Combustion Oscillations in Burners with Fuel Spray Atomisers

M. Zhu, A.P. Dowling and K.N.C. Bray
Department of Engineering, University of Cambridge
Trumpington Street, Cambridge CB2 1PZ, United Kingdom

ABSTRACT

Most types of combustion-driven devices experience combustion instabilities. For aero-engine combustors, the frequency of this oscillation is typically in the range 60-120Hz and is commonly called 'rumble'. The rumble oscillations involve coupling between the air and fuel supplies and unsteady flow in the combustor. Essentially pressure fluctuations alter the inlet fuel and air, thereby changing the rate of combustion, which at certain frequencies further enhances the pressure perturbation and so leads to self-excited oscillations. The large residence time of the liquid fuel droplets, at idle and sub-idle conditions, means that liquid and gaseous phases must both be considered.

In the present work, we use a numerical model to investigate forced unsteady combustion due to specified time-dependent variations in the fuel and air supplies. Harmonic variations in inlet air and fuel flows have been considered and the resulting unsteady combustion calculated. The influence of droplet size distribution has also been investigated. The calculations provide insight into understanding the interaction between atomization, unsteady combustion and flow oscillations.

1. INTRODUCTION

Most types of combustion-driven devices experience combustion instabilities. These instabilities can cause serious problems, such as mechanical vibration, noise, enhanced corrosion, and unpredictable performance. Combustors with fuel-spray atomisers are particularly susceptible to a low-frequency oscillation at idle and sub-idle conditions. For aero-engine combustors, the frequency of this oscillation is typically in the range 60-120Hz and is commonly called 'rumble'. In the past, this has caused an audible, and sometimes disturbing noise in the aircraft cabin. However, combustors are required to have ever wider operating ranges and reduced emissions, and this is leading to design changes which can greatly increase the susceptibility to rumble. Radiated sound levels due to rumble have become excessive, and in some cases the pressure fluctuations have been so intense that they stall the engine. Industry has a pressing need for a better understanding of this instability, if it is to be able to predict the effect of design changes. This should enable aero-engine manufacturers to design combustors which run quietly with low emissions, resulting in obvious benefits for both aircraft passengers and people living near airports.

The rumble oscillations involve coupling between the air and fuel supplies and unsteady flow in the combustor. Essentially pressure fluctuations alter the inlet fuel and air flow rates, thereby changing the rate of combustion, which at certain frequencies further enhances the pressure perturbation and so leads to self-excited oscillations. The large residence time of the liquid fuel droplets, at idle and sub-idle conditions (Tolpadi 1995), means that liquid and gaseous phases must both be considered. The full problem of self-excited oscillation in burners with fuel-spray atomisers is therefore highly complex: it involves a coupled analysis of unsteady two-phase flow, turbulent combustion and acoustics. In the present work, we have concentrated on the most speculative part, using a numerical model to investigate forced unsteady combustion due to specified time-dependent variations in the
fuel and air supplies.

The chosen combustion model involves a Monte Carlo simulation of the Williams' spray equation, together with the equations of turbulent reactive flow, in the form of transport equations for the gas-phase flow field. The spray combustion model has been extended to predict the modulation in the distribution of heat release rate due to specified variations in the fuel mass flow rate and droplet size distribution produced by the fuel injector. These combustion models were applied to the aero-engine geometry.

In section 2, the mathematical models used in the calculations will be briefly described. We have investigated the effect of forced fluctuations in the fuel and air supplies to the atomiser. Several test cases were devised and calculated to examine the possible causes of the rumble phenomenon. The results of numerical calculations will be described in section 3. Finally, general conclusions and further work are discussed.

2. MATHEMATICAL MODELS

The spray flow field is modelled through the equations of motion in Favre ensemble-averaged form. Source terms appear in the continuity and momentum equations due to evaporation and the force exerted by the liquid phase. The frequency of rumble oscillation is far below typical turbulent eddy frequencies so quasi-steady flow conditions will prevail. The turbulence time scales, as well as the local turbulence intensity, are therefore obtained by the well-known $k-\epsilon$ turbulence model, which is known to give satisfactory predictions of a gas turbine combustor flow field under steady flow conditions.

