The effect of heat release modulation on combustion oscillations in lean premixed gas turbine combustors

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Abstract

Combustion dynamics in lean premixed gas turbine combustors is investigated using large eddy simulation (LES) techniques and flamelet model. This study focuses on heat release modulation to change the coupling between pressure and heat release fluctuations during oscillatory combustion. As a first step, a passive control method is incorporated into a model combustor system, based on time-lag theory. The combustor inlet duct length is varied to change the convection time of fuel/air mixture pockets. The phase relation between combustion heat from premixed gas and heat release cycle in the combustor describes the stability of the system. The present numerical method produces the same trend as predicted by the theory. The effect of heat release modulation is significant and if correctly used, combustion field will be controlled by it. We then apply the same methodology to our model combustor at JAXA to examine secondary fuel injection control. Combustion oscillations in a swirler-type combustor are reproduced numerically. Active combustion control using secondary fuel injection is tested to reduce the pressure oscillation amplitude. It is shown that secondary fuel injection can reduce pressure fluctuation amplitude if the phase angle is properly chosen.

1. Introduction

Lean premixed combustion is commonly recognized as a low-emission combustion method because the flame temperature can be comparatively low and NOx production rate is also low. However, lean premixed combustion tends to be unstable and stability margin is usually small compared to conventional nonpremixed combustion methods. Combustion oscillations may cause serious damage to combustor systems, so combustion control is a key technology to realize stable operation over a wide range of flow conditions.

Dynamics and mechanisms of combustion instabilities and control have been intensively investigated worldwide [1]. Both experimental and numerical studies have dealt with various ranges of combustors and operation conditions. In this research project, combustion control for low-NOx gas turbine combustors is our primary goal.

Recent progress in numerical simulation techniques has enabled us to conduct unsteady combustor system simulations [2,3]. Large eddy simulation methodology can deal with unsteady flows and several flame models have been incorporated in LES calculations, for example, artificially thickened flame model [3] and G-equation flame model [2,3]. The G-equation flame model is a simple but useful method to deal with thin premixed flames. A number of authors have demonstrated its effectiveness in unstable combustion research, for example see [4].

We have also demonstrated the effectiveness of CFD based on LES and G-equation to understand combustion oscillation dynamics. In our study [5], the basic mechanism of combustion oscillation is shown to be the coupling between pressure oscillations, vortex shedding due to velocity fluctuations and heat release fluctuations. In that study, the system does not include local fuel/air mixture ratio variation. If the flow has this local equivalence ratio variation, the phenomena will become more complicated because the heat release field will be significantly changed.

The objective of this study is to investigate the effect of heat release on combustion dynamics in gas turbine combustors. The first part deals with a preliminary study of heat release rate modulation. This is an idealized case with passive control based on time-lag theory [6]. This study is conducted to examine the validity of the present numerical approach and a basic effect of heat release on combustion dynamics. In the second part, this numerical method is applied to our model gas turbine combustor. This study, in conjunction with our experimental approach, will help us to further understand physical mechanisms of combustion control. Corresponding experimental results will be also presented in detail in [7,8].

2. Numerical implementation
2.1. Flow field and flame modeling

The governing equations of flow field are three-dimensional Navier-Stokes equations. Pressure wave propagation in the combustor is one of the key phenomena, so compressibility must be included in the formulation. The conservation equations of mass, momentum and total energy are spatially filtered in the LES context. The unresolved subgrid-scale terms are modeled using the classical Smagorinsky model given by

$$\tau_{ij}^{sgs} = \bar{\rho}\left(\widetilde{u_i u_j} - \tilde{u}_i \tilde{u}_j\right) = -2\bar{\rho}\left(C_s \Delta\right)^2 \left|\bar{S}_{ij}\right| \bar{S}_{ij} \quad (1)$$

where the tilde denotes Favre averaging, bar normal averaging and S strain rate. The coefficient C_s is dynamically determined from local flow structures [2,3].

Flame modeling is based on an assumption that premixed flame thickness is usually small unless turbulence intensity is very high. The well-known G-equation formulation is employed in the present simulation.

$$\frac{\partial \rho G}{\partial t} + \frac{\partial \rho G u_j}{\partial x_j} = \rho_u s_L \left|\nabla G\right| \quad (2)$$

where G is set 0 in the unburned region and 1 in the burned region and s_L is called laminar burning velocity. This model assumes that the flame is a discontinuity surface between unburned and burned gases. The propagation speed of flame is given by CHEMKIN calculation. LES filtering makes the apparent (grid-scale) flame surface area smaller, so the flame propagation speed is increased to compensate the effect of filtering. This propagation speed is called turbulent burning velocity and, in this study, given by an empirical formulation as follows:

$$\tilde{s}_T / s_L = 1 + C\left(u'/s_L\right)^n \quad (3)$$

where u' is subgrid turbulent intensity and the coefficients are given by experimental curve fitting [2,3].

