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Analysis of the combustion oscillation in a silo-type gas turbine combustor and its suppression †

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Abstract

The characteristics of combustion oscillation of a silo-type 79.5 *MW* gas turbine combustor in commercial operation and its suppression have been investigated. The oscillation of the lean premixed gas turbine combustor resulting from the combustion instability occurred at near full load operation. An FFT analysis of the combustion dynamics showed that the dominant frequency of the oscillation would be that of the 1st longitudinal acoustic resonance mode of the combustor. To suppress the combustion oscillation, a passive control technique for reducing the combustion instability was employed; that is, the fuel to the combustor was redistributed by adjusting the operational schedule of one of six fuel control valves, which would lead the increase of the local operational equivalence ratio near the central recirculation zone of the combustor. By doing so, the oscillation was successfully reduced to the permissible level while the amount of NO_x emission met proper regulatory level set by the local government.

Keywords: Combustion dynamics; Combustion instability; Combustion oscillation; Gas turbine; Lean premixed combustion

1. Introduction

Gas turbine facilities currently contribute to approximately 23% of the domestic power capacity in Korea, and it is expected that this share will keep increasing consistently for years to come. At the same time, the environmental regulations on emissions for the power generation facilities, both national and local, are becoming more stringent in Korea. Each power plant is required to follow a self-regulatory agreement on the pollutant emission levels set by the local government in whose jurisdiction the plant is located. To reduce emissions within government-regulated guidelines, combined cycle power plants that have been recently constructed and are in commercial operation or that have been planned to be constructed in Korea incorporate low emission gas turbines that adopt the lean premixed (LPM) combustion technology for reducing the amount of emissions including NO_x. The LPM combustion technology reduces the flame temperature by lowering the equivalence ratio of the fuel and air mixture, resulting in decreased formation of thermal NO_x. The LPM gas turbines operate at a low equivalence ratio; however, their stable, reliable operation margin is very narrow. Hence, these turbines are more susceptible to disturbances that can cause combustion oscillation and flashback more frequently than conventional diffusion type gas turbines. This not only shortens the life of the individual combustor components but also causes severe mechanical damage on the turbines, possibly leading to power failure of the whole power plant [1].

Recently, we observed and analyzed the characteristics of the combustion oscillation occurring in a silo-type gas turbine combustor in commercial operation that employs the LPM combustion technology and then successfully suppressed the oscillation to an operable level by practicing a combustion tuning technique [2]. In this paper, we elaborate the analysis and discussion of the previous experimental and numerical results.

2. Combustion oscillation in a silo-type gas turbine

2.1 Combustion oscillation

The LPM combustor, which mixes fuel and air prior to the combustion in such a way that the resulting equivalence ratio of the fuel and air mixture becomes lower than the stoichiometric ratio, can reduce the thermal NO_x formation by lowering the flame temperature. There are no air dilution holes around the LPM combustor, in contrast to the conventional,

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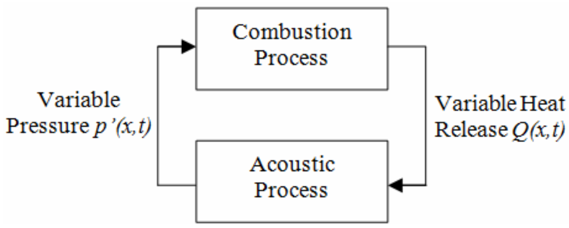

Fig. 1. Schematic of processes occurring during the combustion oscillation [3].

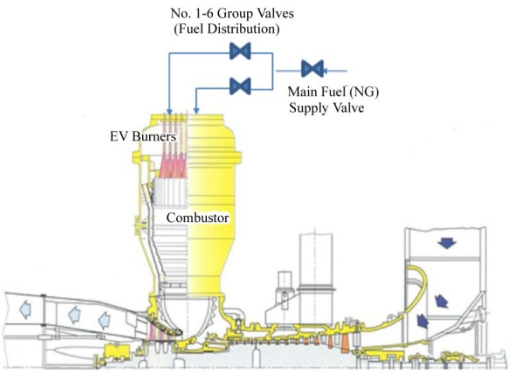

Fig. 2. Cut-away view of a silo-type gas turbine combustor being studied.

non-premixed combustor. Thus, it is more likely to become unstable due to its lean combustion and the resulting instability can be easily amplified to a certain limit cycle of the combustion oscillation. In other words, combustion oscillation in the LPM combustor occurs when the perturbed acoustic pressure wave within the combustor is fed back and coupled with the heat release rate [3].