In the simulation, the coupling between the turbulence and the chemical reaction is considered by the laminar flamelet presumed pdf method. The fuel used in the calculations is kerosene. The thermo-chemistry is described in terms of the mixture fraction and its variance, which are obtained by solving their transport equations together with momentum and $k-\epsilon$ equations. The mixture fraction distribution is represented by a beta function, which has been shown to be a good approximation for turbulent jets. The laminar flamelets have been computed and stored in a library, from which values are "drawn" by interpolation during the computation.

The description of the spray is based on Williams' spray equation (Williams 1985). This equation can be written as

$$\frac{\partial f_s}{\partial t} + \frac{\partial}{\partial x_i} [U_i f_s] = - \frac{\partial}{\partial U_i} [F_i f_s]$$

where $f_s$ is the droplet number density function in the joint space spanned by $x$, $U$, $R$ and $e$. Here $U$ is the droplet velocity; $F$ is the total external force on the droplet per unit mass; $R$ denotes the instantaneous droplet radius and $R$ is its rate of change. $Q$ is the rate of change of a droplet's internal energy $e$. The solution to this spray equation is obtained by means of a Monte Carlo method. The Lagrangian equations for trajectory, velocity, size and temperature are solved. The droplet dispersion in a turbulent flow is calculated according to the Gaussian distribution (Gosman & Ioannides 1983).

The characteristics of the spray to be described in the inlet boundary condition include mean droplet size, droplet size distribution, droplet number density and cone angle. The atomiser and fuel spray cone are small and the rumble frequency is much lower than the spray cone break-up frequency. We therefore assume that the fuel spray responds in a quasi-steady way to the flow through the atomiser. From results obtained over a wide test range, Lefebvre drew the general conclusion that liquid viscosity has an effect that is quite separate and independent from that of surface tension (Lefebvre 1989). He suggested the following equation to describe the steady-state airblast atomization droplet diameter

$$D_{32} = 10^{-3} \left(1 + \frac{m_f}{m_a}\right)^{0.5} \times \left[\left(\frac{\sigma P_e}{\rho_a U_a}\right)^{0.5} + 0.06 \left(\frac{\mu}{\sigma \rho_a}\right)^{0.429}\right]$$

(2)

where $m_f$ is the mass flux rate, $\rho$ the density, $\mu$ the dynamic viscosity and $\sigma$ the surface tension. The suffixes $a$ and $f$ denote the air and liquid in the atomiser, respectively. Once the Sauter mean diameter $D_{32}$ is obtained, the droplet size distribution can be calculated from the non-dimensional expression

$$\frac{dV}{V} = 13.5\left(D_{32} - D\right)^3 \exp(-3(D_{32} - D)) d(D_{32})$$

(3)

where $dV/V$ is the volume fraction occupied by droplets within the size range $D$ to $D + dD$. We use equations (2) and (3) to relate the fuel droplet size to the instantaneous air and fuel flow for low frequency fluctuations. The fuel spray cone angle is described by random numbers which are generated according to a gamma distribution. Reference (Zhu 1996) may be consulted for a detailed description of how the calculation was implemented.

3. NUMERICAL RESULTS

The study has been conducted in the idealised 2D section of an axisymmetric annular combustor shown
in Figure 1, and summarised in Table 1. The boundary conditions used in the numerical computation were specified to be representative of an aero-engine at idle. The inlet air stagnation temperature is 366K, and the stagnation pressure is 225kPa. The exit static pressure is 195kPa. This leads to an overall air mass flow rate of 5.7 kg/s. Fuel is equally divided between inlets 1 and 2 and injected at a mass flow rate of 0.06 kg/s, speed 70 m/s. The temperature of this liquid fuel is 288K. The calculation started from a cold flow, and reached the steady-state solution with artificial relaxation. The time step was selected to ensure that the acoustic waves travel less than one cell per time step.

Table 1: The inlet geometry used in the numerical simulations.

<table>
<thead>
<tr>
<th>inlet No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
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<td>0.0</td>
<td>0.0</td>
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<td>135.0</td>
<td>38.0</td>
<td>77.0</td>
<td>135.0</td>
</tr>
<tr>
<td>θ deg.</td>
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<td>90.0</td>
<td>90.0</td>
<td>90.0</td>
<td>58.12</td>
<td>51.92</td>
<td>77.859</td>
<td>62.745</td>
<td>57.76</td>
<td>71.91</td>
</tr>
<tr>
<td>r mm</td>
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<td>216.0</td>
<td>270.0</td>
<td>180.0</td>
<td>270.0</td>
<td>270.0</td>
<td>270.0</td>
<td>180.0</td>
<td>180.0</td>
<td>180.0</td>
</tr>
</tbody>
</table>

Figure 1: Schematic diagram of the geometry.

