Laminar Smoke Point Based Subgrid Soot Radiation Modeling Applied to LES of Buoyant Turbulent Diffusion Flames

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Abstract. Large eddy simulations (LES) of gaseous buoyant turbulent flames have been conducted with the application of a flamelet based soot-radiation model. The subgrid model applies a turbulent eddy description of soot formation, oxidation and radiation and is based on the laminar smoke point concept. Two parameters, a local turbulent strain rate and prior enthalpy loss/gain fraction influence the soot formation and radiation. Radiation heat transfer is simulated by solving the finite volume discretized form of the radiative transfer equation (RTE) with the subgrid soot-radiation model implemented. The radiant heating of surfaces in close proximity of the flames is computed and predicted heat fluxes and surface temperatures are compared against experimental data. Fire growth in a rack storage arrangement is simulated with the application of a pyrolysis model. Computed heat release rate (HRR) is compared against experimental data.

1. Introduction

Thermal radiation, primarily due to soot, is responsible for fire spread in industrial scale rack storage scenarios. Flue spaces in rack storage setups are typically in the range of 0.15-0.6 m. Radiation in these spaces tends to be in the optically thin regime for moderately sooting pyrolysates from solid fuel surfaces. Accurate spatial and temporal prediction of radiant emission from the turbulent flame sheet therefore becomes the primary factor in predicting fire growth. This article presents results from CFD studies of (1) radiant heat transfer from a 60 kW propane fire to inert parallel panels [1], and (2) fire growth in an industrial-scale rack storage setup [2].

Studies by Markstein [3] have established that radiation from turbulent buoyant diffusion flames correlates with the fuel’s laminar smoke point height. Soot formation, oxidation and radiation all scale with the laminar smoke point. Using this concept, a model for soot radiation was developed by Lautenberger et al [4]. CFD simulations with this model showed good agreement of computed results with laminar diffusion flame experimental data. In turbulent flames the Kolmogorov micro-time scale governs the processes of soot formation, oxidation and radiation. This Kolmogorov mixing occurring in small reaction volumes must be described.

A subgrid soot-radiation model based on the laminar smoke point model [5] is applied to compute radiant emission from buoyant turbulent diffusion flames. The subgrid soot-radiation model includes the effect of turbulence on the soot generation and oxidation rates. The model
requires the application of a characteristic turbulent micro-scale strain rate, $a_0$, and prior radiative loss/gain fraction, $H_p$, as parameters that determine soot release and radiation from the fire. The model has been coupled with an LES solver, FireFOAM [6] and tested against experimental data on radiation from optically thin buoyant turbulent diffusion flames from a range of fuel and fire sizes [5]. In the present study, the model provides an isotropic emission term from the turbulent flame sheet that is introduced in the radiative transfer equation (RTE) and consequently, radiant heat transfer from a propane fire to inert parallel panels and rack storage fire growth are simulated.

2. Numerical Model
In LES of diffusion flames the flame sheet is not resolved by the computational mesh (Fig. 1(a)). It is assumed that, in the grid cell where the flame sheet is present, the entrained air and fuel mix to a molecular scale where reaction takes place. This mixing is the result of dissipating eddies/vortex-structures (Fig. 1(b)). This assumes the presence of sheet-like laminar flamelets strained by the local turbulent eddies (Fig. 1(c)).

2.1. Flamelet Model
The flamelet model [5] comprises of a 1-D transient packet of fluid being strained by a constant strain rate normal to its axis. The model applies the laminar smoke point concept [4] to describe soot formation, oxidation and radiation. Soot is formed and oxidized in the immediate flame with the residual soot being released. The local mixture composition and prior energy loss by radiation are tracked by solving the conservation of mixture fraction and total enthalpy equations. The “radiatively perturbed laminar flamelet” concept [7] is used in which the temperature is computed as a function of mixture fraction and total enthalpy. The emitted radiation is assumed to be a function of fuel type (in particular, its laminar smoke point height $\ell_s$), a prior local heat loss fraction (radiant and convective, $H_p$) and an imposed strain rate at the smallest dissipation scale ($a_0$). The characteristic strain rate at the dissipation scale is estimated as the inverse of the Kolmogorov micro-time, $a_0 = C_\eta/\tau_\kappa$, where $C_\eta$ is a constant. A single transient flamelet is assumed to be of initial thickness $\eta$ (the Kolmogorov length scale). This initial thickness is shown in Fig. 1(c).

