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# Combustion of forest litters under slope conditions: Burning rate, heat release rate, convective and radiant fractions for different loads

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## ABSTRACT

A set of experiments at laboratory scale was conducted to study the combustion of forest fuel beds in order to quantify the contribution of radiant and convective heat transfer under slope condition. To proceed, a Large Scale Heat Release apparatus was used to measure the heat release rate and fire properties such as the mass loss rate, the geometry of the fire front and the heat transfer were assessed. Because of the slope and the size of the fuel bed, the mass loss rate and the heat release rate do not reach a quasi-steady state when the propagation takes place under slope condition. This is due to a V-shape distortion of the fire front, which leads to an increase of the burnt surface rate by the fire over time. The study of this quantity has shown that the heat release over time can be estimated with the fuel load and the time derivative of the burnt surface. The fractions of radiation and convection released by the fire in its environment were calculated. Under a slope of 20°, the convective fraction decreases from 74.9% to 61.1% whereas the overall radiant fraction ranges between 25.1% and 38.9% and increases with increasing fuel loads.

## 1. Introduction

Each year, fires devastate forests from all continents causing casualties and irreparable environmental damages. Determining the parameters that control the spread of wildfires is therefore a major objective for the management and the prevention of these fires. Over the last decades, modeling has been increasingly used for forest management [1], for risk mapping [2] and for studying the fire propagation [3]. However, to implement and test these models, it is necessary to have experimental data that increase the understanding of the phenomena.

The effects of slope, wind and vegetation properties on fire behavior have been extensively investigated at laboratory and field scale. In most studies, the researchers have focused on the rate of spread [4–11], the flame geometry [4,9,12], the temperature profile [4,9,11] or the fuel consumption and mass loss [4,7,11]. However, these parameters are not sufficient to completely understand the fire behavior. Increased understanding about the relative roles of radiant and convective heating in fire spread and energy release are essential needs in wildland fire science.

In the literature, the radiant fraction has been mainly determined in works devoted to assess the thermal radiation incident upon a target [13]. The point source model has been the method most commonly used. This method relies on the hypothesis that the radiant heat flux emitted by the flames is spherically isotropic [13,14]:

$$q_R = \frac{\mathrm{HRR}_{\mathrm{rad}} \cdot \cos\theta}{4 \cdot \pi \cdot R^2} \qquad (1)$$

where $q_R$ is the radiant heat flux density, $\theta$ is the angle between the normal to the target and the line of sight from the target to the point source location, $R$ is the distance between the point source and the target. $\mathrm{HRR}_{\mathrm{rad}}$ is the radiant power given by the radiant fraction $\chi_{\mathrm{rad}}$ times the heat release rate. This method has been widely applied for the study of liquid hydrocarbon pool fires and for hydrocarbon jet flames [13]. Tihay et al. [15] used also this approach to calculate the radiant fraction of laminar flames obtained from the burning of the vegetative fuels. The obtained values of the radiant fraction varied between 20% and 27%. Kremens et al. [16] conducted also experiments in 8 × 8 m outdoor plots using preconditioned wildland fuels characteristic of mixed-oak forests. With a two-band infrared radiometer, they studied the radiant power of a fire and found a radiant fraction of 17%.

However, the point source model approach is only accurate in the far-field. For measurement at distances within a few fire diameters, this method becomes inadequate because it assumes that all of the radiant energy from the fire is emitted at a single point