A contour plot of the steady state mean temperature distribution is shown in Figure 2. It is interesting to note that, even for the basic unforced calculation, the flow properties exhibit a small amplitude, high frequency oscillation of about 1000Hz. We believe that this oscillation is physical and related to the propagation of acoustic waves in the combustor. The frequency varies with the combustor length. Moreover, if the average combustor temperature, and hence speed of sound is altered, due to say a change in the inlet mass flow rate or spray core angle, the frequency of oscillation is modified correspondingly. The engine exhibits a similar high frequency oscillation. However, the 'rumble' unsteadiness is at low frequency and is thought to involve coupling between the combustor and its air and fuel supplies. We have investigated the response to these inlet perturbations through a series of forced calculations. The frequencies of interest (60-120Hz) are an order of magnitude less than the self-excited oscillation (1000Hz) and there is no difficulty in distinguishing between effects at these different frequencies.

Figure 2: Contour plot the mean temperature distribution chamber at idle conditions. The black line indicates the mean position of the stoichiometric curve.

An advantage of a numerical solution is that we are able to vary parameters in the inlet air and fuel independently and so investigate their relative importance. In order to investigate the transfer function between the unsteady flow in the combustor and inlet fluctuations at a single frequency, sinusoidal forced boundary conditions were implemented. Although all the variables are coupled to each other in a real situation, it will be valuable to decouple them artificially, so that the effects of different factors can be identified separately. In this study, forced sinusoidal oscillations in the gaseous total pressure and in the fuel mass flow rate at the inlets are implemented. In this paper, we present sample results in which the frequency of forced oscillation is 65Hz, and the fluctuation amplitude is equivalent to 20 percent of the mean value of the mass flow rate. Two frequencies are evident in the time history of the flow parameters in the combustor, the forced oscillations at 65Hz and the self-excited perturbations near 1000Hz.

Case 1: Forced oscillation in supplied fuel

First, results for forced oscillations in the injected fuel mass flow rate will be discussed. To separate the influence of changes in the droplet size from the direct effects of fuel flow rate, in the results presented as Case 1, the droplet size distribution is artificially kept fixed.
The effects of coupled changes in droplet size and fuel flow rate as described by Eq. (2) are considered as Case 3. The inlet fuel mass flow rate is described by the relationship

\[ \frac{m_f(t)}{m_f} = 1 + 0.2\cos(400t). \]  

(4)

Figure 3 gives time traces of mixture fraction and heat release rate at two sample points. As shown in Fig. 1, the point A is within the combustion zone, and the point B near the tip of the flame. Both the low frequency and high frequency oscillations are seen clearly. An increase in the rate of fuel injection leads to an increase in vapor mixture fraction with a small convective time delay. The convection times from the atomiser to points A and B are larger than the droplet life-times. Therefore the droplet evaporation time does not significantly affect the time delays. The fractional amplitude of perturbation in mixture fraction at points A and B is about 20 percent just as in the inlet disturbance in Eq. (4).

The influence of mixture fraction on the rate of heat release depends on whether the mixture is weaker or stronger than stoichiometric value. For given the variance of mixture fraction, which is mainly affected by turbulence, the laminar flamelet library gives a maximum rate of heat release when the mixture fraction is equal to its stoichiometric value (an equivalence ratio of unity) of 0.0682 for kerosene. When well below the stoichiometric value, an increase in mixture fraction increases the rate of heat release, whereas above it the rate of heat release is reduced as the mixture becomes richer. This observation explains the phase relationship between the rate of heat release and unsteady fuel addition in Fig. 3. As shown in Fig. 3(b), the mixture fraction is larger than the stoichiometric value at point A. An increase in the fuel flow rate further increases the mixture fraction and the rate of heat release/unit volume decreases accordingly (see Fig. 3(c)). On the other hand, for the case in Fig. 3(d) at the point B, the average mixture fraction is smaller than the stoichiometric value, so that the rate of heat release increases in phase with an increase in mixture fraction.

