# Development of a Low $NO_X$ Two-Stage Tubular Flame Burner with Intercooling under Near Stoichiometric Conditions

Y. SHIRAGA[*, 1], T. HORI[2], J. HAYASHI[3], A. KEGASA[4], F. AKAMATSU[5],

[1] Osaka Gas Co., Ltd., 6-19-9 Torishima, Konohana-ku, Osaka, 554-0051, Japan
[2, 3, 4, 5] Osaka University, 2-1 Yamadaoka, Suita-shi, Osaka, 565-0871, Japan

**Abstract**

Lean burn and exhaust gas recirculation are effective to reduce nitrogen oxides ($NO_X$). However, these methods may lead to increasing exhaust loss and deteriorating heating performance. The purpose of this research is to develop a $NO_X$ reduction method under near stoichiometric conditions by improving so-called two-stage combustion. Tubular premixed flame is used in the primary stage. The flame stabilization of tubular flame is significantly high, so that richer flame is formed than that with conventional burners. $NO_X$ concentration in the exhaust gas was measured at different equivalence ratios of the primary stage and the amount of cooling duty under the combustion rate of 1 kW. The results show that $NO_X$ concentration could be reduced down to approximately 10 ppm.

## Introduction

A variety of techniques for reducing nitrogen oxides ($NO_X$) have been researched and developed; these include lean burn, exhaust gas recirculation, two-stage combustion, catalytic combustion and applications of Smithells flame [1-7]. Among these, lean burn and exhaust gas recirculation are effective to reduce $NO_X$ because of low flame temperature. However, these combustion methods may lead to an increasing exhaust loss or deteriorating heating performance. Furthermore, these methods are hard to apply to a high-temperature heating furnace with oxygen-enriched combustion. A low $NO_X$ burner under stoichiometric conditions has been desired to overcome these problems. Two-stage combustion is a low $NO_X$ technology under stoichiometric conditions without lean burn and exhaust gas recirculation. In two-stage combustion, rich flame is formed in the primary stage. In the secondary stage, lean flame is formed to re-burn the incompletely burned gas of the primary stage with secondary air. Furthermore, burned gas is cooled after the primary stage to lower the flame temperature of the secondary stage. Therefore, thermal $NO_X$ formation can be suppressed even under stoichiometric conditions.

In this study, a two-stage burner is developed. Tubular premixed flame [8] and swirling flame are used in the primary and secondary stages, respectively. The flame stabilization of tubular flame is significantly high, so that richer flame is formed than that with conventional burners, such as Bunsen and swirl-type burners. The $NO_X$ concentration in the exhaust gas was measured at different equivalence ratios of the primary stage and the amount of cooling duty under the combustion rate of 1 kW (HHV: Higher Heating Value).

## Experimental Set-up and Procedure

Figure 1 shows a schematic diagram of our experimental apparatus. Town gas (typical composition: methane 88.9%, ethane 6.8%, propane 3.1%, butane 1.2%) used as fuel was supplied from a cylinder to the primary burner, and air was supplied from a compressor to both the primary and secondary burners, via a mass-flow controller (Azbil MQV). The fuel supply rate was fixed at $2.3 \times 10^{-5}$ $m_N^3/s$ (1.05 $kW_{HHV}$). The air supply rate was adjusted to ensure that the equivalence ratio in the primary burner satisfied the settled conditions, while the overall equivalence ratio remained constant at 0.91.

The inner diameter of the primary tubular flame burner was 20 mm, while the inner diameter of the secondary non-premixed flame burner was 30 mm. For both burners, a structure in which premixed gas or air was injected into the burner from the direction tangential to the pipe was adopted. In experiments designed to mimic traditional two-stage combustion by forming a partially premixed flame in the primary burner, a pipe with an inner diameter of 5.6 mm was attached to the upstream edge of the primary burner, and premixed gas (equivalence ratio of 3.3) was injected into the burner.