![Figure 1](https://example.com/figure1.png)

Figure 1. Subgrid model for soot-radiation: (a) turbulent diffusion flame sheet in a LES grid cell, (b) local straining of flame sheet by vortices, and (c) one-dimensional transient flamelet representation.

2.1.1. Flamelet Equations
The original three-dimensional conservation equations follow from the approach of Lautenberger et al [4]. The soot conservation equation is

$$\frac{\partial \rho_g \phi_s}{\partial t} + \nabla \cdot (\rho_g \phi_s \mathbf{u}) = \nabla \cdot \left( 0.55 \frac{Y_s \mu}{T} \nabla T \right) + \dot{\omega}_s'''. \tag{1}$$
Here, we have redefined the soot scalar variable as \( \phi_s = Y_s / (1 - Y_s) \), where \( Y_s \) is the soot mass fraction. The source term in the soot conservation equation, \( \dot{\omega}_s^{m'} \), is computed as a sum of the soot formation and oxidation rates: 
\[ \dot{\omega}_s^{m'} = \dot{\omega}_{sf}^{m'} + \dot{\omega}_{so}^{m'} \]. Lautenberger et al [4] used experimental measurements to guide the selection of general shapes of the soot formation and oxidation functions. The soot formation rate, \( \dot{\omega}_{sf}^{m'} \), was assumed to be a parabolic function of mixture fraction and temperature and is evaluated as a product of \( f_{sf}''(Z) \), and \( g_{sf}(T) \):
\[ \dot{\omega}_{sf}^{m'} = f_{sf}''(Z) g_{sf}(T) \].

**Figure 2.** Soot formation and oxidation functions: (a) mixture fraction dependent soot formation function, \( f_{sf}''(Z) \) [kg/m\(^3\)-s], and oxidation function, \( f_{so}''(Z) \) [kg/m\(^3\)-s], and (b) temperature dependent soot formation function, \( g_{sf}(T) \) [\( \cdot \)], and oxidation function, \( g_{so}(T) \) [\( \cdot \)].

Peak soot formation rates occur at mixture fraction values between 0.1 and 0.15 and over a temperature range of 1500-1600 K. The peak soot formation rate is assumed to be inversely proportional to the laminar smoke point height, \( \dot{\omega}_{sf,P}^{m'} \propto 1/\ell_s \). Soot is oxidized on the oxidizer side of the flame, controlled by the diffusion of oxygen, as long as the temperature is above 1400 K. The soot oxidation rate, \( \dot{\omega}_{so}^{m'} \), is also evaluated as a product of a function of mixture fraction, \( f_{so}''(Z) \), and temperature, \( g_{so}(T) \): 
\[ \dot{\omega}_{so}^{m'} = f_{so}''(Z) g_{so}(T) \]. The mixture fraction dependent soot formation and oxidation functions, \( f_{sf}''(Z) \) and \( f_{so}''(Z) \), respectively are shown in Fig. 2(a). The temperature dependent functions, \( g_{sf}(T) \) and \( g_{so}(T) \) are shown in Fig. 2(b).