The effect of local equivalence ratio variation is included by introducing the mixture fraction variable. The mixture fraction takes a value between 0 and 1 and represents local fuel/air mixing ratio. The governing equation is [2,3]

$$\frac{\partial \rho Z}{\partial t} + \nabla \cdot \left(\rho \mathbf{u} Z\right) = \nabla \cdot \left(\rho D \nabla Z\right) \quad (4)$$

where D is the thermo-diffusion coefficient. The mixture fraction and local equivalence ratio are linked by

$$Z = Z_{st}\phi \big/ \left(1 - Z_{st} + Z_{st}\phi\right) \quad (5)$$

where Z_{st} is the stoichiometric mixture fraction.

Local variation in equivalence ratio has both direct and indirect effects on heat release. The local flame propagation speed and combustion heat are directly dependent on equivalence ratio. They increase as the local equivalence ratio increases in the lean region. Flame surface area may be affected indirectly by local equivalence ratio through the burning velocity and flow field velocity. LES filtering requires modeling of an unresolved term and the classical gradient diffusion modeling is used here.

2.2. Theoretical background

The stability of a combustor system is described by Rayleigh's criterion. The acoustic energy increases when

$$\frac{\gamma - 1}{\overline{\rho c}^2} \int p' \dot{q}' dV > \int p' \overline{u} dA \quad (6)$$

is satisfied. The left hand side is the coupling between pressure and heat release fluctuations and the right hand side is acoustic energy flux balance through boundaries. Usually in premixed combustion systems, the right hand side is small. When the phase difference between pressure and heat release is within 90 degrees, the system is unstable.

The essence of combustion control lies in how to change the coupling between pressure and heat release. Changing the pressure field by loudspeakers or acoustic dampers is one way of combustion control. This method is usually applied to small systems. Another way is to change the heat release field directly. In this project, secondary fuel injection is used to change the flow field both temporally and spatially. Thus, the effect of heat release modulation should be understood to implement effective combustion control.

Time-lag theory describes the essential effect of phase-difference in a combustor system [6]. When acoustic pressure oscillations are excited in a combustor, flow rates of air and fuel in the inlet duct are affected by the combustor pressure magnitude. This will create rich/lean premixed gas pockets periodically at the injection position and they are convected downstream. At the flame position, these pockets burn and induce heat release fluctuations. The ratio of convection time to oscillation period is a key parameter to determine the phase relation between the local equivalence ratio at the flame and the original heat release oscillation. If the heat release fluctuation induced by equivalence ratio stratification is in phase with the original heat release, it will amplify the oscillation. If it is out of phase, the oscillation amplitude will get smaller. The phase relation for amplifying oscillation is described by

$$\tau_{convect} / T = 1/2, 3/2, \cdots \quad (7)$$

for unchoked fuel injection. The convection time is obtained by

$$\tau_{convect} = \left(L_{inlet} + L_{flame}\right) \big/ \overline{u} \quad (8)$$

Figure 1 shows the combustor system considered here.

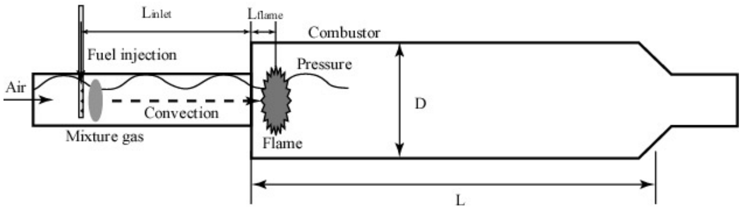


Figure 1 Model combustor configuration

When secondary fuel is actively injected, the basic principle is the same as described above. Changing the heat release field both temporally and spatially by secondary fuel injection, the phase relation between pressure and heat release is modulated. The problem in a real situation is that it is not so simple. Usually there are multiple excited acoustic modes. Thus, active combustion control is needed and an appropriate control algorithm must be found to stabilize the system over a wide range of operation conditions.

2.3. Combustor configurations

Preliminary model combustor

For the preliminary study, we use a combustor that was also used in the previous study [5] and this is schematically shown in figure 1. The combustor has a circular cross section and a length of 0.3m. An inlet duct is connected to the combustor and swirl motion is added to the premixed flow. The fuel used here is methane. Mixing between air and fuel is assumed to occur at the inlet entrance instantly. The fuel injection holes are modeled as unchoked ports. The mass flow rate of fuel is determined by the isentropic relation between the inlet pressure and the reservoir tank pressure that is initially specified. The boundary condition for airflow is based on the characteristic relation.

