Study on Hydrogen Combustors with Two-Staged Combustion Method for Micro Gas Turbines
(Combustion Characteristics of Coaxial Type Injectors)*

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Abstract
With the goal of developing advanced hydrogen combustors for micro gas turbines with low NOx emission at high temperatures up to $T_{in}=1700\, ^\circ C$, single coaxial injectors with two-staged (RQL; rich-quench-lean) combustion method were built, based on the excellent combustion features of hydrogen. The injector consisted of an inner pipe issuing rich premixed gases, and an outer concentric annular hole injecting secondary air at high velocity. The combustion characteristics of small test combustors, which had the injector and a combustion tube of 30mm diameter, were examined at room temperature and atmospheric pressure, concerning NOx emissions, temperature distributions, combustion efficiencies, total pressure losses and flame stability limits. The optimum equivalence ratio of the rich premixed inner flame was confirmed to be about 2.0 for the viewpoint of decreasing NOx emission and total pressure loss. The NOx concentrations were reduced by decreasing the area of the coaxial air hole, and were in inverse proportion to the ratio of the momentum of the coaxial air to the premixed gas. For the injector with the smallest area of air hole, having pressure loss of below 4%, NOx emission levels were low enough, and high combustion efficiencies over 99.95% were attained in the range from 0.3 to 0.6 of overall equivalence ratio, since mixing in the second stage was so rapid. It was confirmed that this type of injector was suitable for two-staged RQL combustion.

Key words: RQL Combustor, Hydrogen Fuel, Two-Staged Combustion, Premixed Combustion, Gas Turbine, Burner, Coaxial Injector, NOx Emission, Combustion Efficiency, Flame Stability

1. Introduction
A distributed electrical power generation system of micro gas turbines with hydrogen fuel makes effective use of energy resources with no CO2 emissions, so it may contribute to prevention of global warming. To achieve such a micro gas turbine for hydrogen, we have been studying hydrogen combustors for micro gas turbines of several kW output. So far, we developed 2 types of experimental combustors, based on the excellent combustion characteristics of hydrogen. One used a swirling diffusion flame, and another a swirling lean premixed flame. The diffusion flame type combustor had an extremely high space
heating rate (maximum 3.5 × 10³ MW/(m³ ·MPa)) and combustion efficiency (99.5% or higher). But the emission concentration of NOx was relatively high, about 15 ppm. The lean premixed flame type combustor(2)(3) achieved extremely low NOx emissions of 1 ppm or less, while having the same level of high space heating rate obtained with the diffusion flame type. However, combustion driven oscillation occurred for the premixed type in the equivalence ratio of about 0.4 or higher, so this problem limited aiming at even higher thermal efficiency. In recent years, research on advanced gas turbines (target turbine inlet temperature: T_in=1700°C) have been conducted(4), using heat resistant oxide composite materials called MGC (Melt-Growth Composite) in the hot parts, for even higher thermal efficiency. In the near future, when micro gas turbines will become able to use hot parts of MGC, it is necessary to develop an appropriate combustion technology, which can achieve stable combustion and low NOx emissions at high temperature, that is, high equivalence ratio conditions.

The authors focused on a two-staged combustion method, with the expectation of achieving the two requirements described above at high temperature, for hydrogen combustors of micro gas turbines. In the combustion method, a fuel rich premixed gas burns in the primary zone, then the temperature of the burned gas is rapidly reduced by quickly supplying and mixing secondary air. This quenches the NOx generation reactions, and achieves lean combustion in the secondary zone. This method is called RQL (Rich-Quench-Lean) combustion, which differs from traditional low NOx combustion methods, such as the lean combustion method by the staging of fuel distribution, or the lean premixed combustion method with pilot flames, etc.(5)(6). Several research projects have been conducted on RQL combustors(7)(8), but the performances obtained were insufficient. Also, as far as we know, there is no report of a hydrogen combustor or a combustor for micro gas turbines, which uses RQL combustion. Exploiting the benefits of excellent hydrogen combustion characteristics, that is, high burning rate and no generation of prompt NO in a fuel rich condition, it is possible to develop an RQL combustor with extremely low NOx emission and high space heating rate.