The downstream components of both the primary and secondary burners were equipped with quartz glass combustion tubes to allow observation of the flame. The external appearance of the flame was photographed using a digital camera (Canon PowerShot G10; settings: shutter speed 1/4″, aperture F4.0, ISO1600).

To estimate the rate of intermediate cooling of the burned gas between the primary and secondary burners, a water-cooling pipe (304 steel, length 100 mm) was installed. The flow rate of the water supply was measured (flow meter: OVAL LSF41C) together with the temperatures of the water at the inlet and outlet; this allowed determination of the heat received by the water and thus the rate of intermediate cooling of the burned gas. To investigate the effect of intermediate cooling, experiments in which the length of the glass tube downstream from the primary burner (i.e., the distance between burners) was varied. The temperature of the primary burned gas flowing into the secondary burner was measured using an R-type (Pt-Pt 13% Rh) thermocouple with a strand diameter of 100 μm

[*] Corresponding author: y-shiraga@osakagas.co.jp

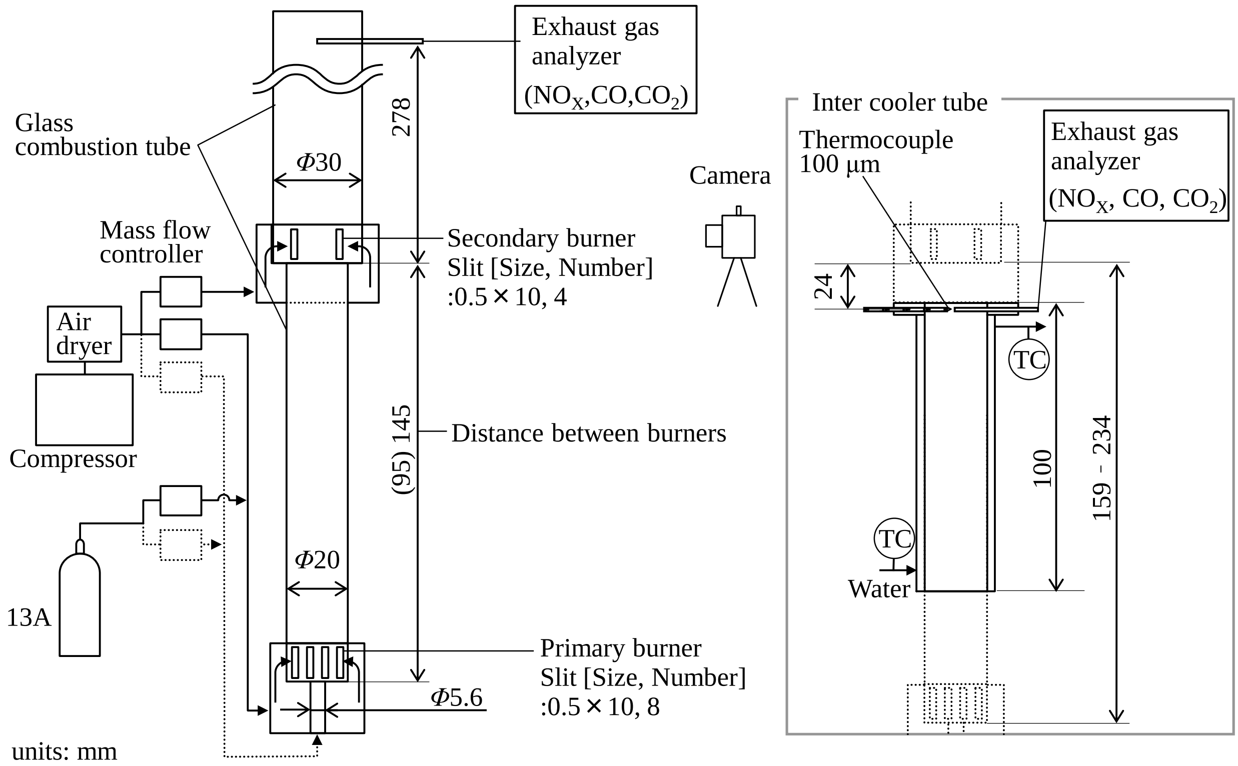


Fig. 1 Experimental apparatus.

positioned above the pipe axis at a point 24 mm upstream from the upstream edge of the secondary burner.

