

A Potentiality of Dedicated EGR in SI Engines Fueled by Natural Gas for Improving Thermal Efficiency and Reducing NO_x Emission

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ABSTRACT

Recently, a potentiality of Dedicated EGR (D-EGR) concept SI engine has been studied. This concept engine had four cylinders and operated with exhaust gas supplied from the single cylinder to the intake manifold. Compared with conventional SI engines, it was able to increase thermal efficiency and decrease CO, HC, and NO_x emission by the high D-EGR ratio 0.25.

In this study, numerical analysis of a SI engine with D-EGR system with various D-EGR ratios was conducted for detailed understanding the potentiality of this concept in terms of thermal efficiency and NO_x emission. #1 cylinder of assumed engine was used as D-EGR cylinder that equivalence ratio varied from 0.6 to 3.4. Entire exhaust gas from #1 cylinder was recirculated to the other cylinders. The other cylinders run with this exhaust gas and new premixed air and fuel with various equivalence ratios from 0.6-1.0. To study the effect of D-EGR ratio, the number of engine cylinders was considered from 3 to 6, same meaning with D-EGR ratio 0.5-0.2. A Laminar burning velocity and a flame temperature were considered to analyze thermal efficiency and NO_x emission.

The results show that D-EGR system is able to make a similar level of thermal efficiency to that of conventional SI engines. NO_x emission were reduced by decreasing flame temperature while the laminar burning velocity was increased without flame temperature rising due to the influence of H₂ and CO that were included in EGR gas as intermediates.

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INTRODUCTION

Thermal efficiency of SI engines is lower than that of diesel engines because SI engines are operated as low temperature combustion by lower compression ratio than diesel engines relatively [1]. When the amount of fuel supplied and compression ratio are constant, Otto cycle has higher thermal efficiency than Diesel cycle and Sabathe cycle. However, diesel engines have merits to get higher thermal efficiency, like the larger specific heat ratio and the lower pumping loss by no throttling. Diesel engines also have the combustion process under the high compression ratio and it makes higher combustion temperature than SI engines. Although Otto cycle operates as conditions of same quantity of fuel supplied and maximum combustion pressure, this thermal efficiency is less than that of Diesel and Sabathe cycles. Thus, SI engines have many disadvantages compared to Diesel engines in terms of thermal efficiency.

Lean combustion technique is one way of increasing the thermal efficiency of SI engines. Although complete combustion is occurred by stoichiometric condition, heat loss is increased due to the high flame temperature and, therefore, thermal efficiency is decreased. At lean combustion of equivalence ratio about from 0.8 to 0.9, the flame temperature is lower than that of stoichiometric combustion. This affects the decrease of heat loss and it is possible to get the higher thermal efficiency in comparison with stoichiometric combustion process. However, the flame temperature in this case is over than 2200K and it is enough to have as oxidation reaction of N₂ with excess amount of O₂. This phenomenon incurs much NO_x formation. Engine power by lean combustion is lower than stoichiometric combustion due to a combustion process by less quantity of fuel. If heat loss is reduced and the SI engine is run other than lean condition, at the same time, not only the decrease of NO_x emission but also the improvement of thermal efficiency could be achieved.

To reduce a quantity of heat loss, Exhaust Gas Recirculation (EGR) technique is introduced as a concept of low temperature combustion. One of effects of EGR dilution is the decrease of flame temperature at combustion process. As reducing flame temperature, improving thermal efficiency by a heat loss to an in-cylinder wall could be achieved as well as reducing NO_x emission. However, introduction of EGR gas has a coincidence of decreasing flame propagation velocity due to low burning velocity. The decrease of burning velocity incurs lessen degree of constant volume (DCV) and combustion stability. Thus, EGR ratio is limited in spite of its advantages.