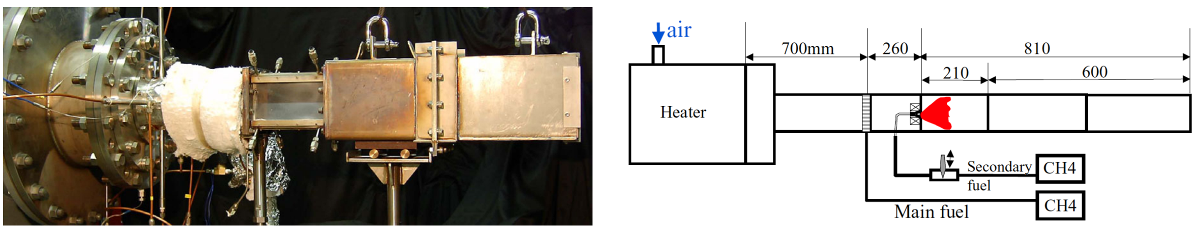
The inlet duct length is varied from 0.056m to 0.095m to change the convection time of premixed gas pockets. The length difference corresponds to the convection length in one acoustic oscillation period of the basic quarter-wave mode. The combustor exit is contracted to accelerate the burned gas.

The numerical grid system used is a comparatively coarse one. The total number of grid points is about 1 million. The flow conditions are listed in Table 1 as cases #1 to #6.

JAXA model combustor

The model combustor considered here is identical to the one used in our experiment at JAXA. Figure 2 shows a picture of the combustor and combustor dimensions. The combustion chamber is a 0.1m x 0.1m square-cross-section duct with length of 0.21m-0.81m. The fuel used here is also methane. A swirler inlet duct is connected to the combustor for flame stabilization. The inner and outer diameters of the swirler are 0.02m and 0.05m, respectively. The experimental swirl vane angle is 30 degrees. In this numerical simulation, swirl vanes are not directly included in the numerical mesh. Swirl motion is added to the premixed gas by applying a hypothetical force in the inlet duct. The swirler has 12 vanes inside and 8 secondary injection holes placed circumferentially near its hub edge. The hole diameter is 1.4mm and the angle of injection is 30 degrees. The injection fuel is supplied from a reservoir tank of methane and the typical ratio of secondary fuel is about 5% of the main fuel. The combustor exit is not contracted but an open end.

The corresponding numerical region is comprised of three sections, inlet duct, combustion chamber and exit region, as illustrated in figure 2. In the outer region, it is assumed that the entire region is filled with burned gas flow. The outer region extends downstream more than 10 times of the combustor length. The total number of grid points ranges from 12 to 15 million depending on the combustor length.



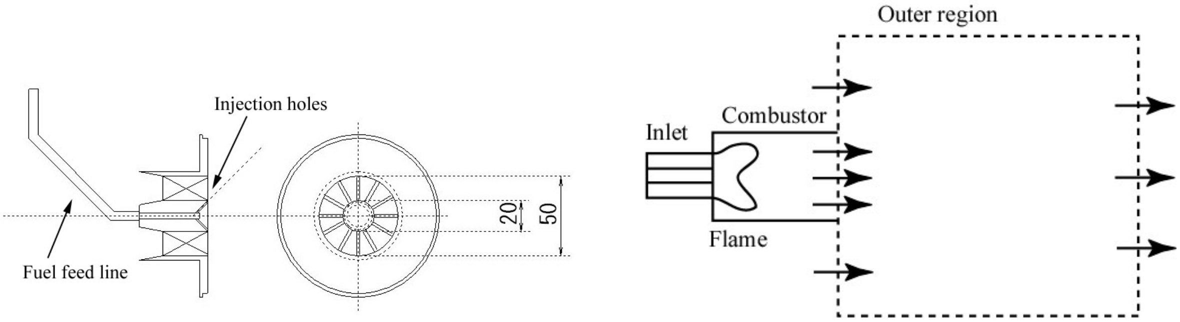


Figure 2 JAXA model combustor, swirler and numerical domain

Boundary conditions for incoming boundary are based on characteristic wave equations. The exit conditions are constant pressure boundary conditions. Typical operation conditions are listed in Table 1 as case #7. In this case, the combustor length is 0.21m. Secondary fuel injection is implemented in cases #8 and #9.

Table 1 Premixed gas conditions

Case	Temperature (K)	Velocity (m/s)	Equivalence ratio	Pressure (atm)	Inlet length (m)	Secondary injection
1	400	30	0.6 (fixed)	1.0	0.0764	--
2	400	30	Variable	1.0	0.0564	--
3	400	30	Variable	1.0	0.0664	--
4	400	30	Variable	1.0	0.0764	--
5	400	30	Variable	1.0	0.0864	--
6	400	30	Variable	1.0	0.0964	--
7	700	30	0.6 (fixed)	1.0	--	No
8	700	30	0.6 (fixed)	1.0	--	Yes (150deg)
9	700	30	0.6 (fixed)	1.0	--	Yes (60deg)

3. Results

3.1. Preliminary passive control based on time-lag theory

Using the first combustor, the effect of heat release modulation is investigated. By taking the combustor pressure history, the basic mode is found to be quarter-wave mode where a pressure node is located at the combustor exit and an antinode is located near the combustor entrance. FFT results of combustor pressure history also confirm this mode [5]. The oscillation frequency is determined by the combustor length and the speed of sound of burned gas. In this case, the frequency is about 700Hz.

