A Potentiality of Dedicated EGR System for Improving Thermal Efficiency in Natural Gas SI Engines

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ABSTRACT: Object of this study is to realize improving thermal efficiency and emission reduction with low temperature combustion. EGR is a solution to low temperature combustion, however it occurs burning velocity decrease. To prevent this phenomenon, Dedicated EGR was adjusted to SI engine run by natural gas. The entire exhaust gas of a single cylinder operated by the equivalence ratio varied from lean to rich was recirculated to the other cylinders. Due to H2 and CO included in EGR, slowed burning velocity problem was overcome with increasing degree of constant volume, thus thermal efficiency could be improved.

KEY WORDS: Heat Engine, Spark Ignition Engine, Numerical Calculation, Exhaust Gas Recirculation, Laminar Burning Velocity, Low Temperature Combustion, Thermal Efficiency [A1]

1. Introduction

Present internal combustion spark ignition (SI) engines are strongly wished for the improvement of both thermal efficiency (\(\eta\)) and emission problems. When exhaust gas recirculation (EGR) system is applied to the SI engine, low temperature combustion is occurred and it can reduce the heat loss (\(Q_{\text{heat loss}}\)) and nitrogen oxides (NOx) emission. However, when EGR ratio (\(\gamma\)) is increased, the laminar burning velocity (\(S_L\)) is decreased and the stability of engine operation is worse.

Alger et al(1) suggested a new EGR system as Fig. 1 and it was called dedicated EGR (d-EGR) system. In this concept engine, the entire exhaust gas from #1 cylinder was recirculated to the entire cylinders with new premixed fuel-air as \(\phi_{#1}=1.0\). #2-#4 cylinders were operated as stoichiometric condition. The result shown that although EGR ratio (\(\gamma\)) was 0.25, the combustion stability was recovered by hydrogen (H2) and carbon dioxide (CO) added. Because each research octane number (RON) of H2 and CO was RONH2>130 and RONCO=106, respectively. In addition, fast burn rate was led by H2 addition. Therefore, the combustion stability at high EGR ratio was improved. However, some information was needed to understand the influence of \(\phi_{#1}\) varied on the composition of EGR gas and the combustion characteristics of #2-#4 cylinders.

Ozaki et al(2) tried numerical analysis of d-EGR system fueled with methane (CH4). In the study, the composition of exhaust gas from #1 cylinder was varied and this was affected by \(\phi_{#1}\) that range was 0.6≤\(\phi_{#1}\)≤3.4. Exhaust gas was mixed with new premixed CH4-air as \(\phi_{#2}=1.0\) and supplied to the intake manifold that connected to #2-#4 cylinders, as Fig. 2. The results shown that in spite of \(\gamma=0.33\), the combustion stability was increased due to a large amount of H2 and CO. Since H2 increased \(S_L\) without increasing flame temperature (\(T_f\)). The increase of \(S_L\) is led the high degree of constant volume (\(\eta_{gl}\)), therefore high \(\eta\) was achieved. This \(\eta\) was higher than that of stoichiometric combustion in the SI engine without EGR system.

In the present paper additional research are carried out to investigate the case of combustion with lean premixed CH4-air in #2-#4 cylinders and its effects on the SI engine with d-EGR system, because the maximum thermal efficiency of the ordinary SI engine is achieved with lean combustion. So, it should be needed to confirm whether this phenomenon also occurs in d-EGR system or not.

2. Influences of In-Cylinder Condition and Equivalence Ratio on Laminar Burning Velocity and d-EGR Gas Composition

This chapter introduces the SI engine with d-EGR system. In-cylinder temperature and pressure of assumed SI engine were set the calculation condition of \(S_L\) by the elementary reaction numerical analysis.

Fig. 1 Concept of dedicated EGR engine of SwRI(1); exhaust gas from #1 cylinder recirculated to the entire cylinders with new premixed fuel-air
2.1. Concept of the SI engine with d-EGR system of the present study

Fig. 2 shows the scheme of the SI engine with d-EGR system in the present study. It had four cylinders in which #1 cylinder had the role as the d-EGR cylinder. #1 cylinder was operated by various equivalence ratio $0.6 \leq \phi_{#1} \leq 3.4$, since this study is interested in the composition of EGR gas varied by $\phi_{#1}$. Alger et al. [3] exhaust gas from #1 cylinder was supplied to all cylinders. In the present study, however, a simpler system is considered for the detailed study as the exhaust gas is not fed to #1 cylinder but only to #2-#4 cylinders, thus $\gamma = 0.33$. At the same time, EGR gas was mixed with new premixed CH₄-air to make the condition of $0.6 \leq \phi_{#2-#4} \leq 1.2$. The calculation of the condition $1 \leq \phi_{#2-#4} \leq 1.2$ was carried out to consider the combustion characteristics around the stoichiometric condition.