The total enthalpy equation is solved with a radiant emission sink term included,
\[
\frac{\partial \rho g h}{\partial t} + \nabla \cdot (\rho g \mathbf{u}) = \nabla \cdot (\rho g D \nabla h) + S_{sf} + S_{so} - \dot{q}_r^{m'},
\]  
where, \( h = h_T + \phi_s h_{T,s} \). Here, \( h_T \) is the gas-phase total enthalpy (sensible + chemical) and \( \phi_s h_{T,s} \) is the soot enthalpy [4]. \( S_{sf} = \dot{\omega}_{sf}' (h_{T,F} - h_{T,s}) \) is the change in sensible enthalpy due to soot formation, and \( S_{so} = \frac{\dot{\omega}_{so}^{m'}}{M_F} \left[ M_F h_{T,s} + \left( x + \frac{y}{4} \right) M_O h_{T,O_2} - x M_{CO_2} h_{T,CO_2} - \frac{y}{2} M_{H_2O} h_{T,H_2O} \right] \) is the release of energy per unit volume due to soot oxidation for an arbitrary hydrocarbon fuel, \( C_x H_y \). The conservation of mixture fraction equation is
\[
\frac{\partial \rho g Z}{\partial t} + \nabla \cdot (\rho g \mathbf{u} Z) = \nabla \cdot (\rho g D \nabla Z) + \dot{\omega}_Z^{m'},
\]  
where, the source term, \( \dot{\omega}_Z^{m'} \) includes the effect of soot formation or oxidation on the mixture fraction (see [4] for details). The Howarth-Dorodnitzyn transformation [8] is applied to eliminate
the effects of variable density by replacing the coordinates \( y, t \) by the dimensionless coordinates \( \xi, \tau \), where \( \xi \) is a non-dimensional space coordinate, \( \xi = \left( \frac{a_{0}}{\rho_{2} D_{\infty}} \right)^{1/2} \int_{0}^{t} \rho_{y} (y', t) dy' \) and, \( \tau \) is a non-dimensional time, \( \tau = \int_{0}^{t} a(t') dt' \). Here, \( \rho_{y} \) is the gas-phase density, \( \rho_{\infty} = \rho_{y} (\infty, t) \). With the approximation \( \rho_{2} D = \rho_{2} D_{\infty} \) applied, we transform the conservation equations to their non-dimensional forms,

\[
\frac{\partial \phi_{s}}{\partial \tau} - \xi \frac{\partial \phi_{s}}{\partial \xi} = S_{h} + \frac{\dot{\omega}_{s}'''}{\rho_{g} a_{0}},
\]

\[
\frac{\partial h}{\partial \tau} - \xi \frac{\partial h}{\partial \xi} = \frac{\partial^{2} h}{\partial \xi^{2}} + \frac{S_{f} + S_{so} - \dot{q}_{r}'''}{\rho_{g} a_{0}},
\]

\[
\frac{\partial Z}{\partial \tau} - \xi \frac{\partial Z}{\partial \xi} = \frac{\partial^{2} Z}{\partial \xi^{2}} + \frac{\dot{\omega}_{Z}'''}{\rho_{g} a_{0}}.
\]  

Here, \( S_{h} \) is the soot thermophoresis term which is computed as \( S_{h} = 0.55 S_{c} \frac{\partial}{\partial \xi} \left[ \frac{\phi_{s}}{(1+\phi_{s})T} \right] \).

### 2.2. Flamelet Radiant Emission

Thermal radiation is assumed to be optically thin and to occur primarily due to soot forming and radiating. The soot absorption coefficient is computed following Tien et al. \[9\], \( \kappa_{s} \approx 1226 f_{c} T \ [m^{-1}] \), where \( f_{c} \) is the soot volume fraction and \( T \) is the temperature. The radiation source term is computed as a function of the soot volume fraction and temperature, \( \dot{q}_{r}''' = 4 \kappa_{s} \sigma T^{4} \). The non-dimensional initial thickness of the 1-D slab/flamelet (see Fig. 1(c)) is estimated as a function of \( \eta \sim (\nu^{3}/\varepsilon)^{1/4} \) and the characteristic strain rate \( a_{0} \) as

\[
\Gamma = \left( \frac{a_{0}}{\rho_{2} D_{\infty}} \right)^{1/2} \int_{0}^{\eta} \rho dy \sim \left( \frac{\rho_{F}}{\rho_{\infty}} \right) S_{c}^{1/2},
\]

where, \( S_{c} \) is the Schmidt number, \( \rho_{F} \) is the density of fuel, \( \rho_{\infty} \) is the ambient density, \( \varepsilon \) is the dissipation rate and \( \nu \) is the kinematic viscosity. The non-dimensional initial width for a propane flamelet is \( \Gamma = 1.2774 \).