Our previous studies revealed the optimum equivalence ratio for primary combustion (below, primary equivalence ratio) by numeric calculations, and we made an experimental combustor with an injector issuing swirling rich premixed gases to examine its combustion characteristics(9). As a result, it became clear from the calculations that a primary equivalence ratio of 2 or higher with rapid-stirred combustion in the secondary zone was effective for NOx reduction. However, the experiment of the combustor showed that good flame stability was obtained at a primary equivalence ratio of 2, but rapid mixing between the high temperature gas formed in the recirculation zone and the secondary air coaxially supplied thorough the injector was not achieved, so NOx emission concentration was not sufficiently reduced. To resolve this, it was necessary to prevent development of a large recirculation zone in the combustor. The authors also found that in a coaxial air flow, a hydrogen–air rich premixed flame which anchored at the rim of a tube burner was extremely stable(10). With this flame stabilizing method, a large recirculation zone was not formed, so it is possible to reduce NOx if secondary combustion progresses rapidly.

This research aimed to build an RQL combustor by this flame stabilizing method. In this paper, a test combustor with a coaxial type single injector was made, which injects rich premixed gas from the central tube and secondary air from its surroundings. NOx concentrations and combustion efficiencies, temperature distributions, pressure losses, and flame stability limits were measured, varying the jet hole area, the jet angle of secondary air (below, coaxial air), and the mass flow rate.

Nomenclature

\[ AR : \text{Area ratio (Coaxial air hole area / premixed gas hole area)} \]
2. Experimental Apparatus and Method

2-1 Single injector test combustor

Increase of $\phi_t$ leads to a relatively small flow rate of secondary air compared to premixed gas, when the primary equivalence ratio $\phi_i$ is constant. Thus an efficient method of supplying secondary air is important to achieve rapid secondary combustion at high $\phi_t$. Considering this, instead of a combustor in which a large flame is formed, one with multiple injectors that distribute secondary air directly to each small rich premixed flame is effective. This type also has an advantage of flexibility of flow rate and combustor shape, for designing a gas turbine combustor to a required specification. To develop this kind of combustor, it is necessary to realize ideal RQL combustion for one flame.

Figure 1 shows the single coaxial injector made based on this concept, and the test combustor that installed the injector. The injector was set in the center of the combustor inlet. Rich premixed gas from the central tube and secondary air from the coaxial annular hole were injected, respectively. The diameter of the premixed gas jet hole was 4 mm. The area of coaxial air hole and jet angle $\theta$ were variable. This $\theta$ was designed with added impinging effects on the coaxial jet, so it promoted generation and development of extremely strong turbulence upstream of the jet, aiming at increase of mixing effect downstream. Table 1 shows details of the injectors used. There were 3 types of area ratio $AR$ (0.62, 1.05 and 1.9), with 3 angles $\theta$ for each one (0°, 6° and 12°), for a total of 9 types. The injector names were described as “$AR - \theta$”, with $AR$ from small to large S, M, L (example: S-12). Also, when this paper indicated the injectors with the same $AR$ and different $\theta$ that were grouped together, they were shown as, for example, “injectors S”. To minimize heat loss to the combustor wall when measuring temperature in the combustor, and NOx and hydrogen concentrations in combustion gas, a combustion tube made of high insulating alumina fiber (heat conductivity at 400°C: 0.2 W/(m·K)) was used. A combustion tube made of quartz glass was used when observing flame shapes and measuring flame stability limits.
2-2 Method of experiment and measurement

We varied \( \phi_t \) while setting the target total air flow rate to \( \dot{m}_{tot} = 0.68 \text{ g/s} \) and varying hydrogen flow rates. The determination of this \( \dot{m}_{tot} \) value was presupposed to install the injectors in a combustor of the same size as the combustors in our previous researches \(^{(1)-(3),(9)}\).

A water-cooled probe with outer diameter of 3 mm, and gas sampling tube inner diameter of 0.7 mm, was used to collect combustion gases at the exit of the combustor. After removing moisture, the gases were analyzed. NOx concentration was measured by a chemiluminescence analyzer, and hydrogen by an analyzer that used the hotwire-type semiconductor and contact combustion method.

