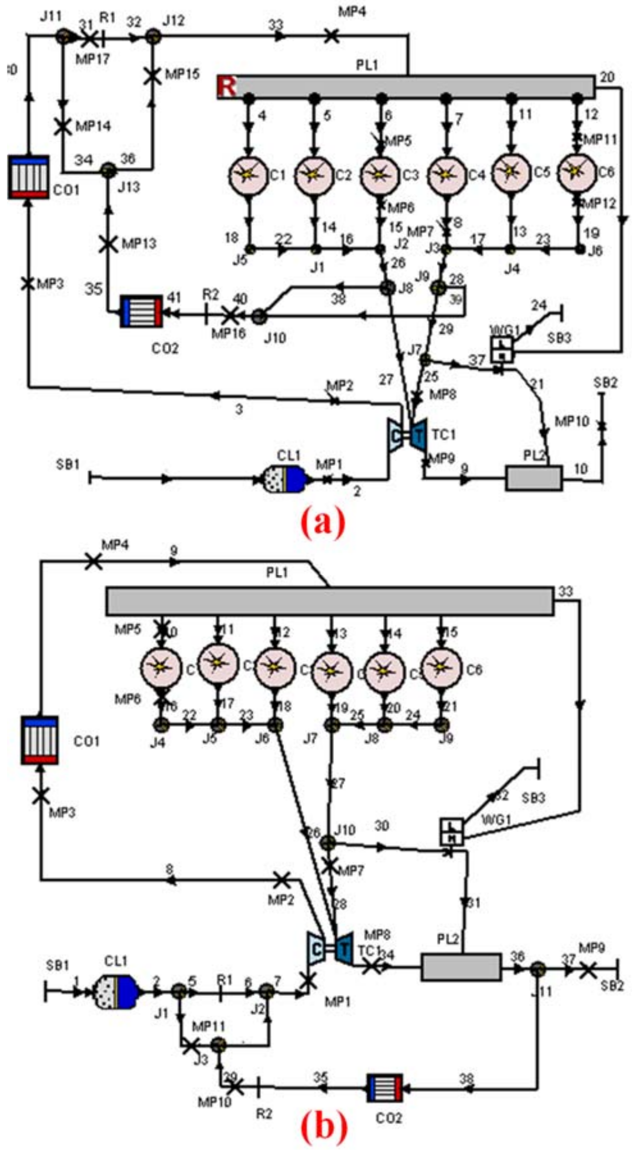

Fig. 1. Engine models built in AVL_BOOST software, (a) original engine, (b) engine retrofitted EGR system

3. Experimental setup of the test engine

The experimental setup to evaluate the performance of the test engine. Fig. 2 shows the experimental setup conducted in Lab of Internal Combustion Engine, Hanoi University of Science and Technology, Vietnam. Testing equipment includes: a test engine, chassis dynamometer, control unit, fuel consumption device, air flow measurement instrument, and temperature sensors. To conduct the specific engine speed, the experiments were conducted with the single cylinder diesel engine corresponding to various engine speeds.

Fig. 2. Experimental setup of the test engine

4. Results and discussion

3.1. Engine performance validation

Fig. 1. shows the comparisons of the engine performance and fuel consumption of the test engine when conducting in the experiment and simulation. As a result, there is a good verification between the experimental and simulation results. Generally, the engine powers of the simulation are higher than that of the experimental test. However, its fuel consumptions are smaller when comparing at the same operating condition. Slight differences between the simulation and experimental data are resulted from the assumptions used in the simulation. However, the differences are very small; then the simulation models of test engine can be used to model the test engine retrofitted EGR system.

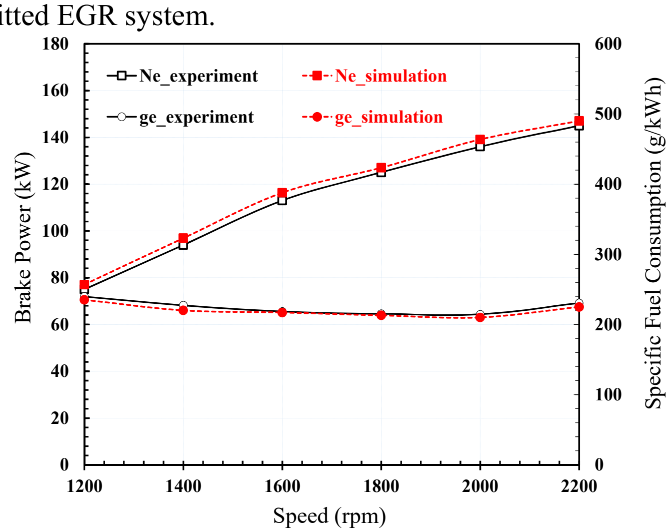

Fig. 3. Comparison of brake power and specific fuel consumption between simulation and experiment results

3.2. Effect of EGR rate on the NOx emissions

Fig. 4 shows the effects of EGR rate on the NOx and smoke emissions corresponding to a variety of EGR rates and engine revolutions. It can be observed that the NOx emission reduced up to approximately 32.5% and smoke increased 33% when the test engine operated at 1400 rpm, 25% load and the EGR rate was 19.98%. These trends were little changed when the engine speed increased to 2000 rpm. The increase of smoke content is not high; consequently, the EGR valve can be controlled to fully open in order to reduce NOx emission.

