Large eddy simulation of suppression of turbulent line fires by base-injected water mist

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Abstract

The objective of this study is to bring basic information on the mechanisms that control flame cooling effectiveness in water mist fire suppression systems. The study is based on well-resolved large eddy simulations (LES) of buoyancy-driven, turbulent, methane-air diffusion flames exposed to water mist injected through a controlled air co-flow. Simulations are performed using an LES solver called FireFOAM and using a modified version recently enhanced with a new flame extinction model based on the concepts of a critical flame Damköhler number for extinction and a critical gas temperature for re-ignition. The numerical simulations provide global information on the flame response to changes in the water mist load as well as spatially-resolved information on the structure of the flame-based heat release rate processes and mist-based evaporation processes. The results suggest that maximum suppression is obtained when mist droplets are entrained into the flame base region.

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1. Introduction

While water-based fire suppression systems provide an attractive and popular solution to the problem of fire safety in the built environment, a fundamental understanding of the factors that control the performance of these systems is still lacking [1–3]. When evaluating the fire suppression performance of water sprays, it is useful to differentiate between systems that achieve suppression by flame cooling, i.e., systems that reduce the fire intensity by acting directly on combustion (e.g., by increasing the probability of flame extinction), and systems that achieve suppression by fuel cooling, i.e., systems that reduce the fire intensity by acting on fuel formation (e.g., by wetting active fuel surfaces and decreasing the rate of pyrolysis, or by wetting inactive fuel surfaces and inhibiting further flame spread). The ability of a given system to achieve flame cooling or fuel cooling depends on its ability to deliver the liquid water to the location of the flame or to that of the fuel surfaces and is therefore strongly related to the droplet size distribution in the liquid spray. Conventional sprinkler systems produce large droplets (larger than 100 microns) that penetrate the gas flow and evaporate slowly; large droplets have a low probability of acting (i.e., evaporating) at the flame location and a high probability of wetting fuel surfaces; sprinkler systems achieve fire suppression primarily by fuel cooling. In contrast, mist systems produce small droplets (smaller than 100 microns) that move with the gas flow and evaporate quickly; small droplets have a relatively high

23 probability of reaching the flame location (at least in confined spaces) and a
 24 low probability of wetting fuel surfaces; mist systems achieve fire suppression
 25 primarily by flame cooling (note that mists systems have also the ability to
 26 block radiation transport via droplet absorption and scattering effects and
 27 can therefore also provide some form of fuel cooling).

28 One of the main objectives of fire suppression research is to fill the
 29 knowledge gap between the engineering-level performance indicators that
 30 are used in the design of practical water-based fire suppression systems and
 31 the research-level performance indicators that are produced by fundamental
 32 studies. Engineering-level performance indicators are global parameters that
 33 typically compare the evaporative cooling power of a sprinkler or mist system
 34 to the size of the fire at activation time: $I_{global} = (\dot{m}_w \Delta H_v) / \dot{Q}_c$, where $\dot{m}_w$ is
 35 the mass flow rate of delivered water (in units of kg/s), ΔH_v the latent heat
 36 of evaporation of water (in J/kg), and $\dot{Q}_c$ the combustion power prior to sup-
 37 pression (in W). Fire suppression requires values of I_{global} that are strongly
 38 configuration-dependent and that vary from values of order 1 to values much
 39 larger than 1 [1–3].

40 Research-level performance indicators are local parameters that take dif-
 41 ferent forms, depending on whether fire suppression is achieved by flame
 42 cooling or by fuel cooling. In the case of flame cooling, a typical indi-
 43 cator compares the rate of heat extraction by evaporative cooling (mea-
 44 sured at the flame surface) to the rate of energy release by combustion:
 45 $I_{flame} = (\dot{\omega}_w'' \Delta H_v) / (\dot{m}_{f,c}'' \Delta H_c)$, where $\dot{\omega}_w''$ is the local mass evaporation rate
 46 of liquid water (in kg/s/m², i.e., per unit area of flame surface), $\dot{m}_{f,c}''$ the
 47 local fuel mass consumption rate by combustion (in kg/s/m², also per unit

48 area of flame surface and measured prior to suppression), and ΔH_c the heat
 49 of combustion (in J/kg). Successful flame cooling requires values of I_{flame}
 50 of order 0.1, which typically corresponds to values of the critical water mass
 51 flux on the order of a few tens of g/s/m² [4–7]. Alternative representative
 52 measures of the water loading requirement for flame cooling are a liquid vol-
 53 ume fraction on the order of 100 ppm or a liquid mass fraction on the order
 54 of 10% [8].

