

Research Article

Investigative Study on Convective Heat Transfer inside Compartment during Fire Situation

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According to the geometry of compartments, quantities of smokes released during fire tend to accumulate at ceiling so as to form a cloud of hot gases. Heat transfer between these hot gases and walls is decisive for the development of fire. An increase in temperature of these gases could lead to dangerous phenomena such as flashovers and backdrafts. Owing to experiments and numerical simulation, the objective of the present paper is to investigate on the influence of natural ventilation on convective heat transfer between hot gases and walls of a room in fire. So, varying the ventilation level, it was firstly about to carry out fire tests in an experimental room. Secondly, study was focused on the numerical simulation of these tests so as to estimate velocity field of burnt gases near walls during fire. Validation of numerical results has been done by confronting simulated results to experimental results. A full-scale extrapolation of results enabled revealing that while the ventilation level in the room changes, the amplitude of convective heat transfer changes according to the regime of fire. It was shown that for the fuel-controlled fire, the convective heat transfer coefficient strongly increases with the ventilation factor, and for the ventilation-controlled fire, convective heat transfer coefficient weakly decreases with the ventilation factor and remains nevertheless close to value $8.75W \cdot m^{-2} \cdot K^{-1}$.

1. Introduction

Fire in confined compartment is very complex issue for firemen because of its impossibility to describe or predict how it would behave. However, based on fire science and experience feedback, it is possible to identify typical phases of the development of confined fire. Indeed, depending on the potential fuels involved, their layout, the geometry of the room, and the level of natural ventilation, it is possible to observe different possible evolutions of fire, each one leading to particular phenomena with identifiable characteristics [1–3]. Building fire thus constitutes that category of fires of which the main characteristic is the lack in fresh air. That difficulty in supplying air is due to the presence of walls. The most important parameter, on which other parameters depend, is the heat release rate, which represents the energy released per unit time [4]. Energy rate generated by the fire source depends on the quality of the combustion reaction, which depends itself on the type and the availability of fuel as well as the amount of air inside the room in fire situation. Partial or total confinement of compartments has two main effects on the behaviour of building fire [5].

Firstly, hot gases accumulating at ceiling heat walls which, along with hot gases, radiate heat far to the fire

source. The spread of smokes inside compartment accelerates therefore the preheating and the combustion speed of surrounding combustibles. Secondly, confinement tends to restrict the availability of the oxygen necessary for combustion, which, in case of nonconfined building fires, is often supplied through openings such as windows, doors, or cracks. Contrary to confined fires, the spread of fire inside compartment with at least one opening is piloted by ventilation. That means it depends mainly on the flow rate of the entering air, the size, and the position of that opening [6–11].

The present paper aims to carry out the impact of natural ventilation on convective heat transfer between hot gases inside compartment in fire and its different walls. To achieve that goal, varying the ventilation factor, fire tests were carried out in an experimental domain in which the heat release rate of the fire source was kept constant. Completed by numerical simulation, results enabled discussing the influence of the variation in the level of natural ventilation on convective heat transfer between burnt gases and walls during fire.

2. Materials and Methods

2.1. Experimental Trials. The experimental domain is a room of dimensions $L \times W \times H$: $0.50 \times 0.50 \times 0.50$ m, including on its front wall a door of dimensions $W_0 \times H_0$: $0.20 \text{ m} \times 0.40 \text{ m}$ (Figure 1). Walls are designed with wood panels of thickness 0.02 m. As fire source, diesel fuel is introduced in a pan of diameter 0.12 m and then set inside the room precisely at the center of the floor. Seven N-type thermocouples (TC1-TC7), connected to the data acquisition switch (Agilent 34970A), were used to measure temperatures during experiments. Table 1 presents the different positions inside the experimental domain where thermocouples have been installed. The ventilation level in compartment is defined by the parameter called ventilation factor. That parameter is mathematically traduced by the formulae $W_0 H_0^{1.5}$ [12, 13]. In view of varying the ventilation factor of the experimental room, the width of the door W_0 has been gradually varied while its height H₀ was kept constant. Four geometric configurations with different ventilation factors have thus been tested (Table 2). Each configuration corresponded then to a fire scene to be experimented. To be sure of the repeatability of experiments, each fire scene was repeated three times and only average results were represented and interpreted.