Research of engine knock behavior in lean-burn of primary reference fuel by using a single-cylinder SI engine was conducted by Jennifer et al. [2]. To improve the knocking problem, H_2 and CO reformed as the partial oxidation reaction by a plasmatron fuel reformer fueled with gasoline were supplied. Each Research Octane Number (RON) of H_2 and CO is over than 130 and 105, respectively, and these species affected improving knock behavior and stable combustion as well as increasing burning velocity in SI engines. In the previous studies [3, 4], however, the partial oxidation reaction is exothermic and its efficiency is varied from 60% to 80%. To get more H_2 adding water can be considered. Nevertheless, the efficiency can be increased to nearly 90%. It means that the plasmatron fuel reformer has a fuel loss. Besides, an additional cost for the installation of fuel reformer is necessary and the entire engine system composition could be complicated as well as its operation.

To extend the range of EGR ratio and improve combustion stability simultaneously, a new concept of EGR system, called Dedicated EGR (D-EGR), was introduced and adjusted to the SI engine and the scheme of this engine is shown in Figure 1 [5]. The reason of applying D-EGR system to the SI engine was able to estimate by Table 1 that each quantity of H_2 and CO included in exhaust gas by the combustion of SI engine was much more than that of combustion in Diesel engine [6]. As Figure 2 by Ozaki et al. [7], H_2 and CO promoted faster laminar burning velocity. These species have high RON and it was favorable to combust accurately in SI engines. A single cylinder was selected as D-EGR cylinder and its exhaust gas was recirculated to the intake manifold, connected to all cylinders. In this case, D-EGR ratio was a high level, as 0.25 at each cylinder. D-EGR cylinder was operated by rich combustion of gasoline and air. Excess quantity of gasoline had incomplete combustion process at a high temperature condition and H_2 and CO were formed as intermediates of this process. In this manner, although high level of EGR ratio was adjusted to the concept engine, securing combustion stability, improving fuel consumption and emission characteristics of NO_x and CO were possible under the various operation conditions; 2000 rpm / 2 bar BMEP, 2000 rpm / 5 bar BMEP, 1500 rpm / full Load, and full load at 1500, 2000, 3000 and

4000 rpm, respectively. D-EGR system had a high economic effect because of no fuel reforming system. Also, it seemed that fuel loss by reforming was not occurred because all reactive products by rich-combustion in D-EGR cylinder, like hydrocarbon-base species that formed with H_2 and CO at the same combustion process, were recirculated to all cylinders. In this experimental research, however, there was a lack of information of composition variation effect of EGR gas due to the narrow range of equivalence ratio in D-EGR cylinder; from 1.0 to 1.35. It seemed that a reason of narrow condition was due to no consideration of intake and exhaust valve control as a first step of D-EGR concept research.

To overcome a limit of experiments, in this paper, theoretical study was conducted by numerical calculation of elementary reaction to investigate an effect of various equivalence ratios in D-EGR cylinder to the thermal efficiency of the SI engine with D-EGR system. To adjust various D-EGR gas compositions, especially H_2 and CO, equivalence ratio of D-EGR cylinder was varied as larger range than research of Alger et al. [5] and the laminar burning velocity was calculated. One of the most important properties of premixed flames is the laminar burning velocity [8, 9]. The laminar burning velocity is a key factor of determining several important aspects of the combustion process of SI engines; the ignition delay time, the thickness of the wall quench layers, and the minimum ignition energy, etc. It was found that a detailed knowledge of laminar premixed flame was able to provide insights into such properties as heat release rate, flame propagation, emission characteristics. Therefore, the results of accurate measurements of laminar premixed flames play a key role for understanding flame characteristics. Although the majority of fuel is combusted by turbulent combustion, data on laminar burning velocity is needed as input to many turbulent combustion models. Also, in internal combustion engines the initial combustion is laminar, thus laminar burning velocity is an important data. In this study, laminar burning velocity was used to estimate DCV while heat loss was forecasted by flame temperature. To accurate estimate thermal efficiency and heat loss of the concept engine, Functions of Wiebe and Woschni were used and it is suggested the potentiality of D-EGR system in SI engines.