55 We consider in the present study the problem of flame cooling by a water
 56 mist and focus in particular on the relationship between the spray effective-
 57 ness indicators I_{global} and I_{flame} . The global indicator I_{global} is not suitable
 58 to measure spray effectiveness because only a fraction of the delivered liquid
 59 water will actually be transported to the flame and thereby contribute to
 60 flame cooling; in other words, I_{global} incorrectly incorporates water mist that
 61 either fails to evaporate or that evaporates at locations that do not impact
 62 the flame. A clear illustration of this problem is found in Refs. [9, 10] which
 63 present a direct numerical simulation study of flame extinction due to water
 64 mist in laminar co-flow methane-air and methanol-air diffusion flames. In the
 65 most efficient case corresponding to a base injection of the mist inside the
 66 air-coflow and at low flow velocities (25 cm/s), flame extinction is achieved
 67 for different values of I_{global} : $I_{global} \approx 0.75$ in the case of 50 microns droplets
 68 and $I_{global} \approx 3$ in the case of 150 microns droplets (in these estimates, we use
 69 $\Delta H_v = 3.7$ MJ/kg, a representative value that corresponds to the enthalpy
 70 change from liquid water at 293 K to steam at 1500 K). Extinction values
 71 of I_{global} are also found to be much higher in the case of higher mist injec-
 72 tion velocities [9]. Thus it is found in Ref. [9] that the value $I_{global} \approx 0.75$

73 may correspond to a suppressed flame (50 microns droplets case), to a flame
74 that has a much reduced (more than 50%) combustion intensity (100 microns
75 droplets case) or to a flame that is only weakly affected by the presence of the
76 water mist (150 microns droplets case or cases with high injection velocities).
77 These results demonstrate that the global indicator I_{global} does not provide
78 an accurate measure of suppression performance.

79 Our objectives in the present study are to provide additional insight into
80 basic mist-flame interactions and to replace I_{global} by a more suitable fire
81 suppression performance indicator. The study is based on detailed numer-
82 ical simulations which provide spatially-resolved information on the distri-
83 butions of evaporation and combustion rates. We choose to consider a new
84 configuration corresponding to a buoyant, turbulent, methane-fueled, slot-
85 burner diffusion flame with a mono-disperse water spray injected through a
86 controlled air co-flow. This configuration is being currently developed at the
87 University of Maryland [11] (note that experimental data with mist are not
88 available yet and that the present study is a standalone numerical investi-
89 gation). The numerical solver is an advanced large eddy simulation (LES)
90 solver called FireFOAM, developed by FM Global [12, 13] and based on a
91 general-purpose Computational Fluid Dynamics (CFD) solver called Open-
92 FOAM [14]. The FireFOAM solver has been recently enhanced with a new
93 flame extinction model [15, 16]. The LES solver and numerical configuration
94 are presented in Section 2. Results are presented in Section 3, including re-
95 sults documenting changes in combustion intensity in response to changes in
96 water mist load. Conclusions are drawn in Section 4.

2. Numerical solver and configuration

2.1. Numerical configuration

The configuration adopted in the present study is an experimental configuration currently being developed at the University of Maryland [11]. The configuration corresponds to a buoyant, turbulent, methane-fueled, slot-burner diffusion flame with a controlled co-flow. Flames are stabilized above a 5-cm-wide and 50-cm-deep Wolfhard-Parker burner surrounded by a small horizontal plate and an air co-flow (Fig. 1). The plate is 15-cm-wide and 60-cm-deep and is made of Kaowool ceramic board. The plate adds an horizontal component to the flow motion near the flame base and was introduced in previous work with the intent to promote faster transition to turbulent flow conditions [11]. The air co-flow is 50-cm-wide and 75-cm-deep. The methane velocity is 6 cm/s; the methane mass flow rate is 1 g/s; the total heat release rate of the unsuppressed flame is 50 kW.

In the baseline numerical configuration with mist, the air co-flow is saturated with water vapor (100% humidity) and loaded with a mono-disperse liquid water spray. The air co-flow velocity is 29 cm/s and the air mass flow rate is 100 g/s. Selected droplet diameters are 30 μm and 60 μm (larger droplets are too heavy to guarantee successful entrainment into the flame region, in particular when the flame strength is reduced). The initial droplet velocity is equal to the air co-flow velocity; the mist mass flow rate varies between 0 and 18 g/s; the mist volume (mass) fraction varies between 0 and 220 ppm (0 and 15.3%).

We also consider below a modified configuration in which the co-flow is split into an inner area corresponding to air (40% humidity) without water

mist and an outer area corresponding to air (100% humidity) with water mist. The no-mist inner co-flow is introduced in order to protect the base of the flame from suppression. The inner area is 25-cm-wide and 70-cm-deep; the air co-flow velocity in the inner area (also referred to as the protective co-flow) varies between 20 and 50 cm/s (and thereby provides different levels of protection). The air co-flow velocity in the outer area is 42 cm/s; the air mass flow rate is 100 g/s; the mist mass flow rate is 18 g/s.