In the experiment, \( \phi_t \) was varied over a wide range. Particularly for high values of \( \phi_t \), the mole fraction of H\(_2\)O in combustion gas increased, resulting in relatively high measurements of NOx or hydrogen concentration. So these measured concentrations could not be compared directly, since the concentrations depended on \( \phi_t \). For example, as \( \phi_t \) became higher, the combustion efficiency was estimated to be lower. To obtain actual concentrations in the combustion gas as correctly as possible, we converted actual measured concentrations of NOx and hydrogen into concentrations in moist combustion gas with the following simple method. The combustion reaction assumed the following overall reaction equation and the mole fractions of H\(_2\)O, O\(_2\) and N\(_2\) after combustion were used. Measurements showed minute quantities of actual concentrations of NOx and hydrogen, so these mole fractions were ignored in the conversion.

\[
\begin{align*}
\text{H}_2 + (\phi SR)^{-1} \cdot (0.21\text{O}_2 + 0.79\text{N}_2) \\
\rightarrow \text{H}_2\text{O} + (\phi SR)^{-1} \cdot (0.21 - 0.5)\text{O}_2 + (\phi SR)^{-1} \cdot 0.79\text{N}_2 \quad \text{(R1)}
\end{align*}
\]

\( \phi_t \): equivalence ratio, \( SR \): stoichiometric mixture ratio of hydrogen–air reaction)

For NOx emission concentrations of gas turbines, a 15% O\(_2\)-conversion with O\(_2\) concentrations remaining in combustion gas (in Japan, 16% O\(_2\)-conversion) is generally used, but the aim of this paper is to compare NOx concentrations at different \( \phi_t \), so the NOx emissions were not converted.

At representative combustion conditions, direct and schlieren photographs of the flames were taken, and the temperature distributions in the combustor were measured. For the temperature measurement, an R-type thermocouple with wire diameter of 0.05 mm and the surface coated with SiO\(_2\) was used. The temperatures of premixed gases and coaxial airs flowing into the combustor were both at room temperatures, and all experiments were performed under atmospheric pressure.

3. Experimental Results and Discussion

3-1 Flame configuration

Figure 2 shows the direct photographs of the flames at \( \phi_t = 2 \), and its schlieren photographs near the injector exit. The schlieren photographs were taken without the combustion tube. Each flame consisted of an inner flame and an outer flame, that is, a “Bunsen flame” was formed. But the outer flames were extremely turbulent in all conditions. The shape and length of the inner flames were independent from the coaxial air conditions at the same \( \phi_t \). The outer flame length became shorter with decreasing \( AR \). It is probable that \( u_0 \) has large effects on mixing rate in the secondary combustion zone. Regarding the effects of \( \theta \), the flame width was a little narrow at \( \theta = 12^\circ \) compared to at \( 0^\circ \), but the lengths of these outer flames and schlieren images were almost the same. This suggested that the degrees of \( \theta \) tested in the experiment were not effective for rapid mixing in the secondary combustion zone. At conditions \( \phi_t = 0.7 \) and \( AR = 1.9 \), the flame reached
outside the combustion tube, so this suggested that the reaction was not complete in the combustor.

3-2 NOx concentration

3-2-1 Effect of the primary equivalence ratio   Figure 3 shows the variation of NOx concentration with $\phi_i$, investigated with the injector M-0 under the condition of $\phi_t = 0.5$. In the experiments, the distributions of NOx concentrations in the combustor exit were almost uniform in radial directions, so the concentrations were represented by values at the central point of the exit. As $\phi_i$ increased, NOx concentration decreased and almost became fixed for $\phi_i \geq 2$. The tendency corresponded to the results from the numeric calculations\cite{9}, so it is possible to realize RQL combustion with this type of injector. For the same $\phi_i$, higher $m_{at}$ resulted in lower NOx concentration. The reason may be that, accompanying increasing $m_{at}$, the residence time of the combustion gas in the high temperature zone decreases, in addition to effects of stronger turbulence.

At representative conditions of $m_{at} = 0.68$ g/s and $\phi_i = 2$, the measured NOx concentration was about 6 ppm, but a 15% O$_2$-conversion gave a value of about 4 ppm, sufficiently satisfying the emission regulations\cite{11}.