By the combustor pressure oscillations, the inlet fuel and air rates are affected. Figure 3 shows the time history of pressure and local equivalence ratio at the inlet duct entrance for case #5. Two profiles are in reverse phase because the fuel injection holes are unchoked and we assume instant mixing. Rich/lean mixture pockets are created periodically and convected downstream. Figure 3 also shows an instantaneous snapshot of spatial equivalence ratio distribution and flame shape. The upper half of the combustor is shown here. Stratification in local fuel/air ratio is clearly observed.

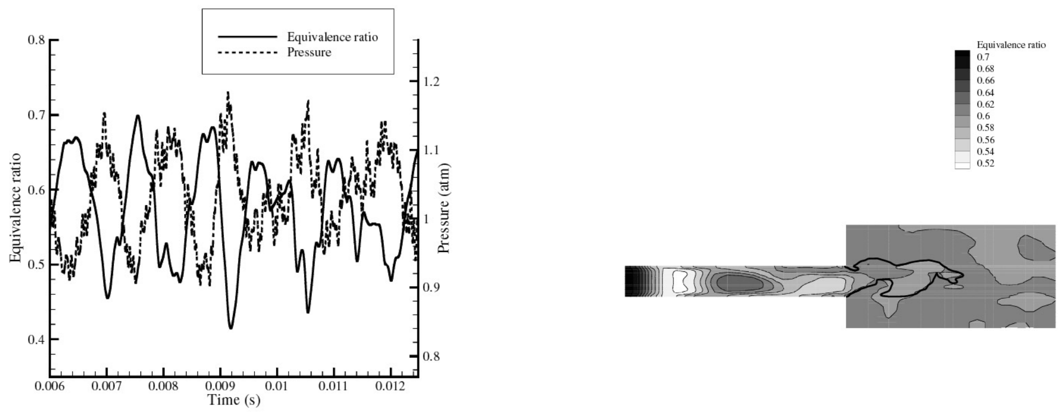


Figure 3 Time history of pressure and inlet equivalence ratio (left) and an instantaneous image of equivalence ratio distribution and flame shape (right)

Variation in the inlet duct length, i.e. the convection time, finally results in the difference in combustor pressure amplitude. Figure 4 shows the trend observed in the present simulation. A shaded bar shown in the figure indicates the maximum amplification condition estimated by the time-lag theory. As the inlet duct length is varied corresponding to one cycle of oscillation, the trend shows one cycle of amplitude variation. The trend is qualitatively in good agreement with theoretical prediction. The minimum pressure amplitude becomes almost half of the maximum amplitude case.

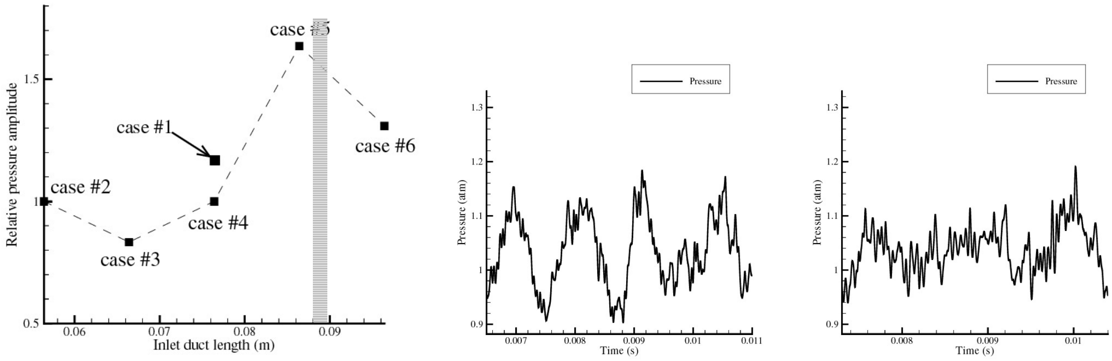


Figure 4 Trend of combustor pressure vs. inlet duct length (left),
pressure traces for case #5 (maximum amplitude) (center) and case #3 (minimum amplitude) (right)

Then, the time histories of equivalence ratio at flame position and global heat release for cases #5 and #3 are compared in figure 5. The global heat release has contributions from combustion heat and flame surface area. The global heat release rate is estimated by

$$\dot{H} = \int \bar{\rho}_u \tilde{s}_T \Delta h d\tilde{A}_f \tag{9}$$

where $\tilde{A}_f$ is the flame surface area. Higher equivalence ratio means larger combustion heat and larger turbulent burning velocity. As can be seen in figure 5, in the maximum pressure amplitude case (#5), the equivalence ratio and global heat release are in phase to amplify each other and in the minimum pressure amplitude case (#3), they are rather out of phase to cancel each other.