$NO_X$ and CO concentrations were measured using a HORIBA PG-340 infrared analyzer. Exhaust gas was sampled along the pipe axis at a point 278 mm downstream from the upstream edge of the secondary burner and at the same point of the thermocouple. The $NO_X$ and CO concentrations of exhaust gas were converted from $CO_2$ measured data to the 0% $O_2$ concentration. To convert the measured $NO_X$ concentration into a mass flow rate, the molar flow rate of burned gas was estimated via equilibrium calculation from the supply gas rate, regarding $NO_X$ as NO.

**Results and Discussion**

The effect of equivalence ratio in the primary flame (tubular flame) on the $NO_X$ concentration in the exhaust gas was investigated. Figure 2 shows the flames at various primary equivalence ratios, while Figure 3 plots the corresponding $NO_X$ and CO concentrations in the exhaust gas. The distance between the primary and secondary burners was set to 145 mm. The primary flame was formed at a primary equivalence ratio below approximately 1.7. The $NO_X$ concentration in the exhaust gas decreased as the equivalence ratio of the primary flame increased. The $NO_X$ concentrations for primary equivalence ratios of 1.34 and 1.44 were nearly equal; this may be due to incomplete combustion in the secondary burner for the primary equivalence ratio of 1.34. At a primary equivalence ratio of 1.34, the CO concentration was higher than that for equivalence ratios of 1.4 and above. This may be attributed to a release of heat from the primary burned gas, one of the causes of incomplete combustion.

When considering the $NO_X$ concentration in the exhaust gas from the two-stage combustion, the release of heat from the primary burned gas (i.e., the degree of intermediate cooling) cannot be neglected. For this reason, a glass combustion tube downstream from the primary burner was replaced with a chilled-water tube for intermediate cooling. The amount of heat received by the cooling water was regarded as the degree of intermediate cooling. In this case, the distance between the primary burner and secondary burner was set to 159 mm.

In Figure 4, the heat received by the water, burned gas temperature immediately before the burner, CO concentration in the exhaust gas, NO volume immediately before the secondary burner (the primary NO) and NO volume near the burner exit (NO emission) are plotted against the equivalence ratio in the primary burner. To compare the primary NO and NO emission, the amount of NO should be described not by concentration but by mass per heat input. Between primary equivalence-ratio values of approximately 1.5 to 1.7, the amount of heat received by the cooling water decreased as the equivalence ratio increased. For the tubular flame, the outer side of the flame was covered by unburned mixed gas; it was difficult for the heat from the burned gas to propagate to the inner tube walls at points in the axial direction at which flame was

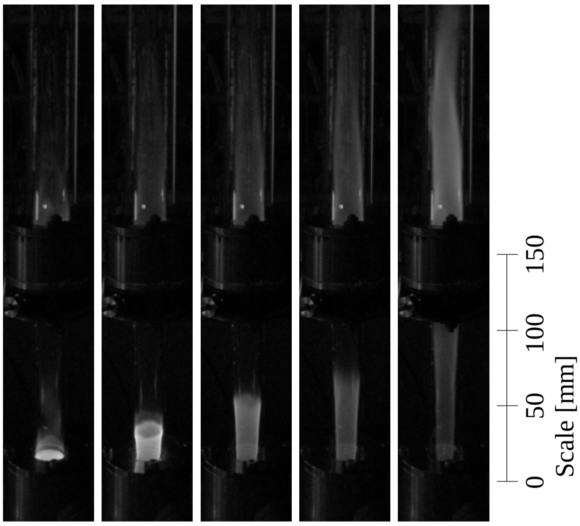


Fig. 2 Direct photos of two-stage combustion using tubular flame at each equivalence ratio in primary burner.
($1.05\ kW_{HHV}$, $\Phi_{Total}=0.91$, $L_{1st\ burner}=145$ mm, without cooling tube)