2.2. Solver

FireFOAM [12, 13] is based on OpenFOAM [14], an open-source general-purpose CFD software package. FireFOAM is an object-oriented, C++-based, second-order accurate, finite volume solver with implicit time integration; PISO and SIMPLE types of solution procedures are used to couple sequentially-solved equations; the solver features advanced meshing capabilities (the mesh geometry is a structured or unstructured polyhedral mesh); it also features a massively parallel computing capability using Message Passing Interface (MPI) protocols.

FireFOAM uses a Favre-filtered compressible-flow LES formulation and provides a choice between several modeling options for the treatment of turbulence, combustion and thermal radiation. In the present study, subgrid-scale turbulence is described using the one-equation eddy viscosity model (a model based on solving a transport equation for subgrid-scale turbulent kinetic energy or TKE) [17]. Combustion is described using the classical concept of a global combustion equation combined with the Eddy Dissipation Concept (EDC) model [18]. Thermal radiation is described by solving the radiative transfer equation (RTE). Radiative properties are considered as

147 spectrally-averaged quantities and a gray medium is assumed. In FireFOAM,
 148 the RTE is solved using a finite volume implementation of the discrete or-
 149 dinate method [19]. In the present study, the description of thermal radia-
 150 tion is simplified by assuming a non-scattering, non-absorbing, optically-thin
 151 medium and using the empirical concept of a user-prescribed global radiative
 152 loss fraction χ_{rad} ; we use $\chi_{rad} = 23\%$ [11]. The assumption of an emitting
 153 but non-absorbing medium as well as that of a fixed radiative loss frac-
 154 tion should be viewed as an intermediate modeling step. Also note that
 155 the present methane-fueled flames are only weakly sooting and that soot is
 156 simply neglected in the model formulation.

157 For the water spray, FireFOAM uses a classical Lagrangian-based model-
 158 ing approach that treats the liquid phase as a discrete phase, injects represen-
 159 tative (spherical-shaped) droplets according to prescribed size and velocity
 160 boundary conditions, tracks the individual trajectories of the representative
 161 droplets, and includes two-way coupling between the gas and liquid phases,
 162 with a limitation to dilute conditions (this limitation is deemed acceptable
 163 for the present application) [20]. We refer the reader to Refs. [4, 21, 22] for
 164 a description of the Lagrangian-based droplet equations and Eulerian-based
 165 gas equations.

166 We now turn to a description of the flame extinction model in FireFOAM.
 167 The FireFOAM solver has been recently enhanced with a new flame extinc-
 168 tion model based on the concepts of a critical flame Damköhler number for
 169 extinction and a critical gas temperature for re-ignition [15, 16]. The model
 170 was initially formulated using the concept of mixture fraction [15]; because
 171 mixture fraction does not apply to problems with multiple fuels and multi-

172 ple oxidizer streams, the model has been recently re-formulated to a more
 173 general framework that does not use mixture fraction [23].

174 The combustion model in FireFOAM uses a classical global combustion
 175 equation: $CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$ (GR1). Global reaction (GR1)
 176 corresponds to a normal combustion step for methane fuel. When con-
 177 sidering extinction, reaction (GR1) is enhanced by two additional steps:
 178 $CH_4 + 2O_2 \rightarrow (CH_4)^* + 2O_2$ (GR2), and $(CH_4)^* + 2O_2 \rightarrow CO_2 + 2H_2O$ (GR3),
 179 where global reaction (GR2) describes flame extinction, treated as a transfor-
 180 mation of fresh fuel CH_4 into non-burning fuel $(CH_4)^*$, and global reaction
 181 (GR3) describes re-ignition, treated as a normal combustion step for $(CH_4)^*$.
 182 The introduction of two separate combustion steps (GR1) and (GR3) allows
 183 for a separate treatment in the model of extinction and re-ignition phenom-
 184 ena. The list of transported species in the combustion model includes CH_4 ,
 185 $(CH_4)^*$, O_2 , CO_2 and H_2O .

186 The closure expressions for the chemical reaction rates are based on the
 187 EDC model [18]:

$$\begin{aligned}
 \overline{\dot{\omega}_{GR1}'''} &= (1 - FEF) \times C_{EDC} \left(\frac{\bar{\rho}}{\tau_t} \right) \times \min \left(\tilde{Y}_{CH_4}, \frac{\tilde{Y}_{O_2}}{r_s} \right) \\
 \overline{\dot{\omega}_{GR2}'''} &= FEF \times C_{EDC} \left(\frac{\bar{\rho}}{\tau_t} \right) \times \min \left(\tilde{Y}_{CH_4}, \frac{\tilde{Y}_{O_2}}{r_s} \right) \\
 \overline{\dot{\omega}_{GR3}'''} &= FIF \times C_{EDC} \left(\frac{\bar{\rho}}{\tau_t} \right) \times \min \left(\tilde{Y}_{(CH_4)^*}, \frac{\tilde{Y}_{O_2}}{r_s} \right)
 \end{aligned} \tag{1}$$