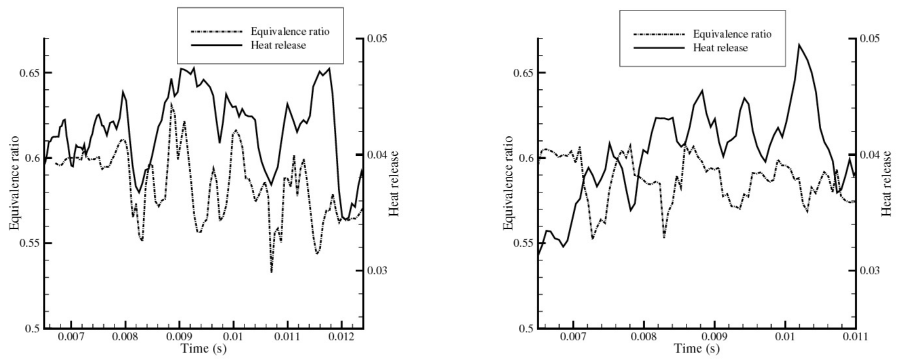


Figure 5 Traces of global heat release (solid) and equivalence ratio at flame (dashed)
case #5 (left) and case #3 (right)

It has been shown that changing local heat release passively by premixed gas stratification has both positive and negative effects on oscillation amplitude in the combustor. If the combustor configuration is properly chosen, it may help to attenuate pressure oscillation amplitude. However, one drawback of this passive control method is that it can only deal with one specified oscillation frequency. And if the flame length is not so compact, it is difficult to define the exact convection time. Thus, active combustion control, in which combustor monitoring and fuel injection are actively linked, is expected to overcome this difficulty.

3.2. JAXA model combustor simulation

Now the effectiveness of heat release modulation has been basically confirmed and the capability of the present numerical method has been demonstrated in the previous section. We then apply this numerical approach to the model gas turbine combustor being investigated at JAXA. The flow conditions are listed in Table 1 as case #7 and #8. The main premixed gas is perfectly mixed and the main-flow equivalence ratio is set constant. The pressure is set at 1atm. These conditions are identical to experimental conditions [7].

Figure 6 shows an instantaneous temperature field formed in the model combustor for case #7. The flame shape changes at every moment due to turbulent flow field. The flame spreads outward due to swirling motion of the premixed gas. In the center region, a recirculation zone is created to push the burned gas upstream. This is the principle mechanism of flame stabilization of swirled flows.

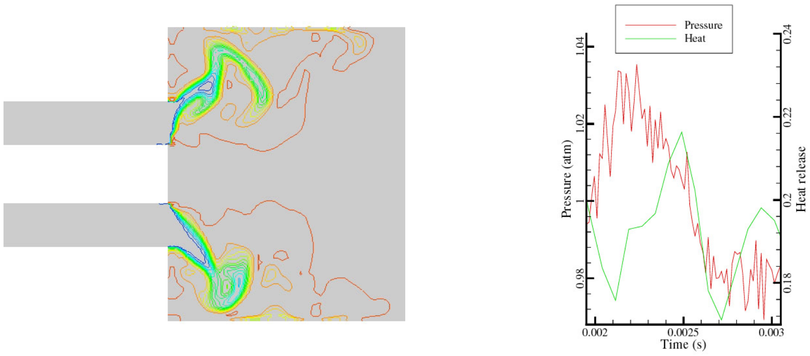


Figure 6 Instantaneous flame shape in the model combustor (The actual combustor length is longer than shown.) (left) and pressure and global heat release traces in one cycle (right)

Inside the combustor, several acoustic modes are excited, and the basic acoustic mode is found to be the quarter-wave mode. In a very simple estimation, the quarter-wave mode frequency can be obtained by the combustor length and the speed of sound of burned gas by

$$f = a/4L \tag{10}$$

In this case, the combustor length is 0.21m and the burned gas temperature is about 2000K, one obtains 1000Hz by (10). But actually the frequency tends to be smaller than this value. This is partly because the pressure node is not located exactly at the combustor open end, but slightly downstream, and the combustor partly includes the unburned gas near the swirler. The observed frequency in the calculation is around 740Hz. Figure 6 shows the combustor pressure and global heat release profiles in one cycle. Oscillations in pressure produce velocity fluctuations in the flow field. This leads to temporal and spatial change in flame shape and position. Even if the equivalence ratio is constant, the heat release rate changes due to flame surface area fluctuations. This coupling mechanism sustains oscillations in the combustor. The phase difference between pressure and heat release is within 90 degrees, which satisfies the Rayleigh criterion. The peak-to-peak oscillation amplitude in pressure is 4.9kPa. This value is close to the experimental value of 5kPa.