Fire tests consisted of introducing a mass of 0.065 kg of diesel fuel in the fire pan and initiating ignition. That fire source was then left burning inside the room until the amount of fuel initially put in the pan is burned. As illustrated in Figure 2, the fire source was indirectly set on an electronic balance so as to measure the mass loss rate of fuel during tests.

2.2. Numerical Simulation. Numerical simulation can be defined here as the reproduction of physical reality in a computer owing to mathematical models implemented in the simulation software. The choice of numerical model then



FIGURE 1: 3D view of the experimental room with the installed sensors.

TABLE 1: Cartesian coordinates of sensors installed in the experimental room.

Sensors	X direction	Y direction	Z direction
TC1	0.35	0.10	0.12
TC2	0.35	0.10	0.17
TC3	0.35	0.10	0.22
TC4	0.35	0.10	0.27
TC5	0.35	0.10	0.32
TC6	0.35	0.10	0.37
TC7	0.35	0.10	0.41

TABLE 2: Values of ventilation factors according to geometric configurations.

Parameter	Scene 1	Scene 2	Scene 3	Scene 4
W_0 (m)	0.20	0.15	0.10	0.075
H_0 (m)	0.40	0.40	0.40	0.40
$W_0 H_0^{3/2} (m^{5/2})$	0.051	0.038	0.025	0.019



FIGURE 2: Illustration of the experimental room with electronic balance used to measure the mass loss rate of fuel during fire experiments.

depends on the precision, storage space, and the necessary time for performing simulation. Indeed, models are based on algebraic equations or simple analytical approaches, very often resulting from correlations established from experimental data. These models are thus incorporated in complex models so as to take into account various phenomena implied in the physical reality to be reproduced. One of the most complex models used to simulate fluids phenomena is the field model, commonly called CFD (Computational Fluid Dynamics). Also used in fire simulation, that model is established from conservation laws, and the main strongly coupled mathematical equations, describing the physics of compartments fires, are continuity equation (1), momentum equation (2), energy equation (3), and transport equation of chemical species (4) [14–18]:

$$\frac{\partial \rho}{\partial t} + \nabla \left(\rho \, \overrightarrow{u} \right) = 0,\tag{1}$$

$$\frac{\partial}{\partial t}\left(\rho\overrightarrow{u}\right) + \nabla\left(\rho\overrightarrow{u}\overrightarrow{u}\right) = -\nabla P + \nabla\tau + \rho\overrightarrow{g},\tag{2}$$

$$\frac{\partial}{\partial t} \left(\rho h \right) + \nabla \left(\rho \overrightarrow{u} h \right) = \frac{\mathrm{d}P}{\mathrm{d}t} + \nabla \left(\frac{\lambda}{C_p} \nabla h - q_r \right), \tag{3}$$

$$\frac{\partial}{\partial t} \left(\rho Y_k \right) + \nabla \left(\rho \, \overrightarrow{u} \, Y_k \right) = \nabla \left(D_k \nabla Y_k \right) + \dot{w}_{Y_k}. \tag{4}$$

Terms ρ , u, h, c_p , and λ represent the density, velocity, enthalpy, specific heat, and thermal conductivity of the fluid, respectively. P is the total pressure; τ is the tensor of the viscous stresses. Y_k and D_k represent the mass fraction of chemical species k and its molecular diffusion coefficient, respectively; q_r represents the net heat flux due to thermal radiation.

The CFD code used in the present paper is called ISiS. It is a calculation code developed by the Fire and Explosion Laboratory of the IRSN [19]. That code, including a set of combustion, turbulence, soot production, and heat transfer models, can be used to simulate the spread of fire in large, naturally ventilated, mechanically ventilated, or confined compartments. The physical models implemented in ISiS enable 3D simulation of unsteady, low compressibility, and turbulent and reactive or chemically inert flows. ISiS has been validated through a set of tests involving flows with or without combustion reactions [20-22]. The different steps used to perform numerical simulation of fire tests carried out in the present paper are described in Figure 3. Based on Finite Volumes Method, numerical models used to define some main fire phenomena, such as combustion, turbulence, and heat radiation, are presented in Table 3 [19].

2.3. Heat Release Rate. The behavior of fire inside buildings depends mainly on the rate of energy released by the fire source. In a predictive study, its determination could give some indications on eventual spread direction of fire and then help with defining the way occupants should leave the building. It could also enable determining whether the structure will resist during fire. Estimation of heat release



FIGURE 3: Steps of numerical simulation of fire tests.

TABLE 3: Numerical models of main physical phenomena involved in compartment fire.