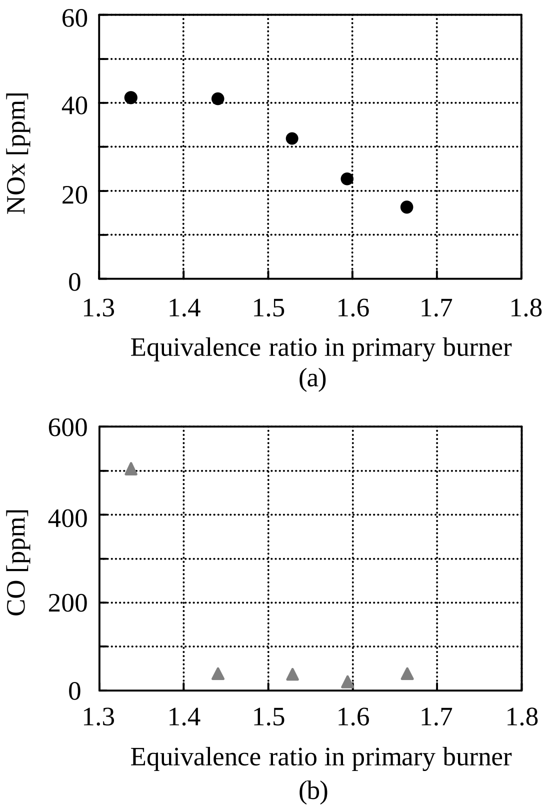


Fig. 3 Influence of equivalence ratio in primary burner without intercooling tube.
($1.05\ kW_{HHV}$, $\Phi_{Total}=0.91$, $L_{1st\ burner}=145$ mm)

present. Heat transfer begins only downstream from the flame, when the burned gas begins to make contact with the walls [9]. As the equivalence ratio increases, the flame length and the heat-transfer area decrease. In addition, the flame temperature is lower for higher equivalence ratios, which tends to reduce the degree of heat transfer to the walls. It is concluded from these observations that intermediate cooling may be reduced as the primary equivalence ratio increases. For values of the primary equivalence ratio below 1.5, the heat received by the cooling water did not change. In these cases, the primary flame did not extend to the chilled-water pipe, and thus its heat is lost to somewhere other than the supply water.

The higher the primary equivalence ratio was set, the higher the gas temperature was achieved just upstream of the secondary burner because of the reduction in intermediate cooling. The fact that the temperature was lower at an equivalence ratio of 1.7 than at the equivalence ratio of 1.67 demonstrates the effect of the reduction in flame temperature as the equivalence ratio increases and the primary flame extends to the thermocouple.

The CO concentration was high when the primary equivalence ratio was below approximately 1.4. This was attributed to significant intermediate cooling, causing incomplete combustion.

The primary NO decreased as the primary equivalence ratio increased, falling to roughly zero for primary equivalence ratios above roughly 1.55. The decrease in primary NO is accompanied by a decrease in NO emission. As noted above, intermediate cooling is reduced at high primary equivalence ratios. Given the specifications of this burner, the effect of primary NO restrictions is inferred to be more significant than the effect of secondary NO restrictions due to intermediate cooling, thus decreasing the NO emission. The rapid increase in NO emission at a primary equivalence ratio of 1.7 may be due to the primary flame reaching the secondary burner. From the point of view of the difference of NO, it is found that the lower the primary equivalence ratio, the smaller the difference between the primary and secondary amount under the conditions of the equivalence ratio in the primary burner of 1.55 or lower. At a primary equivalence ratio of 1.35, there is almost no difference between the primary NO and the NO emission; this is because at this point there is almost no production of NO in the secondary burner. In contrast, for primary equivalence ratios of roughly 1.55 and above, the majority of the NO is produced in the secondary burner. In other words, by restricting NO generation in the secondary burner, even greater reduction in NO emission is expected.