Figure 3 shows the influence of the turbulent micro-time strain rate, \( a_{0} \) and prior enthalpy loss fraction, \( H_{P} \) on the flamelet temperature and soot mass fraction. In Fig. 3(a) due to the strain rate being low \( (a_{0} = 5 \text{ s}^{-1}) \) and no prior loss \( (H_{p} = 0.0) \), large amount of soot forms, oxidizes and radiates, lowering the flamelet temperature to below 1500 K and the flamelet releases unoxidized soot. In Fig. 3(b) for \( a_{0} = 5 \text{ s}^{-1} \) and \( H_{p} = 0.3 \), the flamelet temperature distribution at \( \tau = 0 \) shows lower temperatures due to the 30% lowering of sensible enthalpy uniformly across the flamelet. This reduction in initial temperatures results in lower amounts of soot generation and consequently lower radiation from the flamelet. No soot is released from the flamelet for this condition. Increasing \( a_{0} \) to 30 \text{ s}^{-1} \) causes lower radiant emission from the flamelet and no soot release. This is true both for no prior enthalpy loss (Fig. 3(c)) and with \( H_{p} = 0.3 \) (Fig. 3(d)). The computed radiant energy fraction, \( \Psi_{r} \) and soot released per unit combustion energy, \( \Psi_{s} \) are dependent on \( a_{0} \) and \( H_{p} \). In general, lower values of \( a_{0} \) and \( H_{p} \) result in higher radiant emission and greater soot release from the flamelet. A lookup-table is generated for \( \Psi_{r} \) as a function of \( a_{0} \) and \( H_{p} \). In the LES of turbulent flames, the local radiant emission from the turbulent flame sheet in a grid cell is computed as \( \dot{q}_{r}''' = \Psi_{r} Q''' \), where \( Q''' \) is the turbulent volumetric heat release rate. The soot source term is computed as \( \dot{\omega}_{s}''' = \Psi_{s} Q''' \).
Figure 3. Contour plots of flamelet temperature, $T$, and soot mass fraction, $Y_s$: (a) $a_0 = 5 \, \text{s}^{-1}$, $H_p = 0.0$, (b) $a_0 = 5 \, \text{s}^{-1}$, $H_p = 0.3$, (c) $a_0 = 30 \, \text{s}^{-1}$, $H_p = 0.0$, (d) $a_0 = 30 \, \text{s}^{-1}$, $H_p = 0.3$.

2.3. LES Solver

The current set of simulations has been carried out with a large eddy simulation (LES) code, FireFOAM [6], which is a pressure based segregated solver, solving the Favre filtered fully compressible Navier-Stokes equations. Details of FireFOAM equations can be found in [10]. The turbulent sub-grid scale stress is modeled by the eddy viscosity concept using the one-equation model [11], which solves a sub-grid scale (SGS) kinetic energy equation. The dissipation rate is estimated by the model as $\varepsilon = c_s \Delta^{-1} k^{3/2}$. The turbulent micro-time strain rate required by the flamelet model is computed as the inverse of the Kolmogorov time scale as $a_0 = C_n (\varepsilon / \nu)^{1/2}$. The turbulent combustion model adopts an infinitely-fast chemistry assumption. This mixed-is-burnt model removes the requirement of chemical kinetics, and reasonably represents the physics in well-ventilated fires. The fuel consumption rate is estimated from the eddy dissipation model as $\dot{\omega}_{Y_F}'' \sim \dot{\varepsilon}_{\text{mix}} \min \left( \frac{\bar{Y}_{F, F}}{\bar{Y}_{s, s}} \right)$, where $\tau_{\text{mix}}$ is a turbulent macro-mixing time. The sensible enthalpy equation is solved with a volumetric heat release rate source term $\dot{Q}'' = \Delta H \dot{\omega}_{Y_F}''$ and a radiation sink term $\dot{q}''_r$. The prior heat (enthalpy) loss is modeled following Brookes and Moss [12] as $H_p = \left( \bar{h}_T - \bar{h}_{F, \infty}^0 \right) / \left( \bar{h}^0 - \bar{h}_{F, \infty}^0 \right)$ where $\bar{h}^0$ is the chemical enthalpy and $\bar{h}_T$ is
the total enthalpy (sensible + chemical). A soot transport equation is solved with the source term \( \dot{\omega} Y_s' \) included. The source term provides the released unoxidized soot from the flame sheet. No further oxidation of the transported soot takes place.

A gray media assumption is made for computing radiation heat transfer. This assumption is justified since soot is primarily responsible for radiation in diffusion flames of propane. It is also assumed that radiation in the parallel panel configuration is in the optically thin limit. Radiant emission due to soot from the flame sheet is the only source of radiation. Products of combustion and the soot released from the flame-sheet do not participate in radiation heat transfer. However, the panels, ground, sand burner and the domain boundaries emit and absorb radiation. The flame-sheet volumetric radiant emission, \( \dot{q}_r'''' \), is included as an isotropic emission term in the RTE [10],

\[
\frac{dT}{ds} = \kappa I_b - \kappa I = \dot{q}_r'''' \frac{4}{4\pi}.
\]

In the above equation, we assume that only the flame emits radiation isotropically, i.e. \( \kappa I_b \approx \dot{q}_r'''' \frac{4}{4\pi} \).