188 where FEF and FIF designate the flame extinction and re-ignition factors,
 189 where C_{EDC} is a model coefficient ($C_{EDC} = 4$), $\bar{\rho}$ the (LES-filtered) mass
 190 density, τ_t the subgrid-scale turbulent mixing time scale ($\tau_t = (1/C_\epsilon) \times$

191 $(\Delta/(k_{SGS})^{1/2})$, with C_e a dynamic model coefficient, Δ the local grid cell
 192 size, and k_{SGS} the subgrid-scale TKE), where $\tilde{Y}_{CH_4}$, $\tilde{Y}_{(CH_4)^*}$ and $\tilde{Y}_{O_2}$ are the
 193 (LES-filtered) mass fractions of fuel, non-burning fuel and oxygen, and r_s is
 194 the stoichiometric oxygen-to-fuel mass ratio .

195 In Eq. 1, the flame extinction factor FEF takes values between 0 and
 196 1 and gives a measure of the local probability of flame extinction. Flame
 197 extinction is treated via a critical Damköhler number criterion: $FEF = 0$ if
 198 $Da \geq Da_c$ (no extinction) and $FEF = 1$ if $Da < Da_c$ (full extinction). We
 199 use $Da_c = 1$ [16] and write:

$$Da = C \times \frac{(1/\chi_{st})}{\exp(T_a/T_{st})} \quad (2)$$

200 where C and T_a are model parameters, T_{st} is the flame temperature and χ_{st}
 201 the rate of fuel-air mixing. In Eq. 2, $(1/\chi_{st})$ represents a mixing time scale
 202 while $(1/C) \exp(T_a/T_{st})$ represents a chemical time scale; T_a is an effective
 203 activation temperature which represents the sensitivity of the combustion
 204 chemistry to changes in temperature. C and T_a are fuel dependent properties
 205 that have been calibrated in the present study through comparison with
 206 direct numerical simulation data on extinction limits of methane-air laminar
 207 diffusion flames [24], $C = 1.9616 \times 10^{10} \text{ s}^{-1}$ and $T_a = 36856 \text{ K}$. The flame-
 208 based quantities T_{st} and χ_{st} are obtained via subgrid-scale models and are
 209 discussed next.

210 The flame temperature T_{st} is described as a weighted average between
 211 the adiabatic flame temperature T_{st}^{ad} and the mixing temperature T_{st}^m , using
 212 the deficit enthalpy variable H_{st} as a weight coefficient:

$$T_{st} = (1 + H_{st}) T_{st}^{ad} - H_{st} T_{st}^m \quad (3)$$

213 T_{st}^{ad} is the temperature that would be obtained with adiabatic combustion
 214 (i.e., without radiation losses, convective wall losses, or an evaporating spray);
 215 we use $T_{st}^{ad} = 2240$ K. T_{st}^m is the temperature that would be obtained with
 216 pure fuel-air mixing and without combustion; we use $T_{st}^m = 293$ K. The deficit
 217 enthalpy variable H_{st} takes values between (-1) and 0 and gives a measure of
 218 the magnitude of non-adiabatic effects at the flame location: $H_{st} = 0$ under
 219 adiabatic burning conditions while $H_{st} = -1$ under extinguished conditions.
 220 We write:

$$H_{st} = -\frac{(\tilde{h}_s^{ad} - \tilde{h}_s)}{(\tilde{h}_s^{ad} - \tilde{h}_s^m)} \quad (4)$$

221 where $\tilde{h}_s$, $\tilde{h}_s^{ad}$ and $\tilde{h}_s^m$ are the (LES-filtered) sensible enthalpy, adiabatic sen-
 222 sible enthalpy and mixing sensible enthalpy, respectively. $\tilde{h}_s$ is a principal
 223 variable in the FireFOAM system of governing equations and is therefore
 224 obtained directly from the original FireFOAM transport equation for energy.
 225 $\tilde{h}_s^{ad}$ and $\tilde{h}_s^m$ are obtained from additional modified transport equations. The
 226 calculation of $\tilde{h}_s^{ad}$ is identical to that of $\tilde{h}_s$ except that the thermal radia-
 227 tion and spray evaporation terms in the enthalpy equation are set to 0 and
 228 boundary conditions are changed to adiabatic. Similarly, the calculation of
 229 $\tilde{h}_s^m$ is identical to that of $\tilde{h}_s^{ad}$ except that the combustion term in the enthalpy
 230 equation is set to 0. This treatment provides a simple method to account for
 231 local decreases in flame temperature due to the presence of thermal losses. In
 232 the present study, thermal losses are due to both emission of radiant energy
 233 and cooling from the evaporating water mist.