For this reason, intermediate cooling was subsequently increased to attempt further NO reductions. The primary equivalence ratio was fixed at 1.67 and the effect of intermediate cooling on NO emission was investigated. By varying the length of the glass tube downstream from the primary burner (i.e., by varying the distance between the burners), the degree of

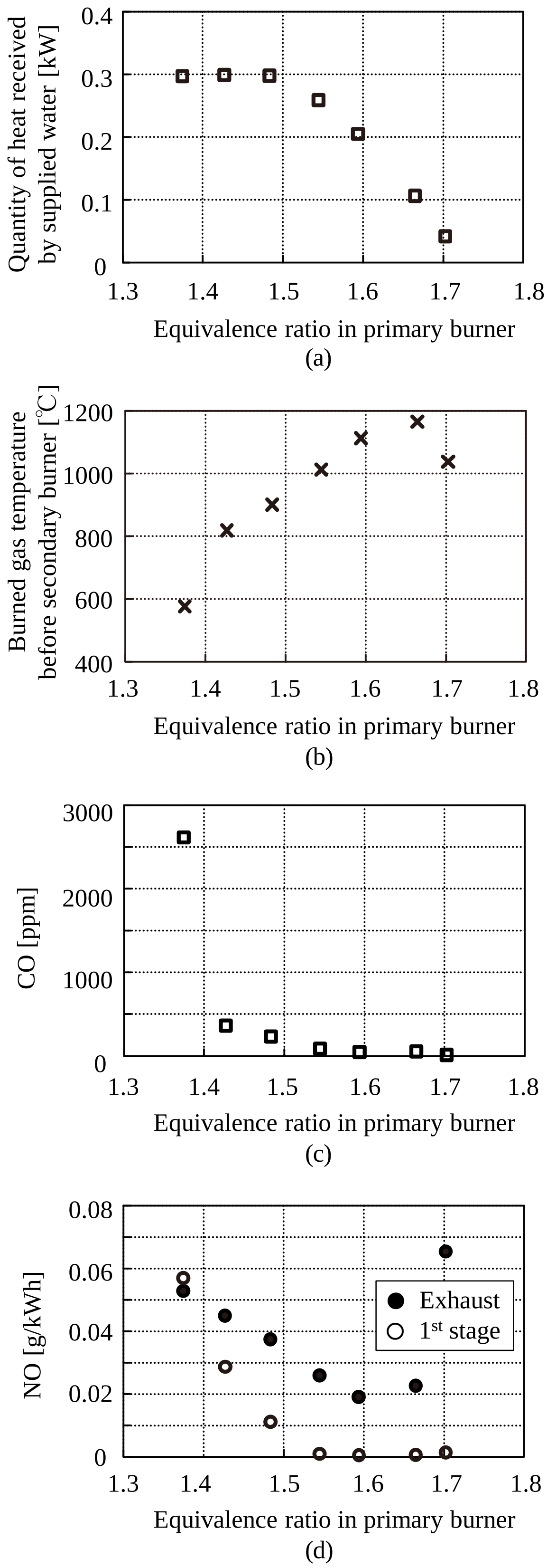


Fig. 4 Influence of equivalence ratio in primary burner with intercooling tube.
($1.05\ kW_{HHV}$, $\Phi_{Total}=0.91$, $L_{1st\ burner}=159$ mm)

intermediate cooling (the quantity of heat received by the supply water) was varied. Figure 5 shows the heat received by the cooling water, gas temperature just upstream of the secondary burner versus the distance between burners and NO versus the heat received by the cooling water. As the distance between burners increased, the heat received by the cooling water increased and the gas temperature just upstream of the secondary burner decreased.