In the simulations, the emitted radiation from the flame sheet is incident on the solid surfaces. It is reflected, absorbed and re-radiated from these surfaces, however this emitted radiation is not absorbed by the flame or the combustion products, i.e. \( \kappa I = 0 \). The net radiative heat flux on solid-fuel surfaces is obtained by calculating the difference between the incident radiative flux and the emitted and re-radiated heat flux components. For opaque materials like corrugated-cardboard (rack storage) and Marinite-I (inert parallel panel), there is no transmitted component of incident radiation. The surface emissivity, which is a function of the char-fraction, is provided by the pyrolysis model. The surface temperature for the emission component is also computed by the pyrolysis model. The net radiative heat flux along with the convective flux is used as input to the pyrolysis model. A finite-volume method [13] is applied for the solution of the RTE using the existing computational mesh.

3. Results and Discussion

Results are presented from two studies: radiant heat transfer from a propane fire onto inert parallel panels and fire spread over corrugated commodities in a rack storage arrangement. The standard parallel panel configuration used by FM Global [1] for characterizing industrial clean room materials [14] is considered. The rack storage fire growth simulations follow fire growth studies conducted earlier for corrugated parallel panels [15]. In addition to vertical fire growth encountered in the parallel panel setup, in rack storage scenarios both vertical and lateral fire spread occurs. Flue spaces in rack storage arrangements vary between 0.15-0.6 m and complex geometrical arrangements of boxes, obstructions and large heat release rates (HRR) make simulation of fire growth in rack storage scenarios even more challenging.

3.1. Flame Heat Transfer to Inert Parallel Panels

Flame heat transfer, primarily radiative, from a 60 kW propane diffusion flame to adjacent walls in a parallel panel configuration [14] is simulated. In this configuration, the flame is anchored on a 0.3 m \( \times \) 0.6 m sand burner placed between two inert vertical panels. The panels are made of a fire-resistant material, Marinite-I. Figure 4 shows a section of the computational domain. The sand burner is elevated by 0.3 m from the ground and the panels are each 2.4 m tall and 0.025 m in thickness with the back boundary insulated. Heat flux (using Schmidt-Boelter gauges) and surface temperature (using thermocouples) are recorded on the panels at vertical centerline and offset locations (as shown in Fig. 4). Six locations on one of the panels are shown in the figure – two centerline locations at 0.3 m and 0.9 m from the bottom and four offset locations at 0.6 m and 1.2 m from the bottom (\( \pm 0.15 \) m offset from the centerline). The computational domain is 1.2 m \( \times \) 1.2 m \( \times \) 3 m high. The finest mesh resolution used comprised of hexahedral cells of
volume \((0.009375 \, \text{m}^3)\). Details of the mesh sensitivity study conducted can be found in [10]. Angular resolution for the RTE solver was selected based on a sensitivity study also described in [10]. The total number of angles used in the present study is 48 (12 azimuthal and 4 polar).

![Figure 4](image_url)

**Figure 4.** Parallel panel setup: (a) 60 kW propane fire between inert parallel panels, and (b) computational mesh with measurement locations.

The sand-burner anchored propane fire emits radiation isotropically and over time emitted radiant power (approximately 18 kW) remains invariant. The 18 kW radiant emission corresponds to an overall radiant fraction, \(\chi_r = 0.3\). Out of the 18 kW flame emission, approximately 6 kW is incident on each of the two panels (see Fig. 5). Initially, the overall incident radiant power on one panel is 7 kW, but this incident power increases to approximately 9 kW at 50 s, as can be observed in Fig. 5. The additional 2 kW is due to the re-radiation from the adjacent panel, which re-radiates approximately 4 kW at 50 s (50% of this re-radiated component is incident on the neighboring panel). The re-radiation from the panels increases with time as their surface temperature increases.