2.2. Numerical analysis tool for the calculation of laminar burning velocity and flame temperature

To calculate $\eta$, Wiebe function and Woschni’s heat transfer coefficient were introduced in this study. Both equations were able to calculate by the $S_I$ data. $S_I$ is provide the basic information to estimate heat release rate, flammability limit and emission characteristics [3]. This study was assumed that the initial combustion proceeded from the laminar flame. In this study, therefore, we tried to calculate $\eta$ of all cylinders in the SI engine with d-EGR system by using $S_I$. To calculate $S_P$, PREMIX code of CHEMKIN-PRO [3] was employed. It is the one-dimensional code for the calculation of $S_I$ and $T_p$, and the calculation concept is shown in Fig. 3. On the x-axis in this figure, a distance is an estimated zone width where the calculation is conducted, from 0 to 100mm. An actual flame thickness is too narrow to confirm. To see the temperature variation, the distance near the flame zone was magnified. For ensuring the sufficient calculation, estimated zone width was set to 100mm. CH₄ was selected which is over the 90% content of natural gas. Atomic ratio of hydrogen to carbon of CH₄ is $A_H:C=4:1$, which is larger than that of the other hydrocarbons fuel. It was expected to get much more H₂ as well as CO by rich combustion in d-EGR cylinder as intermediates. The elementary reaction scheme of CH₄ was chosen as GRI-Mech version 3.0 that contained 53 of the chemical species and 325 of the elementary reactions. Each component of the intake air was assumed by 0.78 of nitrogen (N₂), 0.21 of oxygen (O₂) and 0.01 of argon (Ar) as volumetric fraction, respectively.

The operation condition of each #1 cylinder and #2-#4 cylinders was different. #1 cylinder was operated as the d-EGR cylinder to supply EGR gas to #2-#4 cylinders. #2-#4 cylinders were operated with EGR gas and new premixed CH₄-air. It is note that the calculation methods of #2-#4 cylinders were same, thus the result of #2 cylinder was represented that of #2-#4 cylinders.

2.3. Influences of temperature and pressure of premixed CH₄-air on laminar burning velocity and flame temperature

Temperature and pressure at top dead center ($T_{TDC}$ and $P_{TDC}$) of cylinder were determined for the assumed engine that compression ratio of adiabatic process was $\varepsilon=10.76$. The calculation for $S_I$ and $T_p$ was conducted at the condition of TDC.

Fig. 4 shows the calculation result of $S_I$ which was affected by temperature ($T_u$) and pressure ($P_u$) of unburned premixed CH₄-air. "~" lines are the trend of $S_I$ variation while $T_u$ and $P_u$ were varied. When $T_u$ was varied only at the same $P_u$, $S_I$ was increased.
dramatically. On the contrary, when $P_c$ varied only at the same $T_c$, $S_L$ was decreased. The results show that high pressure of premixed CH$_4$-air suppresses the increment of $S_L$. “—” line on the $T_S$-$P_a$ plane indicates the adiabatic compression condition. Although $e$ on this line reached at 10.76, 694.38K and 2.37MPa, $S_L$ was varied a little compared with compression ratio 1.0 at 298.15K and 0.101MPa.

2.4. Composition of Dedicated EGR gas by combustion of #1 cylinder

Fig. 5 shows the mole fraction of exhaust gas that formed by the combustion process in #1 cylinder with $0.6 \leq \phi_1 \leq 3.4$ of premixed CH$_4$-air. Near the stoichiometric condition, the amount of each H$_2$O and CO$_2$ was larger than the other equivalence ratios. In the case of rich combustion, each mole fraction of H$_2$ and CO was the maximum at $\phi_1 = 2.2$ and 2.0, respectively. When $\phi_1 > 2.2$, excess amount of CH$_4$ was not reacted due to the amount of poor O$_2$ and remained in exhaust gas evidently. In this process, the heat was not enough to form H$_2$ and CO from CH$_4$.