The primary NO was nearly zero at all conditions; this is because the primary equivalence ratio was large, at 1.55 or higher. The volume of secondary NO emission decreases as the heat received by the cooling

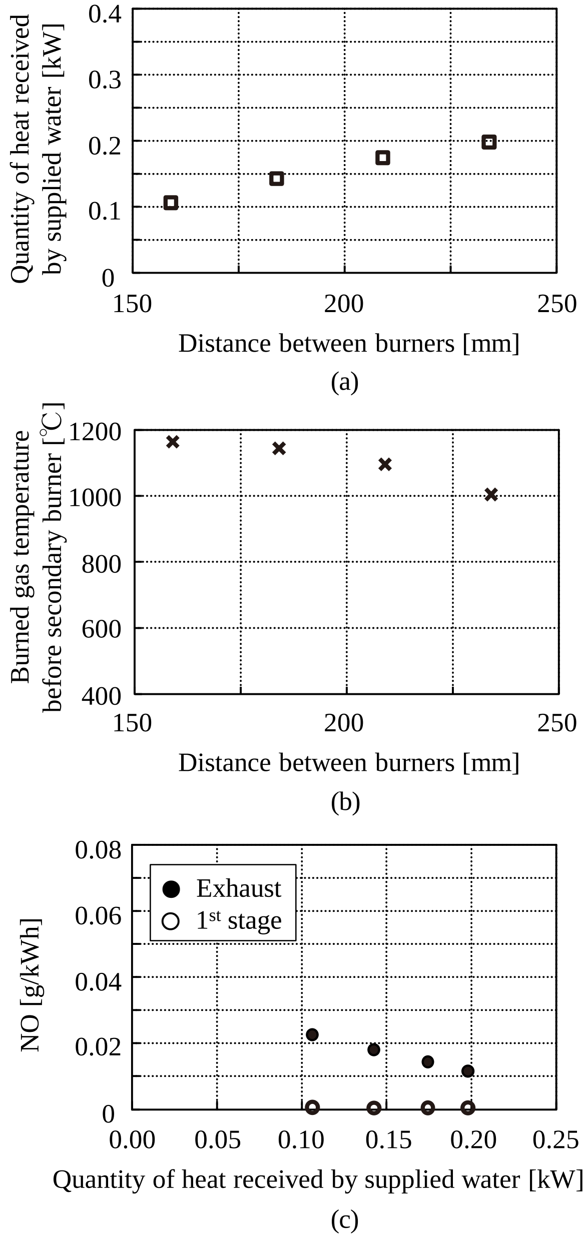


Fig. 5 Influence of intercooling.
($1.05\ kW_{HHV}$, $\Phi_{Total}=0.91$, $\Phi_{1st}=1.67$)

water increases. This is attributed to reduced secondary NO production due to increased intermediate cooling. With an intermediate cooling rate of 0.2 kW (roughly 20% of the input heat), the volume of NO emission decreased to 0.012 g/kWh (10.2 ppm by concentration).

Finally, the $NO_X$ of this method was compared to those of conventional combustion methods. As conventional combustion methods, single-stage partially premixed combustion and two-stage combustion in which the first stage uses partially premixed flame were selected. The same experimental apparatus was used and experiments were conducted with the overall supply equivalence ratio fixed at 0.91. Figure 6 shows the flame, while Figure 7 shows their $NO_X$ concentrations. In the case of single-stage partially premixed combustion, the equivalence ratio for the primary burner was set to 1.72 to prevent the formation of flame in the primary burner (tubular flame). As shown in Fig. 6(a), the flame formed only in the secondary burner. The $NO_X$ concentration for single-stage premixed combustion was 96.2 ppm. For two-stage combustion in which the first stage used partially premixed flame, all of the fuel and primary air was premixed (equivalence ratio 3.3) and injected into the burner in the axial direction from the upstream edge of the primary burner. In addition, air was also supplied in the direction tangential to the burner; the supply equivalence ratio of the primary burner was set to 1.67 and the distance between burners was set to 95 mm. In our traditional two-stage combustion experiment with partially premixed primary flame, the $NO_X$ exhaust concentration was 59.1 ppm. In contrast, for the two-stage combustion apparatus featuring tubular flame in the primary stage considered in this paper, the $NO_X$ exhaust concentration was 10.2 ppm with water cooling (burner distance 234 mm). Thus, two-stage combustion with tubular flame in the primary stage reduces $NO_X$ exhaust concentration to 1/10 compared to single-stage partially premixed combustion and reduces it to 1/6 compared to traditional two-stage combustion.