This concept can be demonstrated in a closed loop system, as shown in Fig. 1. A change in the heat release rate *Q(x,t)* originating from the fluctuation in the fuel and/or air supplies, resulting in the change in the equivalence ratio, causes a sudden disturbance of acoustic field inside the combustor and its periphery. The resulting combustion noise is further reverberated by the combustor that governs the acoustic losses, and thus the acoustic pressure *p'(x,t)* is exacerbated. When the acoustic pressure is coupled with the change in the heat release rate and is amplified, combustion oscillation occurs. The Rayleigh criterion [4], defined by Eq. (1), tells the condition at which the combustion oscillation occurs:

$$\frac{1}{T}\int_0^T \int_V p'(x,t)Q(x,t)dVdt > \textbf{acoustic losses}. \qquad (1)$$

2.2 Silo-type gas turbine combustor

A silo-type gas turbine combustor of ABB 11N (see Fig. 2) operated at a local combined-cycle power plant has been investigated in this study. It is an LPM type combustor that burns natural gas as the main fuel and distillate oil (DO), and

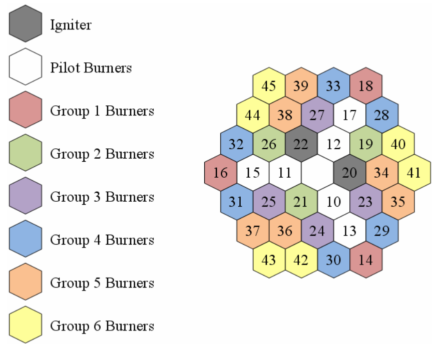

Fig. 3. Schematic arrangement of 36 EV burners installed in the silo-type gas turbine combustor being studied.

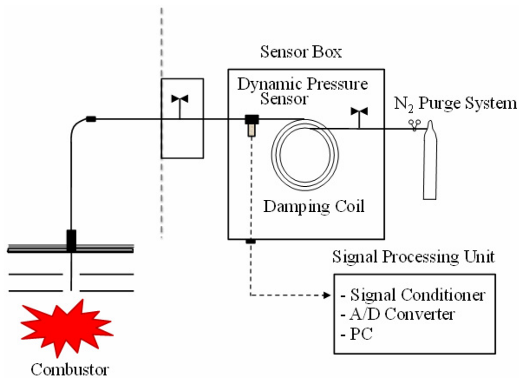

Fig. 4. Schematic of measurement system of the combustion dynamics.

its rated output power is 79.5 *MW*. As shown in the figure, the single silo-type gas turbine combustor contains 36 EV (environmentally friendly V-shaped) burners whose arrangements are shown in Fig. 3.

The EV burners are divided into eight functional groups: igniters, pilot burners, and six groups of fuel burners designated as Groups 1 through 6. Groups 1 through 6 have their own control valves to control fuel flow of each group, according to the operation schedule. For example, the Group 1 and the Group 2 valves are fully open at the relative output power of about 25% and 10%, respectively while the other groups' valves open with different rates at an output power of 30% or higher. The operation schedule can be modified within an allowable limit.

An EV burner, known to be advantageous for preventing flashback, is comprised of two bisectional cones combined together in a staggered mode to form one conic piece. The gaseous fuel supplied through a series of tiny holes is premixed to a high swirling air flow with strong axial momentum prior to its combustion.

3. Experiments and discussion

3.1 Combustion dynamics

Fig. 4 shows a schematic of the system used in this study to

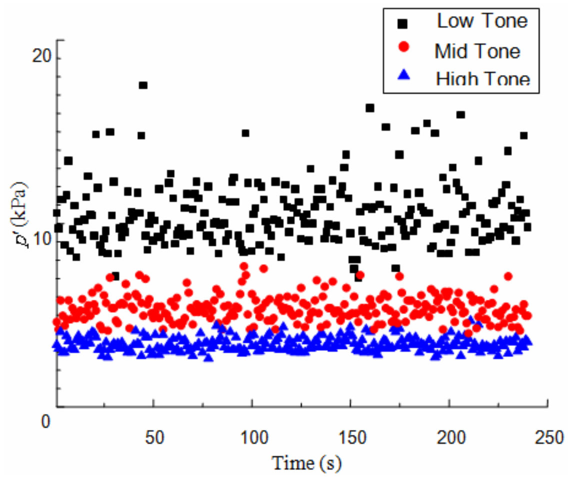

Fig. 5. Typical combustion dynamics in each band frequency.

measure the combustion dynamics occurring in the silo-type gas turbine combustor of interest. A 1/4 inch stainless steel tubing is flush-mounted along the exit plane of the EV burners so that the dynamic pressure within the combustor can be detected by a sensor installed in the external sensor box. The tubing is further connected to the N_2 purge system through the damping coil. The piezoelectric dynamic pressure sensor (PCB M205) picks up pressure changes, converts it into an electric signal, and relays information to the signal processing unit that contains a signal conditioner, A/D converter, computer with data acquisition board (NI PXI4472), etc. A LabView code has been prepared for the analysis of the dynamic signals.