Furthermore, the rate of fuel-air mixing χ_{st} in Eq. 2 is described based on a classical closure expression for the LES-filtered scalar dissipation rate, $\tilde{\chi} = 2(D + D_t) \left| \nabla \tilde{Z} \right|^2$, where D and D_t are the molecular and turbulent thermal diffusivities, and $\tilde{Z}$ is the mixture fraction. Replacing mixture fraction by its expression in terms of fuel and oxygen mass, we get:

$$\chi_{st} = 2(D + D_t) \left(\frac{r_s \nabla \tilde{Y}_{CH_4} - \nabla \tilde{Y}_{O_2}}{r_s + Y_{O_2,air}} \right)^2 \quad (5)$$

where $Y_{O_2,air}$ is the oxygen mass fraction in air, $Y_{O_2,air} = 0.233$.

Finally, in Eq. 1, the flame re-ignition factor FIF takes values between 0 and 1 and gives a measure of the local probability of flame re-ignition. Flame re-ignition is treated via a critical gas temperature criterion: $FIF = 0$ if $\tilde{T} \leq T_{ign}$ (no re-ignition) and $FIF = 1$ if $\tilde{T} \geq T_{ign}$ (re-ignition). We use $T_{ign} = 1100$ K [25].

2.3. Resolution and computational cost

The computational domain is 200-cm-wide in the cross-stream x -direction, 85-cm-deep in the spanwise y -direction and 200-cm-high in the vertical z -direction. The computational grid is generated using the OpenFOAM mesh tool called SnappyHexMesh [14]. The grid is block-structured and features three levels of grid resolution (Fig. 1). The level-one refinement block contains the flame region and features a uniform grid with cubic cells of size equal to 4.167 mm (i.e., 12 grid cells across the burner width). The level-one refinement block is 40-cm-wide, 60-cm-deep and 60-cm-high. The level-two refinement block features a uniform grid with cubic cells of size equal to 8.333 mm; this block is 60-cm-wide, 80-cm-deep and 80-cm-high. The level-three

Figure 1: Computational domain and multi-level mesh refinement; the Wolfhard-Parker burner is 5 cm wide in the x -direction and 50 cm deep in the y -direction; the flame height is approximately 50 cm.

256 refinement block contains the far-field air entrainment and plume regions and
 257 features a uniform grid with cubic cells of size equal to 16.67 mm. The total
 258 number of cells is 1.85 million.

259 Previous work corresponding to simulations of similar flames but without
 260 mist included a grid-convergence study and concluded that for the present
 261 moderate-Reynolds-number flames, a 4 mm grid resolution is sufficiently fine
 262 to achieve approximate grid independence of the LES solutions [23]. The
 263 present LES simulations are well-resolved in the sense that anywhere in the

264 flow, the local ratio of mean grid-resolved TKE divided by mean total TKE
 265 (equal to the sum of mean grid-resolved TKE plus mean subgrid-scale TKE)
 266 is above 90%. In addition, the ratio of mean subgrid-scale eddy viscosity
 267 divided by molecular viscosity remains less than 1.

268 The accuracy of the Lagrangian-based treatment of the water mist is con-
 269 trolled by the number of numerical droplets n_{inj} injected into the computa-
 270 tional domain per second. To minimize sampling error, we use $n_{inj} = 100,000$
 271 [26].

272 All simulations are 14 s in duration. Turbulent statistics are collected
 273 for a duration of at least 10 s at the end of each simulation, once the flow
 274 and flame become statistically stationary. Each simulation is run using 40
 275 processors on a large-scale Linux cluster; with typical runs requiring on the
 276 order of 1500 hours of CPU time.

277 **3. Results and discussion**

278 *3.1. Mist-flow interactions*

279 We first consider a series of preliminary test simulations in which the ex-
 280 tinction model is disabled (the flame power remains equal to 50 kW). These
 281 test simulations provide insight into the interactions between the water mist
 282 and the air co-flow. We consider here our baseline configuration correspond-
 283 ing to a mono-disperse spray of 30 μm droplets moving with the air co-flow
 284 at 29 cm/s. This baseline configuration does not include a protective co-flow.