Physical phenomena	Numerical models
Combustion	EDC model
Turbulence	Standard <i>k</i> -ε model
Heat radiation	P1-approximation model
Walls conditions	Conduction-convection model

rate enables determining other fire parameters such as flame height, position of the hot gas-fresh air interface, hot gas temperature, and concentration in combustion products. These parameters can be used to determine, for example, the tenability and the resistance of the structure. There exist in literature different models estimating the rate of heat released by fire. Some of them are entirely based on experimental data while others try to describe physical phenomena by equations. Pyrolysis and flame propagation are some of those which remain fairly empirical because of their complexity. Heat release rate during compartment fire can be calculated using either (5) or (6) [23, 24]:

$$\dot{Q} = \chi \dot{m}_f H_c, \tag{5}$$

$$\dot{Q} = \chi A_f \dot{m}_f'' H_c, \tag{6}$$

where \dot{m}_f is the mass loss rate of fuel (kg/s); \dot{m}''_f is the mass loss rate of fuel per unit of horizontal surface area (kg.m⁻²/ s⁻¹); A_f is the horizontal area of the fuel pan; H_c is the fuel heat of combustion (kJ.kg⁻¹); χ is the efficiency of combustion, for the case of diesel fuel χ and H_c are equal to 0.82 and 43.4 MJ kg⁻¹, respectively [25].

2.4. Convective Heat Transfer. When fire evolves in confined or semiconfined compartment, insufficiency in air supplying the fire source could relatively lead to incomplete

	V_{f} < 4.88			$4.88 \ll V_f < 30.48$		
	М	N	Р	m	Ň	р
Smooth wall	0.99	0.21	1.00	0.00	0.50	0.78
Rough wall	1.09	0.23	1.00	0.00	0.53	0.78

TABLE 4: Coefficients of the McAdams correlation according to the nature of the wall [40].



FIGURE 4: Image of the flame of each fire scene during the full development phase. (a) Scene 1. (b) Scene 2. (c) Scene 3. (d) Scene 4.

combustion of fuel. The main consequence of that in air is the high production of unburned gases and soot particles. Due to their low density, these combustion products are carried away by the plume and accumulate in the upper compartment part, forming thus a cloud of hot gases. When temperature of that cloud of hot gases reaches some critical values, unpredictable and dangerous phenomena such as flashover or backdraft can occur [26-30]. The increase in temperature of these hot gases depends on the difference between heat generated by the fire source and heat lost through walls and openings. In addition to heat radiated by flames, heat transferred by convection also plays an important role in energy transfers causing compartment fires. As described in (7), besides the difference between fluid temperature $T_{\rm f}$ and wall temperature $T_{\rm w}\text{,}$ convective heat transfer coefficient h_{cv} is also an important parameter necessary to estimate the convective heat flux φ transmitted per unit area [31].

$$\phi = h_{cv} \left(T_f - T_w \right). \tag{7}$$

It is therefore important to remember that the convection coefficient, or convective heat transfer coefficient, is a coefficient that quantifies the heat transferred between a fluid moving against a cold or hot wall. With property like density that changes with temperature and pressure, any fluid in contact with hot or cold solid surface tends to rise or fall along that wall, respectively. That phenomenon consequently generates convective currents in the fluid and involves heat transfer between fluid and wall [32]. Depending on the cause of convective currents, value of h_{cv} will depend whether the convection is natural or forced [33].

Regarding precisely heat transfers inside buildings, the convective heat transfer coefficient depends on the interior

environment of buildings [34], in particular, parameters such as the shape and dimensions of compartment, the distribution in temperature on the surface, the presence of air movements due to the existence of air currents or ventilation, and the surface roughness. In spite of the variability of case studies, that coefficient remains particularly difficult to determine. However, two alternative methods are mentioned in literature. The first method is to suppose h_{cv} constant according to values determined by Inard (1988) [35]. The second one is to estimate h_{cv} owing to experimental correlations. Concerning experimental correlations, many works exist in literature [36-39], but the majority expresses the convection coefficient as a function of the difference between the fluid temperature and the wall temperature. Regarding compartment fires, that difference in temperature is not so important. What is rather observed in compartment fires is the strong motion of hot gases. That explains the reason why during compartment fires, the convection heat transfer coefficient could be estimated using the McAdams correlation given by (8) [40]. That correlation takes into account the speed of the fluid V_f and the nature of the wall defined by characteristics m and n given in Table 4.