In the next step, secondary fuel injection is applied to this combustor to reduce combustion oscillation amplitude. The secondary fuel is injected through eight holes near the swirler hub edge. The amount of injected fuel is about 5% of the main fuel flow rate. The main premixed gas flow remains unchanged, so the total equivalence ratio slightly rises.

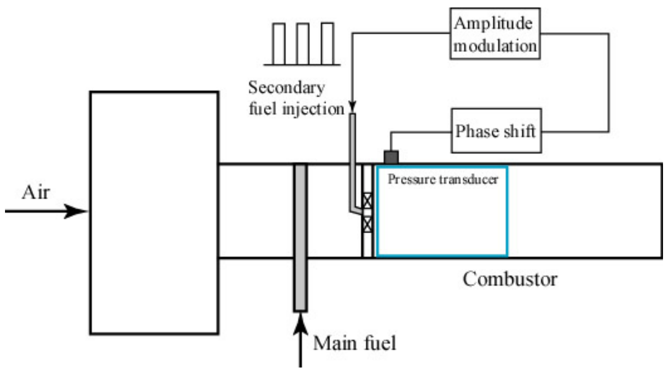


Figure 7 Schematic diagram of feedback control

The injection algorithm used here is a 'phase-shift' approach used also in our experiment [7,8]. Figure 7 shows the schematic diagram of active feedback control for this combustor. A pressure transducer placed 20mm downstream from the combustor entrance measures the combustor pressure. Here, we assume choked fuel injection for secondary fuel, so the mass flow rate is fixed at 25l/min at 1atm. The Rayleigh criterion tells us that the phase angle should set to decouple heat release fluctuation from pressure fluctuation. From the previous result, the phase difference between pressure and heat release fluctuations is 60 degrees, so the phase delay between pressure fluctuation and injection is set at 60 and 150 degrees to make a comparison. The injection is based on on-off flow rate control and an opening (closing) time length is half the oscillation period.

Reducing $\int p'\dot{q}'dV$ in the acoustic energy amplification term in equation (6) by 'good' secondary fuel injection is schematically shown in figure 8(a). The vertical cyan arrow at point A indicates the contribution from secondary fuel injection. Because secondary fuel will generally increase the instantaneous heat release rate, it should be injected when the original heat release rate is smaller than the mean value. Canceling the main heat release rate fluctuation leads to smaller correlation between pressure and heat release fluctuations (see eq. (6)), thus reducing the overall acoustic energy and combustor pressure oscillations as a result. In contrast, when secondary fuel injection timing is not good, for example as shown in figure 8(b) at point B, it will increase acoustic energy and have a negative effect on the system stability.

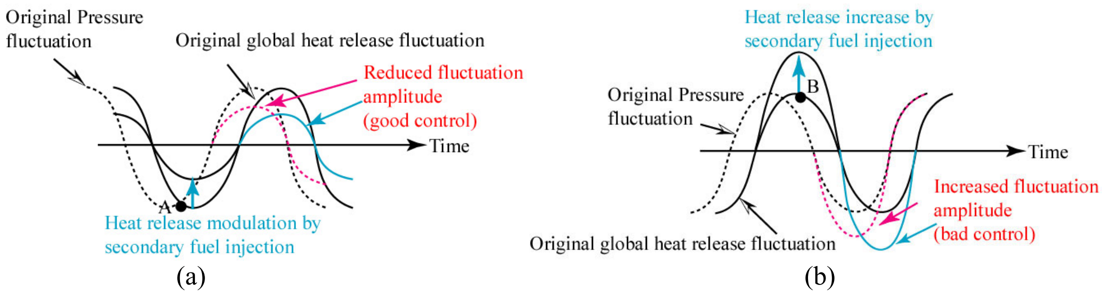


Figure 8 Schematic of correlation between pressure and heat release fluctuations with/without injection (a) good control and (b) bad control

The injected fuel forms another flame region, increasing the global heat release rate as a result. Small partially premixed flames develop from the injection positions when the injection is on. Since the flow field is partially premixed, heat release occurs at two locations, premixed flame front and diffusion flame front. Here, the volumetric heat release rate of a premixed flame is calculated by

$$\dot{q} = \bar{\rho}_u \tilde{s}_T \Delta h \tilde{A}_f / V \tag{11}$$

where V is a volume of a cell considered and $\tilde{A}_f$ is the flame surface area, while that for a diffusion flame is calculated by

$$\dot{q} = \bar{\rho}\sqrt{\tilde{D}\tilde{\chi}/2}\Delta h \tilde{A}_f / V \tag{12}$$

where $\tilde{\chi} = 2\tilde{D}\left|\nabla \tilde{Z}\right|^2$ is the scalar dissipation rate. The injected fuel diffuses into the main flow after passing the flame region, and then the next injection period comes. These are periodically repeated in the cycle.