The combustion dynamics of the silo-type gas turbine combustor during its normal operation (that is, within allowable combustion oscillation) has been acquired and analyzed. Fig. 5 shows a typical time-series analysis of the combustion dynamics in normal operation, representing it into three different frequency bands: below 200 Hz, between 200 Hz and 500 Hz, and above 500 Hz. As shown in the figure, the combustion dynamics prevails in the low frequency range (low tone), and its peak-to-peak value lies in the range of 7 kPa to 17 kPa.

3.2 Combustion oscillation

The silo-type gas turbine combustor did experience combustion oscillation once at the relative load of 98%. The oscillation caused severe mechanical vibration of the single shaft of the gas turbine facility, and thus reduced the output level of the facility to a level much lower than rated. During this unstable operation, its combustion dynamics were also measured with the system described in Fig. 4, and a typical result is shown in Fig. 6. The figure shows that its peak-to-peak value appears to be between 20 kPa and 25 kPa. The corresponding frequency analysis (see Fig. 7) shows that its dominant frequency is around 125 Hz.

Acoustic resonance frequencies f_{mnq} [5] of the gas turbine combustor that can be modeled as a cylindrical cavity at the specific conditions were calculated with Eq. (2):

Table 1. Major dimensions and operation data of the combustor investigated.

Item	Value
Radius of the simplified combustor, *m*	1.000
Length of the simplified combustor, *m*	3.100
Power output,[1)] *MW*	80.0
Fuel flow rate, *kg/s*	5.19
Turbine inlet temperature (TIT), *°C*	1,027.0
Equivalence ratio	0.6

[1)] 98% of the base load which varies with the ambient temperature.

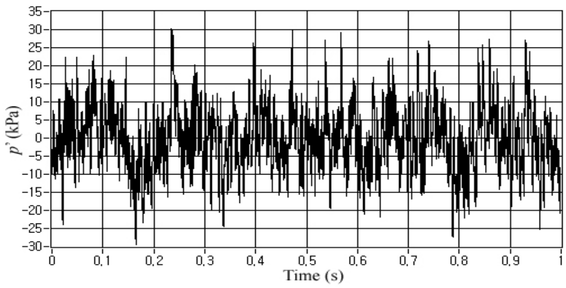

Fig. 6. A typical time-series result of the combustion dynamics when the oscillation occurs.

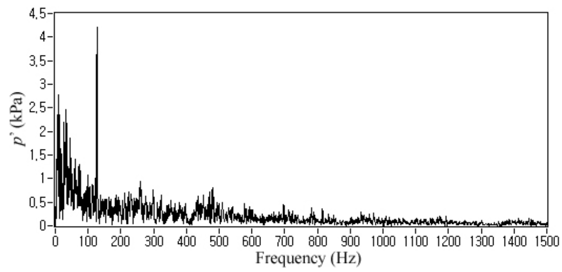

Fig. 7. The corresponding FFT result of the combustion dynamics shown in Fig. 6.

$$f_{mnq} = \frac{c}{2}\sqrt{\left(\frac{\lambda^2{}_{mn}}{R_c{}^2} + \frac{q^2}{L_c{}^2}\right)} \tag{2}$$

where c, λ_{mn}, L_c, and R_c represent the sonic speed (840 m/s for a typical gas turbine combustor [5]), the transverse eigenvalue (0 for 1L mode, that is, m = 0; n = 0; q = 1 [5]), the length and the radius of the simplified cylindrical combustor, respectively. The data used for the calculation are listed in Table 1. The frequency for the 1L (longitudinal) mode is determined to be about 135 Hz, and it is comparable to the measured dominant frequency of the combustion oscillation of 125 Hz. This may