**Conclusions**

Using two-stage combustion with a primary burner utilizing tubular flame, $NO_X$ reduction was attempted with supply equivalence ratios near stoichiometric conditions. The equivalence ratio in a primary stage burner increases up to approximately 1.7 by use of tubular flame. Thus, the $NO_X$ concentration decreased with increasing equivalence ratio at the primary combustion. Finally, the $NO_X$ concentration is reduced to approximately 10 ppm at the equivalence ratio of 1.67 in the primary stage, although the extracted heat was approximately 20% of the combustion rate. This $NO_X$ concentration is almost 1/10 of the single-stage combustion with partially premixed flame, and nearly 1/6 of the two-stage combustion with partially premixed combustion as the primary stage.

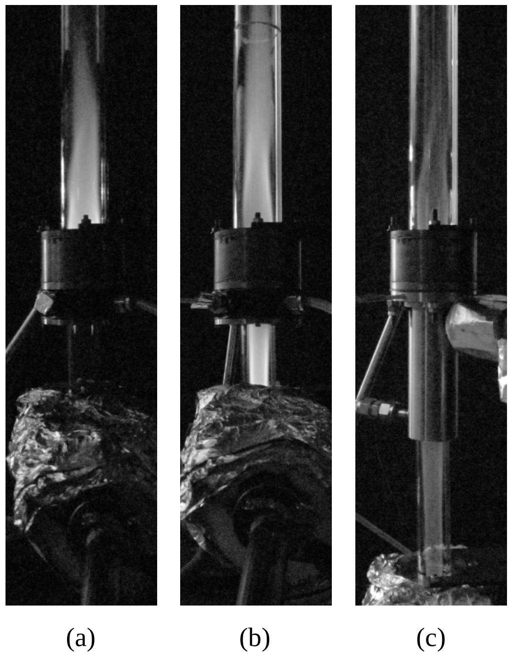


Fig. 6 Direct photos of conventional combustion and two-stage combustion using tubular flame.
((a) Single-stage combustion with Bunsen flame.
(b) Two-stage combustion with Bunsen flame as primary stage, $L_{1st}$=95 mm.
(c) Two-stage tubular flame burner with heat exchanger, $L_{1st}$=234 mm.)

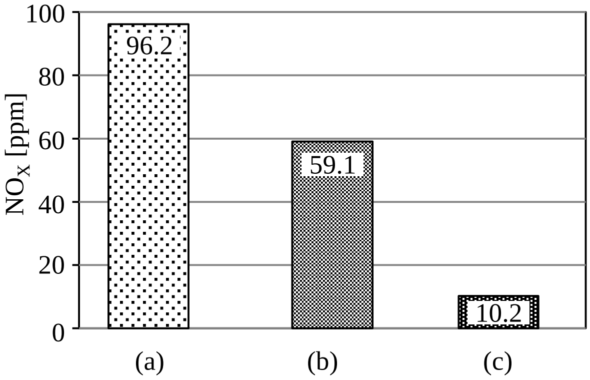


Fig. 7 $NO_X$ concentration of conventional combustion and two-stage combustion using tubular flame.
((a) Single-stage combustion with Bunsen flame.
(b) Two-stage combustion with Bunsen flame as primary stage, $L_{1st}$=95 mm.
(c)Two-stage tubular flame burner with heat exchanger, $L_{1st}$=234 mm.)

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