285 While the water droplets are initially injected at the air co-flow velocity,
 286 the droplets are observed to quickly decelerate due to gravity, causing in turn
 287 a deceleration of the air flow due to drag. Thus, the momentum exchange

288 between the gas and liquid phases is capable of changing the air entrain-
 289 ment process into the flame zone. The intensity of the momentum exchange
 290 between gas and liquid depends on the water load. For small liquid-to-gas
 291 mass flow rate ratios (e.g., of order 1% or smaller), the droplets do not dis-
 292 turb the air co-flow significantly and the buoyant motions are strong enough
 293 to successfully entrain the droplets into the entire length of the flame zone
 294 (Fig. 2-a). In contrast, for large liquid-to-gas mass flow rate ratios (e.g.,
 295 larger than 10%), the droplets significantly modify the air co-flow and the
 296 buoyant motions are now too weak to entrain the droplets beyond a small
 297 region at the base of the flame (Fig. 2-b). Thus, different water loads will
 298 result in different distributions of liquid and vaporized water along the flame,
 299 irrespective of how the flame responds.

Figure 2: Instantaneous mist droplet distribution in test simulations without flame ex-
 tinction; the air co-flow mass flow rate is 100 g/s; the mist corresponds to 30 μm droplets
 with different mist mass flow rates: a) $\dot{m}_w = 2 \text{ g/s}$; b) $\dot{m}_w = 18 \text{ g/s}$. In the pictures, the
 flame is shown as an iso-contour of the heat release rate per unit volume (200 kW/m³).

300 3.2. Mist-flame interactions

301 We now turn to a discussion of simulations in which the extinction model
 302 is activated. We consider three configurations: (1) the baseline configuration,
 303 called Case 1, discussed in the previous section (monodisperse spray with
 304 droplet diameter equal to $30\text{ }\mu\text{m}$, variable mist mass flow rate $\dot{m}_w$ between
 305 0 and 18 g/s , and without a protective co-flow); (2) a similar configuration,
 306 called Case 2, with $60\text{ }\mu\text{m}$ droplets; and (3) a modified configuration, called
 307 Case 3, with a protective co-flow (monodisperse spray with droplet diameter
 308 equal to $30\text{ }\mu\text{m}$, mist mass flow rate $\dot{m}_w$ equal to 18 g/s). In Case 3, we
 309 consider four values of the protective co-flow velocity: $w_{air,p} = 20, 30, 40$ and
 310 50 cm/s ; higher flow velocities provide higher levels protection of the flame
 311 from the water mist.

312 Figure 3 presents the combustion efficiency χ_a as a function of the perfor-
 313 mance indicator I_{global} discussed in the Introduction section. χ_a is defined as
 314 the ratio of the (time- and volume-averaged) heat release rate (HRR) of the
 315 flame with mist divided by HRR of the flame without mist (i.e., 50 kW). I_{global}
 316 is the theoretical evaporative cooling power (ECP) of the mist, $(\dot{m}_w \Delta H_v)$,
 317 divided by HRR of the flame without mist; we use $\Delta H_v = 3.7\text{ MJ/kg}$.

318 Figure 3 shows that consistent with previous results found in the literature
 319 (e.g., [9]), the response of the flame in Cases 1 and 2 corresponds to a decrease
 320 in flame strength with increasing values of I_{global} . The simulated flames
 321 correspond to values of I_{global} between 0 and 1.33: in Case 1, total flame
 322 extinction is (almost) achieved for $I_{global} \approx 0.8$, with a small residual flame
 323 remaining near the fuel injection plane. In contrast, in Case 2, even at the
 324 highest simulated water mist load, the flame remains partially burning. The

Case 2 flame requires higher values of I_{global} for total extinction. In other words, Fig. 3 suggests that the mist with 30 μm droplets is more efficient than the mist with 60 μm droplets (total extinction is achieved for smaller amounts of injected liquid water).

Case 3 corresponds to a dramatically different flame response. Case 3 includes four simulations corresponding to different values of $w_{air,p}$. All simulations feature the same value of the mist mass flow rate $\dot{m}_w$ and therefore appear on a vertical line in Figure 3, $I_{global} = 1.33$. At high values of $w_{air,p}$, the flame is insulated from the outer co-flow and has limited interaction with the mist; the combustion efficiency is high. At low values of $w_{air,p}$, the flame interacts strongly with the outer co-flow; the combustion efficiency is low. These results illustrate the failure of I_{global} to give a robust and meaningful measure of the spray effectiveness: the same value $I_{global} = 1.33$ corresponds to flames with different levels of combustion efficiency, $\chi_a \approx 10\%, 20\%, 50\%, 80\%$ for $w_{air,p} = 20, 30, 40, 50$ cm/s, respectively.