$$h_{cv} = 5.678 \left[m + n \left(\frac{V_f}{0.3048} \right) \right].$$
(8)

3. Results and Discussion

3.1. Experimental Results. After preliminary steps consisting of the preparation and instrumentation of the experimental domain, fire scenes were each performed in its configuration. Figures 4(a)-4(d) show the captured image of flame of fire sources 1, 2, 3, and 4, respectively. According to a visual



FIGURE 5: Illustration of the repeatability of the experimented fire scenes. (a) Scene 1 ($|\text{Error}| \le 3.99K$). (b) Scene 2 ($|\text{Error}| \le 7.22K$). (c) Scene 3 ($|\text{Error}| \le 8.76K$). (d) Scene 4 ($|\text{Error}| \le 8.37K$).

TABLE 5: Results of experimental tests including the mass loss rate of fuel, higher temperature of hot gases, and the heat release rate of the fire source.

	$W_0 H_0^{3/2} (m^{5/2})$	\dot{m}_f (g.s ⁻¹)	$T_{\rm max}$ (K)	Ċ (kW)
Scene n°1	0.051	0.1165	480	4.14
Scene n°2	0.038	0.1241	505	4.50
Scene n°3	0.025	0.1341	540	4.82
Scene n°4	0.019	0.1218	580	4.38

point of view, it can be observed that light emitted from flames decreases while the ventilation factor is reduced. One also observes an increase of the flame instability. In other words, as the ventilation factor decreases, the flame becomes increasingly turbulent and production of soot particles also increases. These observed effects are obviously due to the progressive lack in oxygen. Having repeated each fire scene three times, Figures 5(a)–5(d) show the variation curve over time of burned gases temperature at height h = 0.37 m (measured by sensor TC6). In that figure presenting curves of the three repeated tests as well as their mean curve, the three phases of compartment fire can clearly be identified (growth: from 0 to 200 sec, full development: from 200 to 600 sec, and decline: from 600 to 700 sec). It also appears that



FIGURE 6: Histogram of the variation of the mass loss rate of fuel over ventilation factor.



FIGURE 7: Variation over time of the temperature profiles of hot gases at ceiling.

the absolute error between the average and repetition curves is less than 10 K regardless of the fire scene. That allows approving the repeatability of each fire scene.

Exploitation of data collected during experiments enabled determining average values of parameters such as the mass loss rate \dot{m}_f of fuel, the maximum temperature T_{max} reached by burned gases at ceiling, and the thermal power released \dot{Q} by the fire source (Table 5). Graphical representation of these data shows regarding the rate consumption of fuel (Figure 6) that as the ventilation factor decreases, the mass loss rate of fuel increases linearly till a peak value of 0.029 m^{2.5}. Starting that peak value, one observes a gradual drop of mass loss rate. Indeed, the literature [41] distinguishes two fire control regimes for compartment fire: the fire controlled by fuel and the fire controlled by ventilation. In the present case study, the



FIGURE 8: Temperature fields of flame at plane x = 0.2 during the full development phase (at time t = 400 sec). (a) Fire scene 1. (b) Fire scene 2. (c) Fire scene 3. (d) Fire scene 4.

change over the ventilation level of the direction of variation of the mass loss rate traduces the change of the fire regime. Two variation domains could be highlighted in Figure 6. The first domain is the one where the ventilation factor is less than $0.029 \text{ m}^{2.5}$. It corresponds to the range where the flow rate of fuel increases strongly with the ventilation factor. The second domain is the one where the ventilation factor is more than $0.029m^{2.5}$. It corresponds to the range where the flow rate of fuel gradually decreases while the ventilation factor increases.

Indeed, the first domain corresponds to the regime where effects of ventilation on the behavior of fire dominate over effects of fuel. That means the fire is controlled by the ventilation because, in that interval, the increase of the ventilation factor involves the increase of the combustion rate fuel and therefore a rapid development of fire in the compartment. The second domain is the interval where the fire is controlled by fuel. In fact, as the ventilation factor increases, effects of fuel on the behavior of fire gradually take precedence on effects of ventilation. This is manifested by the decrease of the fuel mass loss rate. If the ventilation factor increases indefinitely, the flow rate of fuel will stabilize around a minimum value corresponding to the mass loss rate of the same fire source in an extremely large compartment or in free atmosphere. In other words, effects of ventilation on compartment fire become negligible as the ventilation factor increases. Concerning temperature of hot gases at ceiling, it was found that the change in the fire regime has no influence on it. It increases continuously while the ventilation factor decreases. This is illustrated in Figure 7, where it can be observed that the four fire scenes have the same temperature profile with different peak values.