In figure 9, the instantaneous flame shape and mixture fraction distribution on the flame for cases #8 and #9 are shown. The left figure (case #8) corresponds to point A shown schematically in figure 8, where the main heat release is minimum in the cycle. The right figure corresponds to point B where the main heat release is largest. The heat release rate from the main premixed flame is linked to the flame surface area. The main premixed flame surface area of case #9 (right figure) at this moment is larger than that of case #8 (left figure) by 20%, while the heat release rate is larger by 25%. The injected fuel flows and secondary flames can be seen in the figure. Eight flames are formed from the injection holes and the mixture fraction here is the stoichiometric value. The phase relations with the main flame heat release fluctuation are different between these two cases and critical to the system stability, as shown below. In case #8 (left figure), the heat release from the secondary flames work positively and, in case #9, negatively.

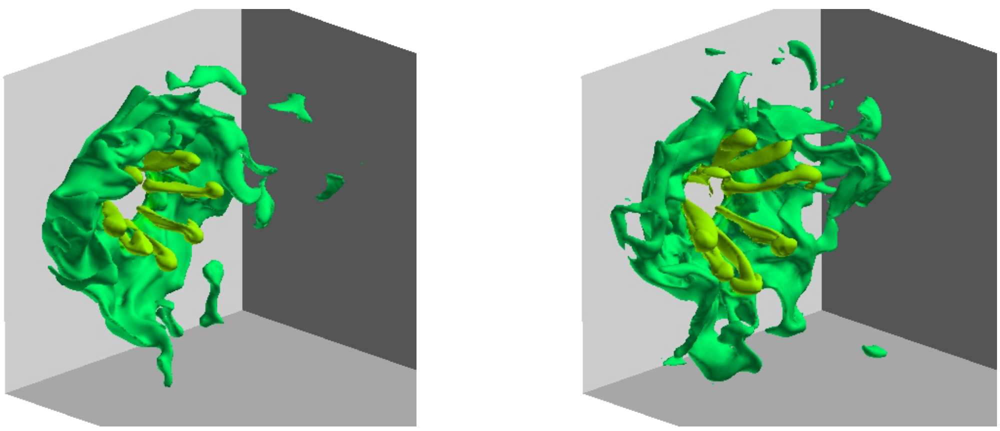
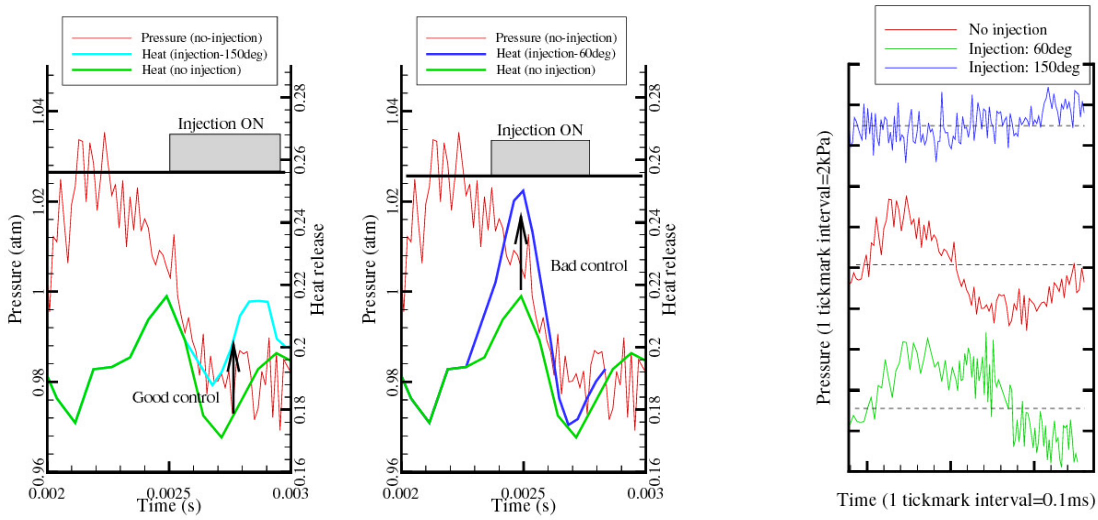


Figure 9 Instantaneous flame shape and mixture fraction distribution on flame (left) case #8 corresponding to point A in figure 8 and (right) case #9 corresponding to point B