The unsteady calculations lead to a large amount of flow data. This data can be managed by integration over the combustor cross-section to investigate the relationship between \( q(x, t) \), the rate of heat release/unit length of combustor and the inlet fuel and air flows. The phase relationship between this integrated rate of heat release and the pressure perturbation is crucial to understanding the driving mechanism to instability (Rayleigh 1896). Colour plate 1 illustrates the change in heat release rate due to the oscillation in inlet fuel. In this plot, we show the heat release rate/unit length, \( q(x, t) \), as a function of \( x \) and \( t \). The rate of heat release at different \( x \) are displaced vertically as indicated by \( z \)-values on the ordinate. The slope of the bright lines correspond to the mean convection speed of the heated fluid. The trends we saw locally at points A and B are also evident in the area-integrated rate of heat release. Downstream of \( x \approx 0.08m \), the mixture fraction is weaker than the stoichiometric value. A low frequency increase in fuel flow rate therefore increases the rate of heat release in this downstream region, after a convective time delay. The mixture is rich upstream of \( x \approx 0.08m \). Here it is a reduction in fuel input that enhances the rate of combustion.

Colour plate 1 shows the influence both of the low frequency forced oscillations and the high frequency self-excited perturbations. The low frequency response can be highlighted by evaluating the temporal Fourier transforms of \( q(x, t) \) and \( m_f(t) \). The magnitude and phase of the transfer function \( \frac{q(x)}{m_f(t)} \) at the forcing frequency are plotted in Fig. 4. The two regions with distinct and different forms of flame response are now evident. The gradual decrease in phase throughout the region \( x > 0.08m \) indicates a convective time delay. Extending this part of the phase curve back to \( x = 0 \), we find that it passes through the origin, confirmation that throughout the downstream region, where the mixture ratio is weaker than stoichiometric, the rate of heat release is in phase with the mixture fraction, which lags the inlet fuel variation by a convective time delay. Near the tip of flame, the rate of heat release is insensitive to fluctuations in mixture fraction and this accounts for the minimum in \( q \) near \( x \approx 0.08m \). Two local maxima in \( q \) near 0.035m and 0.1m can be interpreted in terms of the rich and weak locations where the rate of heat release is most sensitive to changes in mixture fraction. As already noted, in the predominately rich combustion zone \( x < 0.08m \) the unsteady rate of heat release is approximately of phase with the rate of fuel supplied.

**Case 2: Forced oscillation in supplied air**

In this part, the results of the forced oscillations in the total pressure of the air inlets 1 and 2 will be discussed. As in previous work, in order to separate the effects of droplet size distribution from the forced oscillation, the droplet size was artificially fixed as its mean value. Figure 5 shows typical time traces of mixture fraction and heat release rate at sample points A and B. Comparing with Figs. 3, we can see that whereas an increase of fuel mass flow rate leads directly to an increase of mixture fraction, an increase in the total pressure of air supply has a more complicated effect. Since amplitude of the perturbation in both cases is 20 percent, the variation of mixture fraction in both cases is of the same order. It can be seen that both low frequency and high frequency oscillations are shown
fraction. Near the tip of the flame, the rate of heat release is most sensitive to changes in mixture fraction. Again there are two local maxima where the rate of heat release due to the oscillation of total pressure in the primary zone. Transport properties are also changed. The numerical calculations show that the combined effect is that the mixture fraction lags the inlet air acceleration by a convection time delay. Further calculations are required to clarify the dominant physical process. As shown in Fig. 5(b), the mixture fraction is above the stoichiometric value at point A, and so as we have already noted, the rate of heat release is 180° out of phase with mixture fraction, leading to the heat release shown in Fig. 5(c). On the contrary, as shown in Fig. 5(d), at the point B, which is located around the periphery of the flame, the average value of mixture fraction is smaller than the stoichiometric value, so the heat release rate will be in phase with the mixture fraction.

Colour plate 2 illustrates the change in heat release rate due to the oscillation of total air pressure in the inlets 1 and 2. Not surprisingly, through comparison with plate 1, we can see that the different time delay is clearly demonstrated. It is because, as we discussed, the forced oscillation of gas and liquid fuel supplies causes different effects on the variation of heat release rate. In addition, it can be seen that the oscillation of the physical location of the combustion zone in the axial direction is the larger than in plate 1. This can be explained by noting that, in this case, the oscillation of the total pressure of gas supply also induces a variation of the inlet gas velocity at the same frequency. As a consequence, the flow speed will oscillate and the length of combustion zone will vary correspondingly. Plates 2 and 1 also show that, in the primary zone (close to the atomiser), the variation of heat release rate in this case is more apparent due to the strength of recirculation vortices, which are affected by the inlet air velocity oscillation. The movement of the vortex and the variation of vortex strength will alter the distribution of fuel vapor in the combustion chamber. Moreover, we can see that the bright lines in plate 2 are blurred and not as distinct as in plate 1. This is because the oscillation in the total pressure of air supply leads to an addition disturbance downstream, which enhances the fuel/air mixing process.