2.5. Influences of equivalence ratio of premixed CH$_4$-air on laminar burning velocity and flame temperature

Fig. 6 shows the temperature distributions at from the front to the rear of the flame surface in #1 cylinder. The meaning of temperature distribution is a temperature between front and rear near the laminar flame surface. Temperature distribution was effected by $\phi_1$ varied. When $\phi_1$ was close to 1.0, the gradient of temperature profile was steep and the increasing tendency of temperature was larger due to the fast chemical reaction by the stoichiometric combustion. In the contrary, $\phi_1 < 1$ or $1 < \phi_1$, the gradients of temperature profiles were gradual. Since premixed CH$_4$-air was not stoichiometric, chemical reaction was less active.

Fig. 7 shows $S_L$ and $T_f$ at $0.6 \leq \phi_1 \leq 3.4$ in #1 cylinder. Both values were the maximum near $\phi_1 = 1.0$ due to the stoichiometric combustion. When $\phi_1$ was smaller or larger than 1.0, $S_L$ and $T_f$ were decreased as similar trend. In Fig. 6, since when $\phi_1$ was close to 1.0, the gradient of temperature profile became steep and this had the same meaning that the chemical reaction became fast. Thus, $S_L$ was increased.

As Wiebe function$^{(5)}$, the increase of $S_L$ is related to the in-cylinder pressure rising, the piston work is increased and the amount of engine work is increased. As Woschni’s heat transfer coefficient$^{(6)}$, if $T_f$ is increased, it is connected with the in-cylinder
temperature rising, therefore the amount of heat loss ($Q_{heating}$) is increased. As Fig. 8, it is able to confirmed the relationship between $S_t$ and $T_f$ as the variation of $\phi_f$. If the line is horizontal to x-axis, it means the variation of $S_t$ with constant $T_f$. In the contrary, if the line is horizontal to y-axis, it means there is the variation of $T_f$ with constant $S_t$. The former has the meaning of the variation of the amount of engine work with constant $Q_{heating}$. The latter has the meaning of varying amount of $Q_{heating}$ without the engine work variation. In Fig. 8, the line is diagonal from the bottom left corner to the upper right corner. Thus, it is estimated that when $\phi_f$ is increased from 0.6 to 1.05, the amount of each engine work and $Q_{heating}$ will be increased. When $\phi_f$ is larger than 1.05 and increased until $\phi_f=3.4$, both engine work and $Q_{heating}$ will be decreased.

3. A Potential for Improving Thermal Efficiency of the SI Engine with d-EGR System

In this chapter, the influence of $\phi_f$ on the thermal efficiency ($\eta$) of the SI engine with d-EGR system is investigated. In addition, the calculation of the lean premixed CH$_4$-air combustion in #2-#4 cylinders was conducted at the condition of $\phi_f$ that occurred the maximum $\eta$, because general SI engines has the maximum $\eta$ at the condition of lean combustion.

3.1. Influences of H$_2$ and CO content of EGR gas from #1 cylinder on the laminar burning velocity and the flame temperature of #2-#4 cylinders

Fig. 9 shows the temperature distributions that were affected by $\phi_f$ as composition variation of EGR gas, especially H$_2$ and CO. In the case of lean combustion, each amount of H$_2$ and CO products was almost zero. When the combustion was not in stoichiometry, the amount of H$_2$O and CO$_2$ was less than $\phi_f=1.0$. Thus, in #2 cylinder the flame thickness was thinner and $T_f$ was higher than that of $\phi_f=1.0$. When $\phi_f>1.0$, the amount of H$_2$ and CO was increased while the amount of H$_2$O and CO$_2$ was decreased. As Ozaki et al.$^{11}$, the former affected the increasing chemical reaction speed while the latter affected the increasing $T_f$. The flame characteristic in #2 cylinder was affected by this phenomenon, the flame thickness was narrower and $T_f$ was higher than that of $\phi_f=1.0$ case.

Fig. 10 shows the influences of various $\phi_f$ on the combustion in #2-#4 cylinders in terms of $S_t$ and $T_f$. Each result was the lowest around $\phi_f=1.0$ due to the reaction energy absorption by much amount of H$_2$O and CO$_2$ in EGR gas, as Fig. 5. When $\phi_f>1.0$, both $S_t$ and $T_f$ were increased. However, $T_f$ increased gradually with the increase of $\phi_f$ and it was not increase anymore from $\phi_f>2.2$. In the case of $S_t$, the tendency was similar to $T_f$, however this value was increased gradually even until $\phi_f=3.4$. It shows that H$_2$ in EGR gas increased $S_t$ without increasing $T_f$ as mentioned in previous chapter.