3.2. Simulation of Tests. Numerical simulation consisted of reproducing the performed fire tests on a computer. Temperature projection on plane x = 0.2 and at a time t = 400 sec



FIGURE 9: Confrontation between experimental and numerical profiles at ceiling during growth and fully developed fire phase. (a) Fire scene 1. (b) Fire scene 2. (c) Fire scene 3. (d) Fire scene 4.

enabled visualizing the temperature contours at that plane of compartment. Thus, according to a visual analysis (Figures 8(a)-8(d)), one observes an increase in temperature of burned gases at the plume zone depending on the fire scenes. The smaller the opening width of the door is, the larger the flame size is. That is illustrated through the height of the plume.

Numerical results were compared to experimental results so as to validate simulations. Indeed, numerical curves giving evolution over time of burned gases temperature in the hot zone (h = 0.37 m) were compared to experimental curves (Figures 9(a)–9(d)). In these figures, the first two phases of the development of compartment fires, namely, growth phase (from 0 to 200 sec) and full development phase (from 200 sec to more), are distinguished. During the growth phase, it is observed that numerical curves do not agree with experimental curves. That difference observed is due to the fact that the average flow rate of fuel defined in the input file of the ISiS code is those of the full development phase. This means that the variation in the mass loss rate of fuel during growth phase was not taken into account. On the other hand, during that full development phase, simulations are agreed on with experiments. It then means that, during full development phase, simulations are in accordance with experiments. So, following that validation, numerical results obtained during that phase can be exploited and interpreted.

3.3. Simulation of Velocity Fields. During experiments, it was practically impossible to measure velocity of burned gases near the inner part of walls. This is due to the high thermal stresses inside compartment during fire. Such environment could be damageable for velocity captor. The other reason is that velocity is easily measurable with fluid flowing in one direction. This is not the case for burned gases at ceiling of the compartment in fire. That justifies the reason to use CFD tool in the present paper. Figures 10(a)-10(d) thus present evolutions over time of the velocity of burned gases plotted



FIGURE 10: Variation over time of the velocity fields of burned gases near wall. (a) Velocity field during scene 1. (b) Velocity field during scene 2. (c) Velocity field during scene 3. (d) Velocity field during scene 4.

	Small	scale	Full se	cale
	$W_0 H_0^{3/2}$ (m ^{2.5})	$\overline{V}_f (m \cdot s^{-1})$	$W_0 H_0^{3/2} (m^{2.5})$	$\overline{V}_f (m \cdot s^{-1})$
Scene n°1	0.051	0.260	2.83	0.519
Scene n°2	0.038	0.271	2.12	0.615
Scene n°3	0.025	0.275	1.42	0.606
Scene n°4	0.019	0.232	0.70	0.581

TABLE 6: Extrapolation of small-scale velocities into full-scale velocities.

near the wall of different simulated fire scenes. Regarding these curves, and precisely during the full development phases (comprised meanly between 200 and 500 sec), it is observed that velocities of burned gases near walls fluctuate around an average value that almost remains constant throughout the full development phase.

Numerical simulation enabled characterizing turbulent flow of gases in the hot zone. Manifestations of that turbulence can be observed through the frequency and the amplitude of velocity field which varies from one fire scene to another. Values velocity fields \overline{V}_f were extrapolated into the full-scale situation according to the Froude similarity [42] so as to estimate velocities of burned gas in a real fire situation (Table 6). Owing to the deduced full-scale velocities, the convective heat transfer coefficient of each fire scene has been determined, and the obtained values are

	Sm	Small scale		Full scale	
	$W_0 H_0^{3/2} \ (\mathrm{m}^{2.5})$	$h_{cv} (W \cdot m^{-2} \cdot K^{-1})$	$W_0 H_0^{3/2} \ (\mathrm{m}^{2.5})$	$h_{cv} (W \cdot m^{-2} \cdot \leq K^{-1})$	
Scene n°1	0.051	7.303	2.83	8.680	
Scene n°2	0.038	7.350	2.12	8.785	
Scene n°3	0.025	7.367	1.42	8.824	
Scene n°4	0.019	7.183	0.70	8.412	

TABLE 7: Extrapolation of the convective heat transfer coefficient into full-scale fire situation.