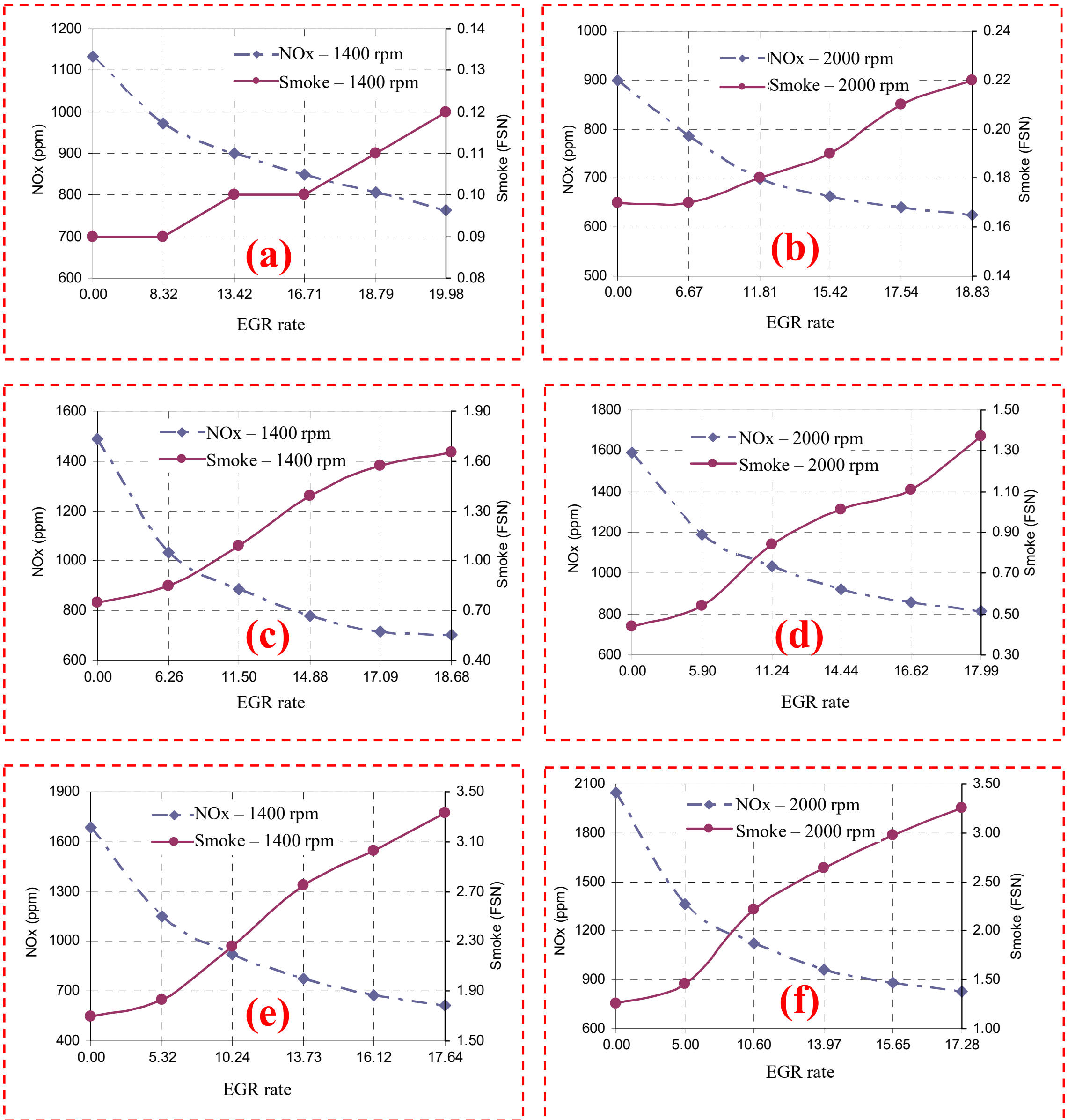

Fig. 4. Effects of EGR rate on the NOx and smoke emissions, (a) at 25% load and 1400 rpm, (b) at 25% load and 2000 rpm, (c) at 50% load and 1400 rpm, (d) at 50% load and 2000 rpm, (e) at 75% load and 1400 rpm, and (f) at 75% load and 20000 rpm

When the engine load increased to 50%, the smoke rate also increased to over 120% and 211% at 1400 rpm and 2000 rpm, respectively. Meanwhile, the NOx emission reduced approximately 50%. Especially, the NOx emission can be reduced to 60% at 70% load; however, the smoke emission also increased dramatically. As a result, it is recommended that DOC and DPF should be applied in order to reduce smoke when retrofitting EGR system for this type of engine.

5. Conclusion

This research presents a solution to reduce the NOx emissions of the bus vehicles in Vietnam. The simulation was conducted in AVL_BOOST software and its result was verified with the experimental test conducted in Lab of internal combustion engine, Hanoi University of Science and Technology. They showed that applying EGR system for the test bus engine can dramatically reduce the NOx emission. However, soot emission was also dramatically increased. Consequently, when using EGR technology to reduce NOx emissions, it is recommended that the solution to control soot emissions should be investigated.

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