Figure 3 also shows PL with combustion at $m_{at} = 0.68$ g/s. PL increased along with $\phi_i$. This was because accompanying an increase in $\phi_i$, the coaxial air flow rate increased, so the total pressure loss at the coaxial air hole became dominant. NOx concentration was sufficiently low at $\phi_i \geq 2$. Higher $\phi_i$ only invited excess PL. The condition $\phi_i = 2$ was also appropriate from this viewpoint.

\begin{table}[h]
\centering
\begin{tabular}{|c|c|c|c|c|}
\hline
$\phi_t$ & 0.3 & 0.5 & 0.7 \\
\hline
Injector & S-0 & S-0 & M-0 & M-12 & L-0 \\
\hline
\end{tabular}
\caption{Table of Typical flame configurations and its schlieren images ($\phi_i = 2$, $m_{at} = 0.68$ g/s)}
\end{table}

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure3.png}
\caption{NOx concentrations versus $\phi_i$ (Injector: M-0)}
\end{figure}
3-2-2 Effects of area ratio and secondary air jet angle

Figure 4 shows variations in NOx concentration versus $\phi_t$, with $AR$ and $\theta$ as parameters. Increased $\phi_t$ was accompanied by steady increases in NOx with all injectors. With the same $\phi_t$, NOx concentration decreased as $AR$ decreased. $\theta$ had small effects under all conditions. In the case of the injectors L, when $\phi_t$ is 0.7 or higher, the rate of increase of NOx concentration became slow versus $\phi_t$. Under these conditions, it seems to be incomplete combustion due to insufficient progress of secondary combustion, as shown in Fig.2.

Based on the calculation results\(^{(12)}\), the NOx concentration generated by the primary combustion was extremely low under $\phi_i = 2$. Therefore, this suggested that the NOx concentration at the combustor exit mostly depended on the mixing rate of the primary combustion gas with the coaxial air in the secondary combustion zone. Figure 5 shows the re-plotted data of the NOx concentrations shown in Fig.4 versus the ratio $M_a / M_j$, where $M_a / M_j$ is the ratio of [momentum in the axis direction of the coaxial air jet ($M_a = \dot{m}_a u_a$; $\dot{m}_a$ is mass flow rate of coaxial air)] to [momentum in the axis direction of premixed gas jet ($M_j = \dot{m}_p u_j$; $\dot{m}_p$ is mass flow rate of premixed gas)]. The effect of $\theta$ on $M_a$ was not considered here. This figure showed that NOx concentrations measured in this experiment could be arranged versus the ratio of the momentum defined by the condition at injector exit without combustion, for all injectors with different $AR$.

Seeking this relationship equation with the method of least squares gave $C_{nox} = 24 \cdot (M_a / M_j)^{0.75}$. These results suggest that the stream lines of the primary combustion gas passing through the rich premixed flame is towards outside, so even if $\theta = 0^\circ$, the primary combustion gas mixes with the coaxial air as impinging jets\(^{(13)}\).

3-3 Hydrogen concentration and combustion efficiency

Even under the condition of the injector L-0 and $\phi_t = 0.7$, which was predicted to be most difficult for rapid mixing in the secondary combustion zone, the hydrogen concentration distribution at the combustor exit in radial directions was a little high in the center, but roughly uniform. The volume flow rate distributions at the exit approximated
from temperature distribution and total pressure distribution were almost uniform for all injectors, so the hydrogen concentration was evaluated at the maximum value at the center of the combustor exit. Figure 6 shows variation of the hydrogen concentration with $\phi_t$. These are shown together with the calculated equilibrium concentration of $\text{H}_2$ species. The hydrogen concentrations measured were minimum when $\phi_t$ was in the range 0.3 to 0.4, although the $\phi_t$ for which it indicated lowest concentrations differed somewhat depending on $AR$. When $\phi_t$ was lower or higher than this range, the hydrogen concentration increased. In the high $\phi_t$ range, the smaller the $AR$, the lower the hydrogen concentration. In contrast, in the low $\phi_t$ range, the larger the $AR$, the lower the hydrogen concentration. Effects of $\theta$ could not be seen clearly.