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## Nomenclature

| | |
|---|---|
| $A$ | cross sectional area of the exhaust duct ($m^2$) |
| $a$ | parameter equals to $\frac{\pi dU}{\dot{M}_{mix} c_{p,mix}}$ ($m^{-1}$) |
| $A_{burnt}$ | area burnt by the fire front ($m^2$) |
| $c_{p,mix}$ | specific heat of the mixture (J/kg K) |
| $d$ | duct diameter (0.4 m) |
| $D$ | distance between the heat flux gauge and the fire front (m) |
| $D_{head}$ | distance covered by the head of the fire front (m) |
| $E$ | heat release per unit mass of $O_2$ consumed (J/kg) |
| $F$ | view factor (–) |
| $h$ | fuel height (m) |
| $H_{c,net}$ | net heat of combustion (J/kg) |
| $H_{eff}$ | effective heat of combustion (J/kg) |
| HRR | heat release rate (W) |
| $K$ | partial derivative of the temperature versus time (K/s) |
| $K_v$ | extinction coefficient ($m^{-1}$) |
| $k_t$ | constant determined via a propane burner calibration (–) |
| $k_p$ | constant of the bi-directional probe (–) |
| $L$ | length of the fire front (m) |
| $l$ | distance between the duct entry and thermocouple TK (m) |
| $L_h$ | length of the part of the fire front where the flames were the highest (m) |
| $L_f$ | flame length (m) |
| $L_{thick}$ | flame thickness (m) |
| $m$ | mass of the fuel (kg) |
| $\dot{M}_{mix}$ | mass flow rate of the chemical compounds–air mixture (kg/s) |
| MLR | mass loss rate (kg/s) |
| $n$ | number of moles (mol) |
| $\dot{n}^{\circ}_{O_2}$ | molar flow rates of $O_2$ in incoming air (mol/s) |
| $\dot{n}_{O2}$ | molar flow rates of $O_2$ in the exhaust duct (mol/s) |
| $P_{rad}$ | radiant power (W) |
| $q$ | radiant heat flux density ($W/m^2$) |
| $R$ | distance between the point source and the target |
| RG | radiant heat flux gauge (–) |
| ROS | Rate of spread (m/s) |
| $S$ | emission surface ($m^2$) |
| $S_{exp}$ | total exposed surface area of the material ($m^2$) |
| $U$ | overall heat transmission coefficient of the duct ($W/m^2$ K) |
| $t$ | time (s) |
| $T$ | temperature (K) |
| TG | total heat flux gauge (–) |
| THR | total heat release (J) |
| TK | thermocouple in the exhaust duct (–) |
| $v$ | gas velocity (m/s) |
| $\dot{V}$ | standard flow rate in the exhaust duct ($m^3/s$) |
| $w$ | fuel load ($kg/m^2$) |
| $W$ | molecular weight (kg/mol) |
| $x$ | abscissa (m) |
| $X$ | mole fraction (–) |

*Greek symbols*

| | |
|---|---|
| $\alpha$ | expansion factor for the fraction of the air that was depleted of its oxygen (–) |
| $\beta$ | flame angle (°) |
| $\delta$ | angle of the fire front (°) |
| $\varepsilon$ | emissivity (–) |
| $\Delta P$ | pressure drop across the bi-directional probe (Pa) |
| $\Delta A_{burnt}$ | variation of the area burnt by the fire front ($m^2$) |
| $\phi$ | oxygen depletion factor (–) |
| $\Phi$ | heat flux density ($W/m^2$) |
| $\gamma$ | slope angle (°) |
| $\eta$ | fuel consumption efficiency (–) |
| $\rho_0$ | density of dry air at 298 K and 1 atm ($kg/m^3$) |
| $\theta$ | angle between the normal to the target and the line of sight from the target to the point source location |
| $\chi_{conv}$ | convective fraction (–) |
| $\chi_{rad}$ | radiant fraction (–) |
| $\chi_T$ | sum of the convective and radiant fractions (–) |

*Subscript*

| | |
|---|---|
| 0 | initial value |
| a | ambient |
| *air* | air |
| $CO_2$ | carbon dioxide |
| conv | convective |
| em | embers |
| fl | flame |
| flank | flank of the fire front |
| g | gas |
| head | head of the fire front |
| $O_2$ | oxygen |
| rad | radiant |
| RG | radiant heat flux gauge |
| tot | total |

*Superscript*

| | |
|---|---|
| ° | incoming air |
| a | ambient |

rather than distributed over the fire. Conversely, in these situations, the solid flame model can be employed to overcome the inaccuracy of the point source model [13]. This model considers that the flame has an idealized geometrical shape and emits radiant energy uniformly throughout its surface. The thermal radiant heat flux received by an element outside the flame envelope is given by the expression:

$$q_R = F \cdot E \cdot \tau \qquad (2)$$

where $F$ is a geometric view factor, $\tau$ factor and $E$ is the radiant emittance of flame surface given by $E = \varepsilon \cdot \sigma \cdot T^4$ where $T$ is the temperature of the fire, $\varepsilon$ the emissivity and $\sigma$ the Stefan–Boltzmann constant. As for the point source model, the solid flame model has been mainly used to study the thermal radiation of pool fires and jet fires [13,17,18]. The main difficulty of this approach is the determination of the emissivity and flame temperature.

Another approach to obtain the radiant heat release rate emitted by the fire is the integration of the measured spatial distribution of radiant flux. This method necessitates several radiometers positioned radially and vertically over a cylindrical control surface surrounding the fire. This method has been especially used for the study of pool fires [19,20] and jet flames of hydrocarbons [21,22].

More recently, several works focused on the determination of the fire radiative energy. For this, a new approach based on the works of Kaufman et al. [23] and Wooster et al. [24] has emerged in the field of biomass fires. This new method determines the fire radiative energy (FRE) by the analysis of fire pixel radiances in the middle infrared spectral region. Freeborn et al. [25] conducted 44 small-scale experimental fires in a combustion chamber by using ponderosa pine needles, Douglas fir twigs and foliage. With two MIR thermal imaging systems, they measured the fire radiative energy and obtained a radiant fraction of 12.4% based on the