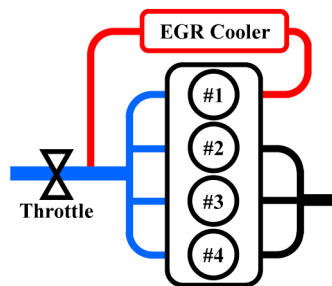


Figure 1. Concept of Dedicated EGR engine of SwRI [3]; exhaust gas from cylinder 1 recirculated to the entire cylinders with new premixed fuel-air

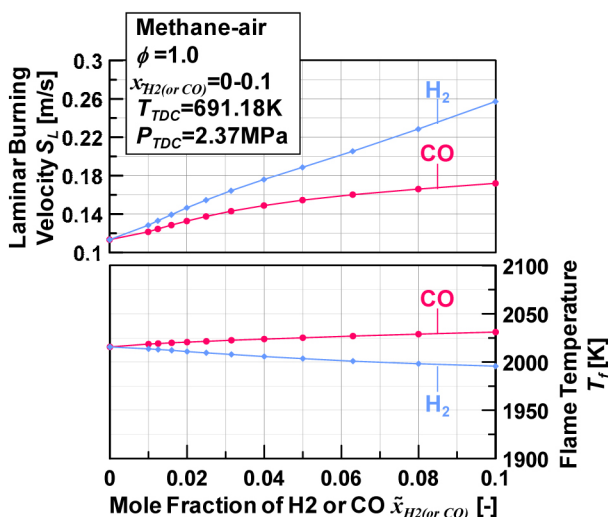


Figure 2. Influence of H_2 and CO on laminar burning velocity and flame temperature of premixed methane-air combustion [7]

Table 1. Major and minor compositions of exhaust gas

Major Compositions (mole than 1%.mol)	H_2O , CO_2 , N_2 , O_2 , $CO^{(a)}$, $H_2^{(a)}$
Minor Compositions (less than 1%.mol)	SO_2 , SO_3 , NO , NO_2 , $HCHO$, $HCOOH$, CH_3OH , C_nH_m , $CO^{(b)}$, $H_2^{(b)}$, Smoke, etc

^(a) SI engine

^(b) Diesel engine

ANALYSIS METHOD

Tool of Numerical Analysis

To calculate thermal efficiency, the turbulent burning velocity is needed which has a relationship with the laminar burning velocity. To calculate the laminar burning velocity, therefore, PREMIX code in CHEMKIN-PRO [10] for one-dimensional calculation was used. For ensuring the sufficient calculation, estimated zone width was set to 10cm. Fuel was selected as methane that consisted 90% of natural gas. Atomic ratio of hydrogen to carbon of methane is $A_{H,C}=4:1$, which is larger than that of gasoline. It was expected to get much more H_2 as well as CO by rich combustion in D-EGR cylinder due to an incomplete combustion. Elementary reaction scheme of methane was chosen as GRI-Mech Version 3.0 that contained 53 of chemical species and 325 of elementary reactions. Each component of the intake air was assumed to 78 % of N_2 , 21 % of O_2 and 1 % of Ar, as volume percents, respectively.

Concept of a SI Engine with D-EGR System

Figure 3 is a scheme of the SI engine with D-EGR system in this study. It had four-cylinder and #1 cylinder took a role of D-EGR cylinder. D-EGR gas was supplied to #2-#4 cylinders by 0.33 of D-EGR ratio. In case of Alger et al. [5], exhaust gas by #1 cylinder supplied to all cylinders. However, estimate of D-EGR gas composition by #1 cylinder was not easy. Because,

in #1 cylinder, the exhaust gas composition by n^{th} cycle combustion had an affect on the premixed gas composition supplied to the next cycle combustion and the loop calculation for D-EGR system could be complicated. In this study, therefore, it was suggested to recirculate exhaust gas from #1 cylinder to #2-#4 cylinders, as D-EGR ratio 0.33 ($\gamma_{D-EGR}=0.33$). Thus, the composition of D-EGR gas from #1 cylinder could be estimated by the equivalence ratio in #1 cylinder only. D-EGR gas by #1 cylinder was supplied to #2-#4 cylinders with new premixed air-fuel to form $\phi_{\#2}=0.6-1.0$. In general, the maximum thermal efficiency in SI engine occurred at lean condition, thus it was needed to confirm this phenomenon in the SI engine with D-EGR system.