Fig. 11 shows the relationship between $S_t$ and $T_f$ as varied $\phi_f$ at constant $\phi_e=1.0$. Compared with Fig. 8 and Fig. 11, when $1.05<\phi_f<3.4$ the line of Fig. 11 was more gradual than that of Fig. 8. Also, in the case of $0.6<\phi_e<1.05$, the line was also more gradual than that of Fig. 8. Thus, it is estimated that when $\phi_f$ was lean or rich, $\eta$ of #2 cylinder would be increased due to the larger growing gap of $S_t$ than that of $T_f$.

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**Fig. 9** Influence of equivalence ratio variation in #1 cylinder on the laminar burning velocity and the flame temperature in #2-#4 cylinders

**Fig. 10** Influence of equivalence ratio in #1 cylinder on the laminar burning velocity and the flame temperature in #2 cylinder

**Fig. 11** Relationship between the laminar burning velocity and the flame temperature in #2 cylinder by equivalence ratio variation of #1 cylinder in the SI engine with dedicated EGR system; $\phi_f=1.0$(constant)
3.2 Calculation method of thermal efficiency and heat loss

To figure out the amount of engine work, heat release rate \(dQ/d\theta\) was calculated by using Wiebe function (1)(5) and the mean temperature and the in-cylinder pressure were estimated by equation (3). As equation (4), the combustion duration \((\theta_c)\) in Wiebe function was assumed by the turbulent burning velocity \(ST\) that was proportional to \(S_t\).

\[
\frac{Q}{Q_{fuel}} = 1 - \exp\left\{-\left(\frac{\theta - \theta_c}{\theta_p}\right)^{m+1}\right\}
\]

\[
\frac{dQ}{d\theta} = a Q_{fuel} \left\{m+1\right\} \left(\frac{\theta - \theta_c}{\theta_p}\right)^m \times \exp\left\{-\left(\frac{\theta - \theta_c}{\theta_p}\right)^{m+1}\right\}
\]

\[
\frac{dQ}{d\theta} = \frac{1}{k-1} \left(\frac{v dP}{dt} + k_0 \frac{dT}{dt}\right)
\]

\[
\theta_p = \frac{b}{2} N_c \times 360
\]

\[
a = -\ln(1-0.999)
\]

where, \(\theta\) is the crank degree at ignition timing [degree], \(\theta_c\) is the crank angle of ignition delay [degree], \(\theta_p\) is the combustion duration (an equation of \(S_t\) [degree], \(m\) is the parameter which is decided by the form of Wiebe function, \(Q_{fuel}\) is the amount of fuel supplied (In the case of #1 cylinder, the amount of fuel supplied is multiplied by a combustion rate which is varied by \(\phi_{t1}\), \(N_c\) is the rotational engine speed [rpm], \(b\) is the diameter of piston bore [m], \(S_t\) is the turbulent burning velocity [m/s], \(S_c\) is the laminar burning velocity [m/s], and the specific heat ratio is constant as \(k=1.39\), respectively. It was set as \(\theta_c=0\), \(m=5\), \(a=6.9\) (a value for 99.9% of accumulated heat at the end of combustion process), \(N_c=1500\), \(S_t/S_c=25\), and the ignition timing was -10 deg.ATDC, respectively.

\(Q_{heatloss}\) was calculated by Woschni’s heat transfer coefficient (5). In this study, however, turbulent flow was not considered and temperature of cylinder wall was assumed by 498.15K(60).

Using the quantity of work that calculated by Wiebe function and supplied fuel, thermal efficiency of all cylinders \((\eta_{#1-#4})\) was calculated by equation (6). Temperature, pressure of in-cylinder and \(Q_{heatloss}\) were used to calculate thermal efficiency of all cylinders considered \(Q_{heatloss} (\eta_{heatloss_{#1-#4}})\) by equation (7).

\[
\eta_{#1-#4} = \frac{W_{#1} + 3W_{#2}}{Q_{fuel_{#1-#4}}}
\]

\[
\eta_{heatloss_{#1-#4}} = \frac{W_{#1} - Q_{heatloss_{#1}} + 3(W_{#2-#4} - Q_{heatloss_{#2-#4}})}{Q_{fuel_{#1-#4}}}
\]

where, \(W_{#2}\) is the amount of work by #1 cylinder [J], \(W_{#2-#4}\) is the amount of work by #2-4 cylinders [J], \(Q_{heatloss_{#2-#4}}\) is the amount of heat loss by #1 cylinder [J], \(Q_{heatloss_{#2-#4}}\) is the amount of heat loss by #2-4 cylinders [J], \(Q_{fuel_{#2-#4}}\) is total fuel supplied to all cylinders [J], respectively. The calculation was carried out between the intake valve close timing (-145deg.ATDC) and the exhaust valve open timing (145deg.ATDC).