In the equilibrium state, $\text{H}_2$ concentration generated through thermal dissociation increases with increasing $\phi_t$ due to temperature rise. When $\phi_t$ was high, the hydrogen concentrations measured for the injectors S and M were approximately the same as the equilibrium concentrations, so the secondary combustion was probably almost complete in the combustor. For the injectors L, the measured hydrogen concentrations were quite high compared to the equilibrium concentrations. In this situation, the flame temperature became high due to forming a diffusion flame, and unburned hydrogen was exhausted because the mixing of primary combustion gas with the coaxial air progressed slowly in the secondary combustion zone. On the other hand, when $\phi_t$ was low, defining the velocity ratio as $u_a / u_j$, for example if $\phi_t = 0.2$, the velocity ratios for the injectors L, M, and S were 2.5, 4.5, and 7.8, rising along with decreasing $AR$. In these conditions, it is possible that local quenching occurs by flame stretch due to large velocity gradients at the boundary between the coaxial air and the flame front, resulting in detection of much unburned hydrogen.

Figure 7 shows the combustion efficiencies approximated from the unburned hydrogen concentrations shown in Fig. 6, assuming uniform hydrogen concentration and velocity distributions at the combustor exit. Corresponding to the measured hydrogen concentrations, for high $\phi_t$, the higher the $AR$, the higher the combustion efficiency, and for low $\phi_t$, the lower the $AR$, the lower the combustion efficiency. Under practical $\phi_t$ conditions as $0.3 \leq \phi_t \leq 0.6$, the combustion efficiency was sufficiently high for all injectors, at 99.3% or higher. Maximum values were used for hydrogen concentrations, so actual combustion efficiency becomes even higher than this. Among these, the combustion efficiency of the

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**Figure 6** H$_2$ concentrations at the exit of the test combustors versus $\phi_t$, ($\phi_i = 2.0$, $m_o = 0.68\text{g/s}$)

**Figure 7** Combustion efficiencies versus $\phi_t$,

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**Table:**

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injector S-0 was highest at 99.95% or higher, and, in addition, the NOx concentration previously mentioned was lowest, confirming that it is the most excellent injector from the aspect of emission characteristics.

3-4 Temperature distribution in the combustor

Figure 8 shows temperature distributions in the combustor for the injectors S-0 and S-12, when $\phi_t = 0.5$, measured at height $h \geq 30$ mm downstream from the injector exit. The temperatures further upstream could not be measured, because the thermocouple coating melted due to high temperatures. Corrections for heat loss from the thermocouple were not made.

The temperatures in the combustor were highest along the center axis, decreasing in radial direction and downstream direction, showing axial symmetric distribution. Compared to the injector S-0, the temperature contour was somewhat long and thin for the injector S-12, but large differences were not observed between the temperature fields of these injectors. This indicated that there were almost no substantial effects of $\theta$ in the mixing. Volume of the high temperature region was approximately the same for both, which supported the result that differences could not be recognized in the NOx concentrations.

For this coaxial type injector, as with typical free jets, progress of temperature uniformalization in the secondary combustion zone was slowest on the center axis. Therefore, we can roughly estimate the size of the high temperature zone in the combustor, in other words, the state of progress of mixing of the primary combustion gas with the coaxial air only from the temperature distribution on the center axis. Figure 9 shows the temperature distributions on the center axis of the combustor for all injectors. Upper Fig.9 showed that for the same $\phi_t$, smaller $AR$ led to a smaller high temperature region formed in the combustor. In lower Fig.9, when $\phi_t$ was varied for the injector M-0, larger $\phi_t$ led to a larger high temperature region in the combustor.

Considering that NOx generation from hydrogen-air combustion originates from only thermal NO,
differences in the NOx concentrations due to $AR$ shown in Fig.4 can be explained by differences in sizes of the high temperature region in the combustor. The shapes of the high temperature region formed in the combustor were somewhat different for the injectors S-0 and S-12, but for the injectors M and L, the center axis temperature distributions were not affected by $\theta$. Regarding their $AR$, there was almost no effect of $\theta$ on mixing in the secondary combustion zone.

3-5 Total pressure loss ratio

Figure 10 shows the results of $PL$ measured with combustion for the injectors S, which were the most excellent injectors in the present experiment for the points of low NOx and high combustion efficiency. In the present experiment, separate high-pressure air sources were used for the primary air and for the coaxial air. Thus different total pressure loss occurred at the premixed gas hole and the coaxial air hole, but the total pressure loss ratio at the premixed gas hole was under 1% for all $\phi_t$, a level which could be ignored. Therefore, the illustrated $PL$ are the values at the coaxial air hole.