RESULT

Influences of Temperature and Pressure of Premixed Methane-Air on Laminar Burning Velocity

In-cylinder temperature and pressure at top dead center (TDC) were calculated by the assumed SI engine that had 82 mm of bore, 64 mm of stroke, and 10.76 of compression ratio, respectively. At the same time, the calculation of laminar burning velocity was conducted at the conditions of adiabatic compression.

Figure 4 shows laminar burning velocity that was affected by temperature and pressure of unburned premixed methane-air. Red line on the temperature-pressure plane means the condition of adiabatic compression. Although compression ratio on this line reached at 10.76, laminar burning velocity was not varied significantly compared to compression ratio 1.0.

Black lines are the trend of laminar burning velocity at various temperature and pressure. When only temperature varied at the same pressure, laminar burning velocity increased dramatically. On the contrary, when only pressure varied at the same temperature, that velocity seemed to decrease. By these results, it seemed that high pressure of premixed methane-air incurred the negative factor to laminar burning velocity.

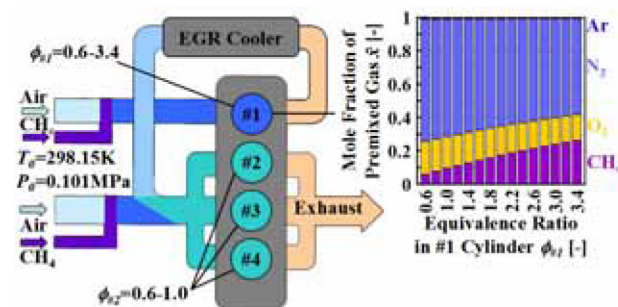


Figure 3. Scheme of the SI engine with Dedicated EGR system and mole fraction of premixed gas that was intaked to #1 cylinder; exhaust gas by #1 cylinder was supplied to #2-#4 cylinders by EGR ratio 0.33

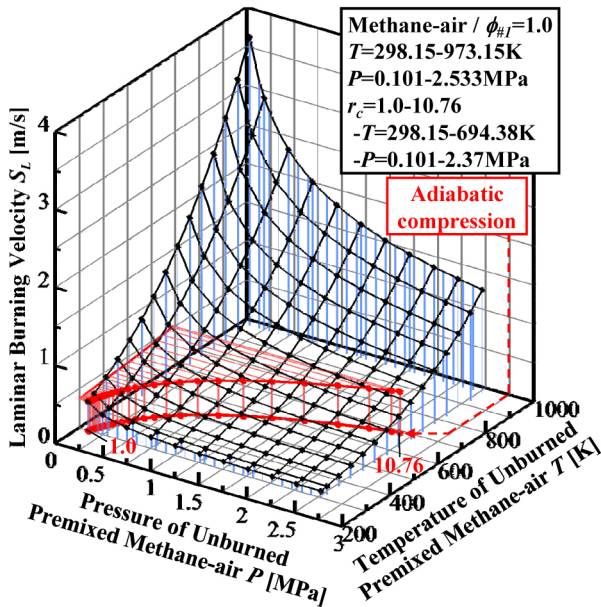


Figure 4. Influences of various adiabatic compression ratios, temperature and pressure of unburned premixed methane-air on laminar burning velocity