3.3 Comparison of thermal efficiency between #1 and #2 cylinders; influence of equivalence ratio of #1 cylinder on stoichiometric combustion of #2 cylinder

Fig. 12 shows the results of \(\eta_{#1}\) "" line is #1 cylinder and "" line is #2 cylinder. Since fast \(S_t\) affected the increase of in-cylinder pressure at the combustion process, the tendency of each \(\eta_{#1}\) was similar with \(S_t\) in Fig. 7 and 10, respectively.

In Fig. 13, thermal efficiency of #1 cylinder (\(\eta_{#1}\) "" line; the calculation result of \(\eta_{#1-#4}\) that consider only exhaust loss) was the highest at \(\phi_{t1}=0.9\). However, when \(Q_{heatloss}\) ("" line; the calculation results of \(\eta_{heatloss_{#1-#4}}\) that consider both exhaust loss and heat loss) was considered, the maximum \(\eta_{#1}\) was occurred at \(\phi_{t1}=0.8\). As Fig. 7, since \(T_f\) at \(\phi_{t1}=0.9\) was higher than that of \(\phi_{t1}=0.8\), \(Q_{heatloss}\) was increased, \(\eta_{#1}\) was decreased more than \(\phi_{t1}=0.8\).

Fig. 14 shows thermal efficiency of #2 cylinder (\(\eta_{#2}\) which was affected by \(\phi_{t2}\) varied. When only exhaust loss was considered, the tendency of \(\eta_{#2}\) was similar with \(\eta_{#1}\). When exhaust loss and \(Q_{heatloss}\) were considered, \(\eta_{#2}\) was not increased compared with only consideration of exhaust loss. Since \(S_t\) affected to the increase of heat transfer coefficient (\(\eta\)), it is increased gradually until \(\phi_{t2}=3.4\) in Fig. 10, and the amount of H2 in EGR gas was the maximum at \(\phi_{t2}=2.2\). As these reasons, the maximum \(\eta_{#2}\) was occurred at \(\phi_{t2}=2.2\).
3.5. Comparison of thermal efficiency between lean and stoichiometric combustion in #2 cylinder

Previous chapter shown that $\eta_{22}$ was the highest at $\phi_{22}=2.2$ due to the influence of H$_2$. Therefore, the effect of lean combustion was investigated at $0.6 \leq \phi_{22} \leq 1.2$. The reason of this investigation was referred in introduction of this paper.

Fig. 16 shows the $T_f$ distributions in #2 cylinder. The highest $T_f$ occurred at $\phi_{22}=1.1$ with the steepest temperature curve. When $\phi_{22}$ was smaller, $T_f$ was lower and its slope was decreased, therefore $S_L$ was slow as Fig. 17.

Fig. 18 shows the relationship between $S_L$ and $T_f$ at in #2 cylinder. It is expected that the highest $\eta$ would be occurred at the condition of $\phi_{22} \geq 1.0$. Since $S_L$ was too slow to achieve high $\eta$ when lean combustion was occurred in #2 cylinder.

Fig. 19 shows the $\eta_{gl}$ of #2 cylinder. The results show that the maximum $\eta_{gl}$ was occurred at $\phi_{22}=1.1$. When $\phi_{22}>1.1$, the $\eta_{gl}$ would be occurred at $\phi_{22}=0.8$. In Fig. 14, $\eta_{gl}$ was decreased continually after $\phi_{22}=0.8$.

Fig. 15, combined Fig. 13 and 14, is the thermal efficiency of all cylinders ($\eta_{21-4c}$) in the SI engine with d-EGR system. In Fig. 13, $\eta_{21}$ was decreased continually after $\phi_{21}=0.8$. In Fig. 14, $\eta_{22}$ after $\phi_{22}=1.8$ is roughly similar. Although $\eta_{22}$ was the highest at $\phi_{22}=2.2$ in Fig. 14, $\eta_{21}$ was lower than that of $\phi_{22}=1.8$ in Fig. 13. Therefore, the maximum $\eta_{21-4c}$ was occurred at $\phi_{21}=1.8$.
combustion in #2 cylinder incurred low $\eta_{SL}$ as Fig. 19, thus $\eta_{SL}$ was decreased.