Upper Fig.10 shows that when $\dot{m}_{at}$ was fixed at 0.68 g/s, an increase in $\phi_t$ was accompanied by a decrease in $PL$. This is because when $\phi_t$ is fixed at 2.0, even if $\dot{m}_{at}$ is fixed, the air flow rates distributed for the premixed air and the coaxial air differs by $\phi_t$. Differences due to $\theta$ were almost unrecognized. Lower Fig.10 shows that for the injector S-0, when $\dot{m}_{at}$ was increased, $PL$ appeared to increase by the square of $\dot{m}_{at}$. For the $\dot{m}_{at} = 0.68$ g/s set in this experiment, $\phi_t = 0.6$ had about 3% $PL$, and $\phi_t = 0.4$ about 4%. These values are acceptable for practical use.

3-6 Flame stability limits

Figure 11 shows the flame stability limits for the injectors S. These measurements were obtained by gradually increasing $u_a$ after forming a stable rich premixed flame with $\phi_t = 2$ and predetermined $u_j$.

For all values of $\theta$, when $u_a$ increased, the stable flames
transitioned to high frequency combustion driven oscillation. The $u_a$ for the transition limit steadily increased along with $u_j$, but for $\theta = 0^\circ$ and $u_j \geq 40$ m/s, it almost became fixed. For $\theta = 6^\circ$ or $12^\circ$, when $u_j$ was greater than 30 m/s, the flame blow-off occurred. A tendency was seen for a low velocity range that the higher the $\theta$, the smaller the $u_a$ that transitioned to combustion driven oscillation. But it was not clear that $\theta$ affected the blow-off limit of the high velocity range. In sum, the injector with $\theta = 0^\circ$ had the widest range of flame stability.

During combustion driven oscillation, the flame front near the flame base was indistinct and fluttering strongly, so it is supposed that the flame base anchored in the recirculation zone behind the injector inner pipe rim becomes unstable, leading to a transition to high frequency combustion driven oscillation. Further investigation of the cause is a future issue.

For the injectors S, even with any $\theta$, that stable combustion zone sufficiently widely covered the flow rate established in the present experiment ($\phi_i \geq 0.4, \dot{m}_{at} = 0.68$ g/s). This type of injector was confirmed to be extremely excellent for flame stability.

## 4. Conclusions

Based on the two-staged combustion method (Rich-Quench-Lean combustion) concept of rapidly cooling primary combustion gas with secondary air, achieving quenching of the NOx generating reactions and complete lean-combustion of surplus fuel at the same time, we proposed a coaxial injector which injects secondary air with high velocity from its surroundings on the central hydrogen-air rich premixed flame. Combustion experiments were performed using a test combustor with this injector, and the following conclusions were obtained.

- When the primary equivalence ratio $\phi_i$ was 2 or greater, the NOx concentration became sufficiently low after two-staged combustion. Along with an increase in the overall equivalence ratio $\phi_t$ of the combustor, the NOx concentration increased. But for the same $\phi_i$, the higher the coaxial air velocity $u_a$, the faster mixing occurred in the secondary combustion zone, the high temperature region behind the rich premixed flame became smaller, and the NOx concentration decreased.
- For a fixed $\phi_i$, NOx concentration at the test combustor exit was controlled by the momentum ratio for the condition at the injector exit without combustion, defined by $[\text{momentum of coaxial air jet}] / [\text{momentum of premixed gas jet}]$, and the concentration was in proportion to the -0.75 power of the ratio.
- For all injectors, in the range of $0.3 \leq \phi_t \leq 0.6$, the combustion efficiency obtained was sufficiently high at 99.3% or higher. The combustion efficiency decreased when $\phi_i$ was lower than this, because the flame partially quenched due to large velocity gradient at the flame front. In contrast when $\phi_i$ was high, $u_a$ was small and the secondary combustion was incomplete in the combustor, reducing the combustion efficiency.
- In the range for the present experiment, the jet angle $\theta$ of the coaxial air had small effects on the combustion field in the combustor.
- In the present experiment, the injector S-0 was the most excellent for flame stability and emission performance. The test combustor with this injector achieved the NOx emission concentration of about 7 ppm, the combustion efficiency of 99.95%, and the total pressure loss ratio of about 3%, at the high equivalency ratio of $\phi_t = 0.6$.

## References