As mentioned in the introduction, the main deficiency in the performance indicator I_{global} is that it uses the total amount of liquid water delivered rather than the fraction that evaporates and interacts with the flame. We consider now a modified version of the indicator, noted I_{global}^* , based on the simulated (time- and volume-averaged) evaporation rate of the liquid phase: $I_{global}^* = (\dot{\omega}_w \Delta H_v) / \dot{Q}_c$, where $\dot{\omega}_w$ is the global mass evaporation rate of liquid water (in units of kg/s),

Figure 4 presents the combustion efficiency χ_a as a function of the modified performance indicator I_{global}^* . It is seen that the values of I_{global}^* range between 0 and 0.4 and as expected, are much lower than those of I_{global}

Figure 3: Combustion efficiency χ_a as a function of I_{global} for Case 1 (circles), Case 2 (triangles) and Case 3 (squares). Case 1 and Case 2 data points correspond to different values of the mist mass flow rate $\dot{m}_w$; Case 3 data points correspond to different values of the protective co-flow velocity $w_{air,p}$ (from high to low velocity as one moves from the top to bottom points in the plot).

Figure 4: Combustion efficiency χ_a as a function of I_{global}^* . See Fig. 3.

350 ($\dot{\omega}_w < \dot{m}_w$). The curves in Fig. 4 feature an abrupt transition from high to
 351 low values of the combustion efficiency: this transition occurs at $I_{global}^* \approx 0.25$,
 352 0.15 and 0.35, in Cases 1, 2 and 3, respectively.

353 When comparing Case 1 and Case 2, it is seen that Case 2 corresponds
 354 to less evaporation (I_{global}^* takes lower values) and a more efficient spray
 355 (transition to extinction occurs at a lower critical value of I_{global}^*). In other
 356 words, and in contrast to the conclusion drawn earlier from Fig. 3, Fig. 4
 357 suggests that the mist with 60 μm droplets is more efficient than the mist with
 358 30 μm droplets (total extinction is achieved for smaller amounts of evaporated
 359 liquid water). Thus, while the Case 2 mist is less efficient than the Case 1
 360 mist at delivering water to the flame (Fig. 3), the delivered water in Case 2
 361 is more efficient than that in Case 1 at suppressing the flame (Fig. 4).

362 When comparing Case 1 and Case 3, it is seen that Case 3 corresponds
 363 to more evaporation (I_{global}^* takes higher values) and a less efficient spray
 364 (transition to extinction occurs at a higher critical value of I_{global}^*). The
 365 result that Case 3 corresponds to more evaporation is somewhat surprising
 366 since it may have been assumed that the protection provided by the inner
 367 co-flow would negatively impact evaporation. This result is explained next
 368 by considering the spatial distribution of HRR and ECP as a function of
 369 elevation above the burner.

370 We now consider spatially-resolved information and compare combustion
 371 and evaporation intensities inside the flame region. This comparison is made
 372 in the context of reference simulations in which the extinction model is dis-
 373 abled. The idea here is to evaluate the spray dispersion and evaporation
 374 properties prior to suppression and to thereby differentiate between mist-

375 flame configurations in which the bulk of the evaporation takes place inside
 376 the flame and mist-flame configurations in which the bulk of the evapora-
 377 tion takes place downstream. Figure 5 presents the vertical variations of
 378 horizontally-integrated HRR and ECP, $\dot{Q}'_c$ and $(\dot{\omega}'_w \Delta H_v)$ (both in units of
 379 W/m). Figure 5 includes results from three simulations with similar val-
 380 ues of the combustion efficiency: a Case 1 simulation with $\dot{m}_w = 8$ g/s and
 381 $\chi_a \approx 0.28$; a Case 2 simulation with $\dot{m}_w = 18$ g/s and $\chi_a \approx 0.28$; and a Case
 382 3 simulation with $\dot{m}_w = 18$ g/s, $w_{air,p} = 30$ cm/s and $\chi_a \approx 0.21$.

383 Figure 5 shows that the HRR profiles are similar in all three cases (the
 384 flame power is equal to 50 kW) with some variations associated with the
 385 impact of the spray and/or protective co-flow on fuel-air mixing. Figure 5
 386 also shows that the ECP profiles feature more pronounced variations. Case 1
 387 and Case 3 feature high evaporation rates up to mid-flame height, $z \leq 25$ cm,
 388 and negligible evaporation beyond. Case 2 features high evaporation rates
 389 at the base of the flame, $z \leq 10$ cm, and negligible evaporation at higher
 390 elevations. This result for Case 2 is consistent with the discussion of mist-flow
 391 interactions in the previous section that suggests that high concentrations of
 392 small droplets (Fig. 2-b) or equivalently more moderate concentrations of
 393 larger droplets can limit the region of influence of the spray to the base of
 394 the flame.

395 Figure 5 sheds some light on some of the results presented in Fig. 4.
 396 The total rate of evaporation $\dot{\omega}_w$ is equal to the area under the ECP curves
 397 plotted in the figure and it is seen that Case 1 evaporates more than Case 2
 398 but evaporates less than Case 3. In Case 3, increasing values of $w_{air,p}$ result
 399 in more evaporation occurring in the downstream portion of the flame as well

400 as in the plume region; this result explains the surprisingly high values of $\dot{\omega}_w$
 401 found in Case 3. Figure 5 also shows that the ECP curves take comparable
 402 values near the base of the flame (at $z \leq z_{base}$ with $z_{base} \approx 10$ cm) but differ
 403 significantly at higher elevations. Because the three cases presented in the
 404 figure correspond to similar values of the combustion efficiency, one can then
 405 postulate that χ_a is controlled by the rate of evaporation in the base region
 406 of the flame.