Composition of D-EGR Gas by Combustion of #1 Cylinder

Figure 5 shows the mole fraction of exhaust gas that formed by the combustion process of $\phi_{\#1}=0.6-3.4$. H_2O and CO_2 were exhausted as higher level at the condition of stoichiometric combustion than the other equivalence ratios. When rich combustion occurred, each mole fraction of H_2 and CO was increased until $\phi_{\#1}=2.2$ and 2.0 , respectively. In this figure, remarkable chemical species of D-EGR system, H_2 and CO , are indicated. As referred, each RON of H_2 and CO is high, H_2 is larger than 130 and CO is 106, and it has the improvement of combustion stability with high EGR tolerance [11]. In the case of H_2 , especially, its thermal conductivity and this could affect fast laminar burning velocity without flame temperature rising that was analogized [12]. When lean combustion occurs, each quantity of H_2 and CO is extremely less, or almost zero. When equivalence ratio is larger than 1.0, each quantity of H_2 and CO became increased. Each mole fraction of H_2 and CO became the maximum at equivalence ratio 2.2 and 2.0, respectively. When equivalence ratio is higher than these values, mole fraction of both H_2 and CO started to decrease. Exhaust gas as these compositions was mixed with new premixed methane-air as the equivalence ratio 0.6-1.0, and supplied to #2-#4 cylinders. In this case, D-EGR ratio in #2-#4 cylinders was 0.33.

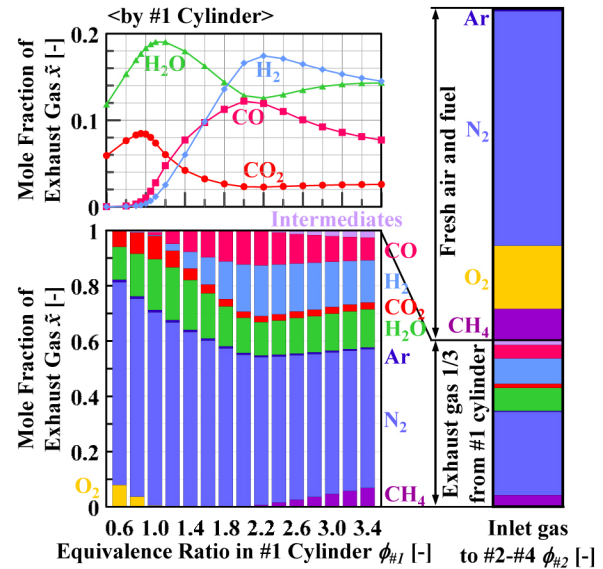


Figure 5. Mole fraction of exhaust gas composition by combustion of various equivalence ratios of premixed methane-air in #1 cylinder at TDC condition and the mixing method for new inlet gas to #2-#4 cylinders by D-EGR ratio 0.33

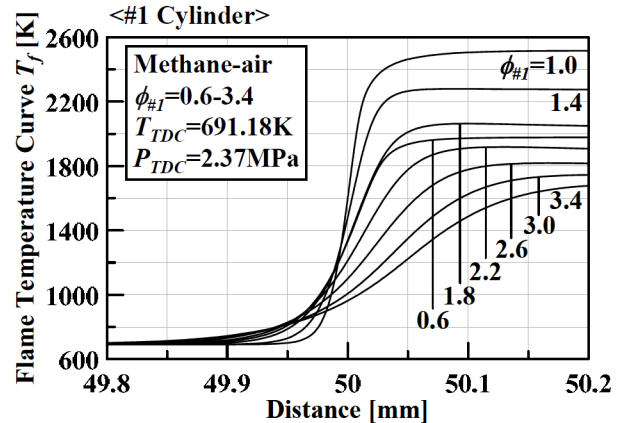


Figure 6. Temperature curves of #1 cylinder that varied by equivalence ratio in #1 cylinder

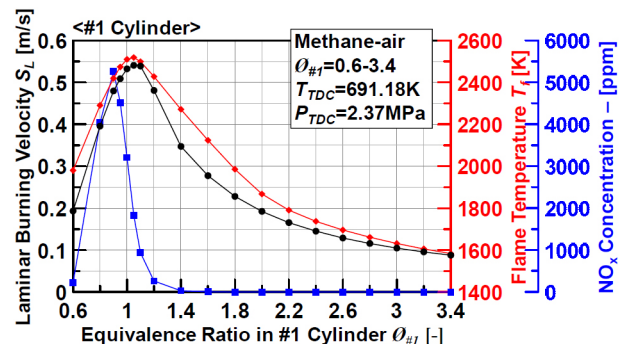


Figure 7. Influences of equivalence ratio variation on laminar burning velocity, flame temperature and NO_x concentration in #1 cylinder