Fig. 21 shows $\eta_{SL,4}$ of the SI engine with d-EGR system when lean combustion was occurred in #2-#4 cylinders. As similar with Fig. 20, $T_f$ was decreased and $Q_{heatloss}$ was reduced, no effect of lean combustion expected in chapter 3 was found. Since $S_L$ was too low to achieve high $\eta$.

Fig. 22 shows the comparison of $\eta_{SL,4}$ among $\phi_1=1.0, 1.8$ and 2.2 while lean combustion occurred in #2-#4 cylinders. When #1 cylinder was operated by rich combustion, $\eta_{SL,4}$ was always higher than the case of stoichiometric combustion in #1 cylinder. At $\phi_1=1.0$, because each amount of H$_2$O and CO$_2$ was large and that of H$_2$ and CO was rarely formed in EGR gas, it was similar with the result of $\gamma=0.33$ in general EGR system. The maximum $\eta_{SL,4}$ that considered $Q_{heatloss}$ of each $\phi_1=1.8$ and 2.2 was 0.413 and 0.405, respectively. The reason of this phenomenon was that $\eta$ that considered $Q_{heatloss}$ of $\phi_1=1.8$ in #1 cylinder was less than the case of $\phi_1=2.2$ in #1 cylinder, as Fig. 13. Although $S_L$ at $\phi_1=2.2$ was faster than the case of $\phi_1=1.8$, $T_f$ became higher due to the influence of CO that was included as more quantity than that of $\phi_1=1.8$. CO combustion was improved $S_L$ without $T_f$ increasing. CO also improved $S_L$ but its effect was less than H$_2$ combustion and accompanied $T_f$ increasing$^{(2)}$. This occurred more $Q_{heatloss}$ than that of the condition of $\phi_1=1.8$.

4. Conclusion

To increase $\eta$ of SI engines, low temperature combustion is considered and it is able to reduce $Q_{heatloss}$. However, $S_L$ is decreased simultaneously and $\eta$ is decreased. Therefore, the fast $S_L$ during low temperature combustion is needed for high $\eta$.

A potential of d-EGR system, which is expected to increase $\eta$ of the four-cylinder SI engine fueled with CH$_4$ was investigated. $\phi_2$ was varied from 0.6 to 3.4 to investigate the influence of the EGR gas composition on $S_L$ and $T_f$, especially H$_2$ and CO. $\phi_{2-4}$ was varied from 0.6 to 1.2 to investigate the effect of lean combustion for high $\eta$. $S_L$ and $T_f$ were calculated by PREMIX code of CHEMKIN-PRO which was able to the elementary analysis. Wiebe function and Woschni's heat transfer coefficient
were applied to estimate $\eta$ with $Q_{\text{heatloss}}$.

1) When combustion of rich premixed CH$_4$-air occurs, not only H$_2$O and CO$_2$ but also H$_2$ and CO are exhausted as intermediates. From the results of the present investigations, H$_2$O and CO$_2$ caused to decrease the $T_f$. On the other hand, H$_2$ and CO increased the $S_L$. Especially, H$_2$ increased $S_L$ without increasing $T_f$. In the case of CO, $T_f$ increased slightly.

2) In #1 cylinder, the maximum $\eta$ considered $Q_{\text{heatloss}}$ was achieved at $\phi_{#1}=0.8$. When $\phi_{#1}$ was increased, $\eta_{#1}$ was decreased. $\eta_{#2}$ reached the highest at the condition of $\phi_{#1}=2.2$ and $\phi_{#2}=1.0$. $\eta_{#1-#4}$ was the maximum at the condition of $\phi_{#1}=1.8$ and $\phi_{#2}=1.0$, since $Q_{\text{heatloss}}$ of #2 cylinder at $\phi_{#2}=2.2$ was larger than the case of $\phi_{#1}=1.8$.

3) When combustion of lean premixed CH$_4$-air occurred in #2-#4 cylinder, $\eta_{#2-#4}$ was decreased. Similarly, $\eta_{#1-#4}$ of all cylinders was decreased that was different from the SI engine without EGR system. Thus, the SI engine with d-EGR system is recommended to operate by stoichiometric combustion in #2-#4 cylinders rather than other $\phi_{#2-#4}$.

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