The transfer function between heat release and forced oscillation in the total pressure of air supply at the low frequency is shown in Fig. 6. It is found that the variation in the amplitude with axial position is similar in form to that due to fuel forcing in Fig. 4. Again there are two local maxima where the rate of heat release is most sensitive to changes in mixture fraction. Near the tip of the flame, the rate of heat release is insensitive to fluctuations in mixture fraction and this accounts for the minimum. In the downstream region, the phase of the heat release slowly decreases in the axial direction, indicating an increasing time delay. The rapid change in phase near x = 0.08m is because of the different phase relationship between rate of heat release and mixture fraction on the two sides of the stoichiometric curve. Similar conclusions can also be drawn from Fig. 8.

Case 3: Effects of droplet size distribution

In reality, forced perturbations in the fuel mass flow rate and the total pressure of the gas supplies not only alter the air/fuel mass flow rate, but also the droplet size distribution. In the above calculations, in order to separate the effects of the flow rate and droplet size distribution, droplet size was artificially kept fixed. In the following, coupling between droplet size and the forced oscillation at the inlet boundary was implemented according to Eq. (2).

The effects of the unsteady oscillation of the fuel mass flow rate are shown in Figs. 7,8 and colour plate 3. The variation of the Sauter mean diameter of droplets due to the total air pressure oscillation at inlets 1 and 2 is shown in Fig. 7(a) with a dash–dotted line. The time traces of the mixture fraction and heat release rate at the sample points A and B, which are shown in Fig. 7, show no apparent difference from those in Fig. 5. Thus, the effects of droplet size variance due to the fuel mass flow rate oscillation on the unsteady combustion are small. Only a small change is seen in the transfer function between the heat release rate and the oscillated boundary condition, which shown in Fig. 8(a). A small step is introduced in the first peak in the amplitude relationship. One possible explanation is that the combustion region is slightly shifted due to the droplet size perturbation.

The effects of unsteady oscillation in the total pressure of air supply on the droplet size distribution has also been investigated. From Eq.(2), it is shown that an increase of the air mass flow rate and gas velocity lead to a decrease of droplet size due to the larger momentum difference between the two phases. In this situation, because of the so-called the $d^2$–law, the evaporation time of small size droplets is shorter than that of droplets with the mean size. From Fig. 5(a), it can be seen that the amplitude of droplet size oscillation is much larger than in Fig. 7(a). Comparing the time traces of the mixture fraction and heat release rate in Fig. 5 with Fig. 9, it can be seen that the variation of droplet size changes the phase relationship between mixture fraction and inlet air velocity. The smaller droplets evaporate more quickly, moving the combustion zone closer to the atomiser. All injected droplets are therefore heated more quickly. In this situation, the heating process will control the life-time of droplets. Comparing the time traces of the mixture...
fraction at point B (see Figs. 5(d) and 9(d)), it can be seen that the amplitude of oscillation in Case 2 is stronger than in this case. This means that more fuel vapor is consumed more quickly in combustion in this case. Therefore, the droplets with small size play an important role in the combustion process, and their variation has a strong influence on the combustion instability.

From colour plate 4(b), the shift of the combustion zone is clearly demonstrated. This is related to the variation of droplet size shown in plate 4(a). It is seen that the combustion zone moved very close to the exit of the atomiser after a short time delay after the smaller droplets appear. The transfer function between heat release rate and forced boundary condition is shown in Fig. 10. In comparison with Fig. 6, although the general tendency is similar, it is seen that there is an additional local maximum in the magnitude and one more step in the phase relationship of oscillation. This is because the oscillation of droplet size and gaseous total pressure at the atomiser cause a variation not only in the position of the combustion zone, but also in its length. The smaller droplets cause the front of the combustion zone to move to the atomiser periodically. We can see that the positions of the steps in phase in Fig. 10(a) correspond to the edges of the combustion zone in the limiting situations in colour plate 4. Consistently, as shown in Fig. 10(b), there are three jumps in the phase relationship. They represent the phase shifts due to the oscillation of the combustion region in the physical space.