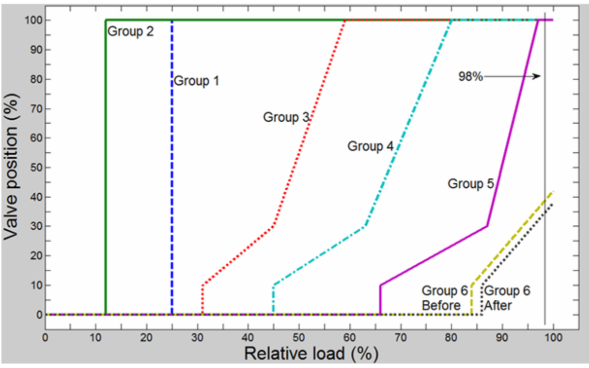

Fig. 8. Change in an operation schedule of a silo-type gas turbine combustor being studied.

be explained by the combustion instability of the silo-type combustor, causing the combustion oscillation observed to be coupled with the 1L mode acoustics of the combustor, and thus change the heat release rate during the combustion.

3.3 Suppression of combustion oscillation

To prevent excessive combustion oscillation, which is determined by the acoustics and combustion conditions of the combustor, it may be logical to consider either modifying the geometry of the combustor hardware or changing its operation condition. Since we dealt with a silo-type gas turbine combustor in commercial operation and it is not feasible to alter the geometrical configuration of the combustor, we chose to vary its fuel distribution pattern: the local equivalence ratio of the fuel and air mixture. This is one of the promising passive control techniques for reducing combustion instability [6] which tend to be applied on a trial and error basis. It is also reported that when the operation equivalence ratio of an LPM gas turbine is increased close to the stoichiometric ratio, the change in heat release rate becomes lower, possibly leading to the reduction of the combustion instability [7].

To increase the local operational equivalence ratio of the silo-type LPM gas turbine, the fuel supply schedule of Group 6 valve in Fig. 3 was altered slightly (see Fig. 8). We chose Group 6 because the maximum opening ratio of its control valve is a fraction of the full opening even though its relative load reaches the base load, compared to the other groups whose opening ratios reach 100% opening at a comparable partial load. According to its original schedule, the Group 6 valve was designed to start to open to 10% of the valve position at the relative load of 84% and then continue to open to approximately 38% at the relative load of 98% at which the combustion oscillation occurred. We varied the opening relative loads (for example, 82%, 84%, 86%, etc.), and valve positions at 98% relative load, resulting in the change of the slope of the curve. After several trials, a modification case shown in Fig. 8 (Group 6 After) did reduce the combustion dynamics of the silo-type gas turbine; the Group 6 valve started to open at around 86% and its position at 98% relative load was 34%.

Table 2. Relative change in gas flow for each burner Group at relative load of 98% when the schedule of Group 6 valve is modified.

Burner group	1	2	3	4	5	6	Pilot
Relative change in gas flow (%)	0.47	0.57	0.57	0.57	0.58	-3.65	0.60

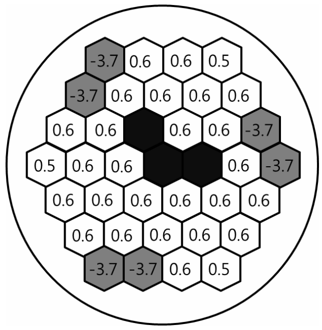

Fig. 9. A sketch indicating the percentile change in local equivalence ratios after the operational schedule change. Region in black color indicate two igniters and a burner that is not in use. (See Fig. 3)

This would have resulted in an increase in the local equivalence ratio of the fuel and air mixture through the other EV burners.

To see how much the operation schedule change, as shown in Fig. 8, affects the local equivalence ratio of the gas turbine, a rather simple numerical simulation was conducted by utilizing a commercial code, Flowmaster [8]. Similar to a previously reported study [9], dimensions of the fuel supply system and its operation conditions of the silo-type gas turbine including the schedule were provided to the code, and transient flow analysis was conducted. Based on an assumption that the total gas flows before and after the operational schedule change are the same, the simulated relative changes in gas flow through each burner group at the relative load of 98% after the Group 6 gas control valve schedule was modified as listed in Table 2.