![Figure 5](image_url)

**Figure 5.** Incident radiation on left panel and re-radiation from the right panel.

The computed incident heat fluxes on the Marinite-I panels are now compared against experimentally observed values for point measurements. As the panels heat up, the surface
temperature increases. The predicted heat flux and surface temperature comparisons against experimental data are shown in Fig. 6. The predicted heat flux for the locations on the left panel show good agreement with the experimental data. Surface temperatures are predicted with reasonably good accuracy, except for the 0.9 m location on the centerline. Further investigation is needed to determine the difference in surface temperature at the 0.9 m location. The prediction error at this height also is large due to the low temperatures encountered.

![Figure 6](a) (b)

**Figure 6.** Parallel panel incident heat flux and surface temperature: (a) centerline locations, and (b) offset locations. —— computation, ◦ experiment.

### 3.2. Fire Growth in a Rack Storage Setup

Fully coupled CFD simulations (pyrolysis, soot and radiation, gas-phase combustion) of fire spread were conducted in a $2 \times 4 \times 3$ rack storage configuration [2]. Pyrolysis of charring materials was simulated with the solution of a 1-D, finite-rate model coupled to the gas-phase solver through boundary conditions for mixture fraction, temperature and velocities [15]. The simulations used solid fuel material properties determined from bench scale fire propagation apparatus (FPA) [16] tests and with the application of optimization schemes [17]. Several instrumented fire tests were conducted under a $20 \text{ MW}$ fire products collector (FPC) with the purpose of providing validation data. Each of the 24 units in the $2 \times 4 \times 3$ configuration consisted of three nested boxes made of double-wall corrugated cardboard enclosing a sheet metal liner. Chemical heat release rates were measured for comparison against computed results.

Figure 7 shows four snapshots showing fire growth in the rack storage arrangement. The fire initially grows vertically between four boxes and lateral spread begins at a later time (HRR is above $3 \text{ MW}$). Initial vertical and lateral spread are due to ignition of corrugated cardboard primarily by incident radiation. Incident heat fluxes of the order of $30 \text{ kW/m}^2$ to greater than $200 \text{ kW/m}^2$ were observed in the computations. Measured heat fluxes in the experiment also showed heat fluxes in excess of $100 \text{ kW/m}^2$ on the central vertical faces of the corrugated boxes.
Figure 7. Rack storage fire growth depicted by the stoichiometric mixture fraction contour at different times corresponding to heat release rates of (a) 0.5 MW, (b) 1 MW, (c) 3 MW, and (d) 9 MW.

The computed HRR is compared against experimentally measured HRR in Fig. 8. Comparison is made after a HRR of 500 kW is reached as the initial ignition process and fire growth do not show a similar behavior in the experiments. This is because for practical testing scenarios, igniter setup and ambient conditions can vary and affect initial ignition of the corrugated cardboard. However, ignition and initial fire growth are known to not influence large scale fire growth. The experimental curves and the computed curve have been time-shifted so that they coincide when 500 kW HRR is reached. Computed HRR shows good agreement with experimental data. The uncertainty/variability of the experimental measurements is illustrated by the scatter of data for repeated tests.

4. Conclusions and Future Work

Results from a CFD study simulating radiation heat transfer from a buoyant turbulent flame to inert parallel panels have been presented. A LES code, FireFOAM, was used in the study with a laminar smoke point based subgrid soot-radiation model implemented and with the solution of a finite volume based RTE solver. A conjugate heat transfer model was applied to compute panel heat-up. Computed radiant fluxes are compared against experimental data and predicted panel surface temperatures are also compared. Overall, the simulation has shown good agreement with experimental data. Fire growth in a rack storage arrangement was also simulated with the application of a pyrolysis model for corrugated cardboard. Computed HRR from the simulation shows good agreement with experimental measurements.

This study assumed optically thin radiation. The model needs to be extended for larger fires where radiation is in the optically thick limit. Also, even though angular discretization tests were performed, the dependence of radiation transfer on the computational mesh has to be further investigated. Ray effects were observed in the parallel panel simulations and their causes will be investigated. Gas-phase radiation was neglected to speed-up the computation of these large-scale simulations. Future work will improve the radiation model with the inclusion of wide-band models for gas-phase radiation developed using statistical narrow-band models.
Figure 8. Rack storage fire growth heat release rate compared against experimental data. —— computation, ◦ experiments. Experimental data beyond peak HRR are not shown since water-spray based suppression was applied in the experiment.

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