4. CONCLUSIONS

The oscillating behaviour of spray combustion has been investigated through time-dependent calculations of the combustion process. The coupling between forced oscillations in inlet fuel and air, atomization and chemical kinetic processes greatly influences the dynamical flame behaviour. Due to the complexity of this interaction and to identify the main influences on engine rumble, several test cases were calculated. Harmonic oscillation of inlet fuel and air flows have been considered and the resulting unsteady combustion calculated. The effects of droplet size distribution due to these perturbations have also been investigated.

Some conclusions can be drawn from the current study. The change of local heat release rate is influenced by the local variation of the mixture fraction, which is caused by the forced oscillation at the atomiser inlets. The maximum rate of heat release occurs when the mixture fraction is closest to stoichiometric. The phase of the heat release therefore changes across the stoichiometric line. Despite the fact that perturbations in both air and fuel flow rates through the atomiser affect the heat release rate, for the same percentage change based on the mass flow rate, variations in the air-flow rates through the atomiser have been identified as having a stronger effect on the instantaneous rate of heat release. Modulations in this air-flow alter the inlet fuel droplet size distribution and hence the local ratio of fuel vapor to oxidiser throughout the combustor, thereby changing the location and time delay to combustion.

Although encouraging results are obtained in the present study, it represents only the first step in the full investigation. Further work will concentrate on (a) development of simple models of the air and fuel supply system so that changes in inlet fuel and air can be coupled to changes in the combustion for simulation of self-excited oscillations, (b) incorporation of the transfer function between \( q(z,t) \), and injected air and fuel flow as functions of frequency, obtained by Fourier transformation of data like that in Fig. 10. This will used into one-dimensional linear stability analysis (Dowling 1995) to interpret the numerical results, to give a better understanding of the feedback mechanism and to enable extrapolation from the axisymmetric geometry to three dimensions.

Acknowledgement

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REFERENCES


Figure 3: Sinusoidal changes of the fuel mass flow rate in the atomiser, which are shown in (a), lead to the oscillations in the mixture fraction and heat release rate at point A and B. In this case, the droplet size was artificially fixed as its mean value. The location of sample points A and B are shown in Fig. 1.

Figure 4: The transfer function between the heat release rate per unit length and the normalized sinusoidal changes of the fuel mass flow rate in the atomiser.

Figure 5: Sinusoidal changes of the total pressure in the air inlets, which are shown in (a), lead to the oscillations in the mixture fraction and heat release rate at point A and B. In this case, the droplet size was artificially fixed as its mean value. The location of sample points A and B are shown in Fig. 1.

Figure 6: The transfer function between the heat release rate per unit length and the normalized sinusoidal changes of the total pressure in the air inlets.
Figure 7: Sinusoidal changes of the fuel mass flow rate in the atomiser, which are shown in (a), lead to the oscillations in the mixture fraction and heat release rate at point A and B. In this case, the droplet size was coupled with the oscillations according to Eq. (2). The location of sample points A and B are shown in Fig. 1.

Figure 8: The transfer function between the heat release rate per unit length and the normalized sinusoidal changes of the fuel mass flow rate in the atomiser.

Figure 9: Sinusoidal changes of the total pressure in the air inlets, which shown in (a), lead to the oscillations in the mixture fraction and heat release rate at point A and B. In this case, the droplet size was coupled with the oscillations according to Eq. (2). The location of sample points A and B are shown in Fig. 1.

Figure 10: The transfer function between the heat release rate per unit length and the normalized sinusoidal changes of the total pressure in the air inlets.
Plate 1: The variation of the area-averaged rate of heat release / unit length due to the sinusoidal changes of the fuel mass flow rate in the atomiser, which are shown in (a). In this case, the droplet size was artificially fixed as its mean value.

Plate 2: The variation of the area-averaged rate of heat release / unit length due to the sinusoidal changes of the total pressure in the air inlets, which are shown in (a). In this case, the droplet size was artificially fixed as its mean value.

Plate 3: The variation of the area-averaged rate of heat release / unit length due to the sinusoidal changes of the fuel mass flow rate in the atomiser, which are shown in (a). In this case, the droplet size was coupled with the oscillations according to Eq. (2).

Plate 4: The variation of the area-averaged rate of heat release / unit length due to the sinusoidal changes of the total pressure in the air inlets, which are shown in (a). In this case, the droplet size was coupled with the oscillations according to Eq. (2).