Figure 5: Heat release rate per unit height (black) and evaporative cooling power per unit height (grey) as a function of elevation for three representative simulations corresponding to Cases 1, 2 and 3. Reference simulations without flame extinction. The flame height is approximately 50 cm.

407 We now test this postulate by introducing a third performance indicator
 408 focused on spray effectiveness at the base of the flame, I_{base}^* . I_{base}^* is defined
 409 as the ratio of mean ECP integrated over the flame base region divided by
 410 mean HRR integrated over the same region:

Figure 6: Combustion efficiency χ_a as a function of I_{base}^* . See Fig. 3.

$$I_{base}^* = \frac{(\int_0^{z_{base}} (\dot{\omega}'_w \Delta H_v) dz)}{(\int_0^{z_{base}} \dot{Q}'_c dz)} \quad (6)$$

Our baseline choice for the extent of the flame base region is $(z_{base}/L_f) = 0.2$, where L_f is the flame height ($L_f \approx 50$ cm), or $z_{base} = 10$ cm. Figure 6 presents the combustion efficiency χ_a as a function of I_{base}^* . The horizontal bars in the figure correspond to variations in the estimates of I_{base}^* when (z_{base}/L_f) changes from 0.1 to 0.3. While there is some scatter, it is seen that the collapse of the data is much better than that obtained in Figs. 3 and 4.

This result supports the idea that flame suppression is controlled by the ability of the spray to weaken the flame in the base region. The importance of the base region comes from its role as a high combustion intensity region (HRR is maximum at low elevations) as well as its role for flame stabilization (flame elements at low elevations have a large impact on HRR at downstream

423 locations).

424 4. Conclusion

425 This paper presents a numerical study of suppression performance by a
426 water mist injected at the base of a buoyant, turbulent, methane-fueled, slot
427 burner diffusion flame with an emphasis on engineering- and research-level
428 indicators used to evaluate spray effectiveness. The study is based on a series
429 of well-resolved large eddy simulations performed with FireFOAM [12] and
430 using a modified version recently enhanced with a new flame extinction model
431 [15, 23].

432 The present study provides basic information on the flame response to
433 changes in the water mist load and in particular on the variations of the
434 combustion efficiency with respect to droplet size, number density or the
435 presence of a protective co-flow. Results highlight the limitations of global
436 spray effectiveness indicators that are based on delivered water rather than
437 evaporated water, or are based on evaporated water without an appreciation
438 for where the evaporation is taking place with respect to the flame structure.
439 Results also show that a local spray effectiveness indicator defined as the
440 mean evaporative cooling power of the mist, measured in the flame base
441 region, divided by the mean heat release rate of the flame, also measured in
442 the flame base region, provides a fair collapse of the flame extinction data.

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502 **Figure Captions**

503 Figure 1. Computational domain and multi-level mesh refinement; the Wolfhard-
504 Parker burner is 5 cm wide in the x -direction and 50 cm deep in the y -
505 direction; the flame height is approximately 50 cm.

506 Figure 2. Instantaneous mist droplet distribution in test simulations without
507 flame extinction; the air co-flow mass flow rate is 100 g/s; the mist corre-
508 sponds to 30 μm droplets with different mist mass flow rates: a) $\dot{m}_w = 2 \text{ g/s}$;
509 b) $\dot{m}_w = 18 \text{ g/s}$. In the pictures, the flame is shown as an iso-contour of the
510 heat release rate per unit volume (200 kW/m^3).

511 Figure 3. Combustion efficiency χ_a as a function of I_{global} for Case 1 (circles),
512 Case 2 (triangles) and Case 3 (squares). Case 1 and Case 2 data points
513 correspond to different values of the mist mass flow rate $\dot{m}_w$; Case 3 data
514 points correspond to different values of the protective co-flow velocity $w_{air,p}$
515 (from high to low velocity as one moves from the top to bottom points in the
516 plot).

517 Figure 4. Combustion efficiency χ_a as a function of I_{global}^* . See Fig. 3.

518 Figure 5. Heat release rate per unit height (black) and evaporative cooling
519 power per unit height (grey) as a function of elevation for three representative
520 simulations corresponding to Cases 1, 2 and 3. Reference simulations without

521 flame extinction. The flame height is approximately 50 cm.

522 Figure 6. Combustion efficiency χ_a as a function of I_{base}^* . See Fig. 3.