Figure 10 shows the temporal traces of global heat release rates and combustor pressure in one cycle with and without secondary fuel injection (cases #7-9). When the phase delay is set at 150 degrees (case #8), which is indicated in figure 10(a) as 'Good control', the global heat release fluctuation amplitude is reduced when the injection is on, because the main heat release is increased by the injected heat release when it is small. This results in decrease in pressure fluctuation amplitude by more than 70%, even when the total equivalence ratio is slightly increased by secondary fuel injection. In contrast, injecting fuel at the phase delay of 60 degrees (case #9) results in increase in the global heat release rate fluctuation, as indicated in figure 10(b) as 'Bad control', causing the combustor pressure fluctuation amplitude to slightly increase by 5% as a result. This is because the injected fuel causes the global heat release to increase when the original heat release is high, i.e. $p'\dot{q}'$ increases.

The resultant combustor pressure profiles in one cycle are shown in figure 10(c). This figure shows three profiles in one figure, and trace plotting for cases #8 and #9 (blue and green lines) starts just before injection begins. The red line (case #7) is plotted for reference to see the amplitude without injection. The real time axes of three cases are in fact shifted to see the difference in one figure. The vertical axis is for comparison of the magnitude of oscillation, and the dotted lines indicate the mean pressure values. The actual mean pressure values are almost the same between the three cases. As can be seen, the pressure amplitude is reduced in case #8. This result indicates that appropriate choice of injection phase angle is significant in determining the response of a combustor system.



(a) (b) (c)

Figure 10 Temporal traces of global heat release rates ((a) case #8 and (b) case #9) and (c) resultant combustor pressure histories with and without injection in one cycle

It has been shown that changing the global heat release rate by secondary fuel injection has an effect on pressure fluctuation amplitude. However, we have not yet obtained optimal injection conditions, for example, injection mass rate, velocity, angle, etc. And we have only focused on the global effect of injection. More simulation cases will be implemented in the future to investigate the local effect in detail.

Simulation cases with a longer combustion chamber duct are also calculated. This will reduce the basic acoustic frequency and secondary fuel injection will be more easily implemented in experiment. In experiment with this duct length, oscillation attenuation has been successful under certain conditions [7,8]. Detailed investigation of physical mechanisms is now underway both experimentally and numerically.

In terms of NOx production, secondary fuel injection may increase NOx emission under some conditions. In this numerical method, NOx emission estimation is not directly included. This will be our future issue.

4. Concluding remarks

LES calculations based on the flamelet assumption have been conducted to investigate the effect of heat release modulation on combustion dynamics in gas turbine combustor systems. The following major conclusions have been obtained by the present analysis.

- In the first stage, passive control based on the time-lag theory is applied to a model combustor. Changing the inlet duct length leads to difference in pressure oscillation amplitude. The convection time of premixed gas pockets plays an important role in determining the stability of the system. This is because it determines the phase relation between the original heat release rate fluctuations and rich/lean heat release rate fluctuations induced by convected premixed gas pockets. The trend is in good agreement with theoretical prediction, and the basic mechanism of heat release modulation has been confirmed.
- The same numerical method is applied to our experimental model combustor installed at JAXA. The basic

flow field is also reproduced by our simulation and pressure oscillations are present in the combustor. Active control using secondary fuel injection is being tested to reduce the oscillation amplitude. Heat release rate can be modulated both locally and globally by secondary fuel injection. The fuel injection is based on the phase-shift approach, and it is shown that the combustor pressure oscillation amplitude is affected by injection timing. The injection should be made when the original heat release is small to reduce the oscillation amplitude, as the Rayleigh criterion describes. It is clear that the injection phase angle is one of the most important parameters for successful active control. Optimal injection conditions have not been obtained yet, and this is one of our future issues.

5. References

1. Candel, S., Proc. Combust. Inst. 29 (2002) 1-28
2. Peters, N., *Turbulent combustion*, Cambridge University Press, 2000
3. Poinsot, T. and Veynante, D., *Theoretical and numerical combustion*, Edwards, 2001
4. Stone, C., Menon, S., Proc. Combust. Inst. 29 (2002) 155-160
5. Shinjo, J., Mizobuchi, Y. and Ogawa, S., Proceedings of the 4th Symposium on Smart Control of Turbulence, Tokyo, 2003
6. Lieuwen, T., Zinn, B.T., Proc. Combust. Inst. 27, pp1809-1816, 1998
7. Tachibana, S., Zimmer, L., Yamamoto, T., Kurosawa, Y., Yoshida, S. and Suzuki, K., Proceedings of the 5th Symposium on Smart Control of Turbulence, Tokyo, 2004
8. Zimmer, L. and Tachibana, S., Proceedings of the 5th Symposium on Smart Control of Turbulence, Tokyo, 2004