As shown in the table, the fuel requirement of the Group 6 burners has decreased by delaying the valve opening and decreasing the opening ratio of the Group 6 control valve, while the requirements of the other burners have increased. Since the amount of the combustion air supplied to the combustor of interest would remain constant, the change in the gas fuel supply can be interpreted as equivalent to the change in the operational equivalence ratio. It is therefore expected that the local operational equivalence ratio of the mixture through Groups 1 through 5 and the pilot burners have increased, and we show the increase to be in the range of 0.47% to 0.60%, while that of the Group 6 burners has decreased by 3.65%, as schematically shown in Fig. 9. As shown, the uniformity of local equivalence ratios of 36 EV burners becomes lower, that is, those ratios of central burners get higher after the operational schedule change. This may cause the LPM flame to stretch in the axial direction, resulting in weakened energy coupling between the fluctuation of heat release rate and the

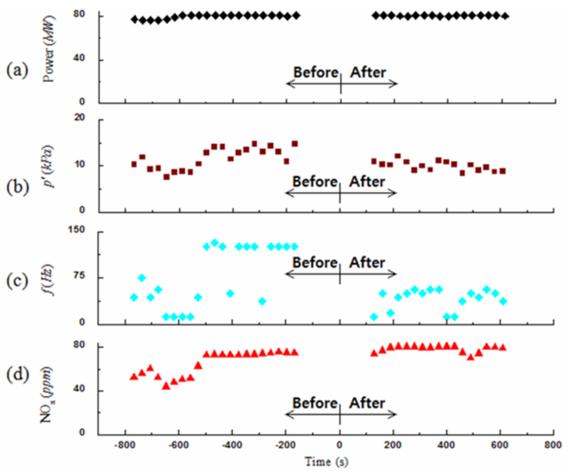

Fig. 10. Time-series of (a) the output power; (b) the magnitude of the combustion oscillation; (c) its dominant frequency; (d) NO_x emission before and after combustion tuning.

perturbed acoustic pressure inside the combustor. This would be the reason that the combustion oscillation was reduced after the operational schedule change. Also, the equivalence ratio fluctuations would be the main mechanism of the suppression of combustion oscillation for this study, even though the corresponding investigation was not conducted, mainly due to the difficulty for performing the research on a gas turbine in commercial operation.

Fig. 10 shows the time-series of the output power, the magnitude of the combustion dynamics, its dominant frequency, and NO_x emission at the vicinity of the combustion adjustment (i.e., $t = 0$ in the figure). Fig. 10(a) indicates that the output power of the silo-type gas turbine remains relatively constant at 80 MW, which is 98% of the base load at that time. Before the combustion adjustment, that is, lowering the opening ratio of Group 6 control valve from 38% to 34%, the dominant frequency of around 125 Hz prevails with the average magnitude of 14 kPa. As previously mentioned, it is believed that the frequency is related to the resonance frequency of the 1L acoustic mode of the combustor.

After the combustor was adjusted, the 125 Hz component of the combustion dynamics disappeared, while the 60 Hz component increased, and the average magnitude decreased to the permissible level. The mechanical vibration of the gas turbine shaft was also decreased to the operable level (data not shown). In contrast, the NO_x emission of the silo-type gas turbine has slightly increased from 72 ppm @ 13% O_2 to 80 ppm @ 13% O_2. This value, however, still met the regulatory level of 100 ppm set by the local government at which the silo-type gas turbine of interest is located.

4. Concluding remarks

Based on the experiment on the combustion oscillation of a silo-type gas turbine in commercial operation, the following conclusions can be drawn:

- The combustion oscillation occurring in the silo-type gas turbine combustor investigated is dominant in the low frequency range below 200 Hz and its peak-to-peak magnitude lies between 7 kPa and 17 kPa.
- More severe combustion oscillation occurred at a specific load of the silo-type gas turbine, with the average magnitude of around 14 kPa at 125 Hz. It appears that the combustion oscillation is related to the resonance frequency of the 1L acoustic mode of the combustor.
- The combustion oscillation could be reduced by applying a passive control technique, that is, the change in the fuel delivery system [6]. This has been done by adjusting the operational schedule of the Group 6 valve, possibly increasing the local operational equivalence ratio of the fuel and air mixture near the central recirculation zone. On the contrary, the NO_x emission has increased slightly due to the combustion tuning, but remained within regulatory levels.
- A more systematic study may be needed to find a better modification of the operational schedule to further lower combustion oscillation while maintaining the NO_x emission below the regulatory level.

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Nomenclature

c : Sonic speed; 840 m/s at a typical gas turbine combustor [5]
f : Frequency
L_c : Length of the simplified combustor
p' : Perturbed pressure
Q : Heat release rate
R_c : Radius of the simplified combustor
T : Period of oscillation
t : Time
V : Volume of the simplified combustor
x : Spatial coordinate
λ : Transverse eigenvalue; 0 for the 1L mode [5]

Subscripts

m,n,q : Indices

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