FIGURE 11: Variation of the convective heat transfer coefficient over the ventilation factor.

given in Table 7. Further investigation allowed noting that turbulent flow of hot gases in high part of compartment increases while the ventilation factor decreases. This means that the level of fluctuations of hot gases at ceiling of room in fire decreases while the ventilation level increases.

Graphical representation of convective coefficients deduced from numerical simulation allowed noting that the variation's direction changes according to the fire control regime. Indeed, when fire is controlled by ventilation, velocity of convective currents of burned gases in the hot zone increases with the ventilation factor, thus implying an increase in the convective heat transfer coefficient. That is for the case of fires with a very small ventilation factor such as confined fires. However, as effects of ventilation begin fading, particularly with the increase of the ventilation factor, velocity of hot gases gradually decreases, leading to the decrease in the convective heat transfer coefficient between hot gases and walls. That variation changing over the ventilation factor is illustrated in Figure 11, where two variation zones, respectively, represented by item (A) and item (B), can be identified.

Indeed, the domain where the convective heat transfer coefficient increases with the ventilation factor (A) corresponds to fire tests in which the effects of ventilation are dominant. The fire regime is then the fuel-controlled fire and really refers to fires in nearly confined environments. On the other hand, the domain where the convective heat transfer coefficient decreases with the ventilation factor (B) corresponds to fire scenes during which effects of ventilation no longer dominate over effects of fuel. Here, the fire regime is the ventilation-controlled fire. It refers to cases of fires in semienclosed environments such as most building fires; this is because during fire, it at least one opening will exist which could be a door or a window supplying continuously the compartment in fresh air. For that category of fires, it can be denoted that the convective heat exchange coefficient weakly decreases while the ventilation factor increases. A regression of that decreasing curve allowed establishing the mathematical relation (with $R^2 = 0.93$) given by (9) and expressing the convective coefficient h_{cv} as a function of the ventilation factor $A_0 \sqrt{H_0}$:

$$h_{cv} = 8.9797 - 0.102A_0\sqrt{H_0}.$$
 (9)

Despite the fact that the convective heat transfer coefficient decreases while the level of ventilation increases, it remains nevertheless close to the average value $8.75W \cdot m^{-2} \cdot K^{-1}$ which is lightly higher than value $7.0W \cdot m^{-2} \cdot K^{-1}$ recommended in literature [43]. That new value of h_{cv} deducted from experimental and numerical studies could be used in the modeling of confined fires so as to better evaluate thermal balance during buildings fires.

4. Conclusion

Heat transfer between burned gases and walls during compartment fire depends strongly on the convective heat transfer coefficient, which is related to the convective velocity of burned gases in the compartment. Owing to experimental and CFD studies, the present paper aimed to investigate the behavior of burned gases inside room during fire situation. Varying the ventilation level of compartment, temperature, velocity, and convective heat transfer coefficient of burned gases were analyzed. Results revealed that the convective heat transfer coefficient of gases in hot zone increases with the ventilation factor when the fire is piloted by ventilation effects and decreases progressively when effects of ventilation begin fading. However, that convective coefficient remains close to constant value. The maximal temperature of burned gases continuously increases while the ventilation factor of the room decreases.

Abbreviations

- L: Length of the room, m
- W: Width of the room, m
- H: Height of the room, m
- W_0 : Width of the opening, m
- H_0 : Height of the opening, m
- ρ : Densityk $g \cdot m^{-3}$
- u: Velocity of fluid $m \cdot s^{-1}$
- c_p : Specific heat of fluid $J \cdot kg^{-2} \cdot K^{-1}$
- λ : Thermal conductivity $W \cdot m^{-1} \cdot K^{-1}$
- h_{cv} : Convective heat exchange coefficient $W \cdot m^{-2} \cdot K^{-1}$
- P: Total pressure in the room, Pa
- τ : Tensor of the viscous stresses $N \cdot m^{-2}$
- Y_k : Mass fraction of chemical species k
- D_k : Molecular diffusion coefficient $m^2 \cdot mol^{-1} \cdot s^{-1}$
- q_r : Net heat flux of thermal radiation $J \cdot s^{-1}$
- χ : Efficiency of the combustion reaction
- H_c : Combustion heat of fuelk $J \cdot kg^{-1}$
- \dot{m}_f : Mass loss rate of fuelk $g \cdot s^{-1}$
- $\dot{m}_{f}^{''}$: Mass loss rate per unit area of fuel surface $kg \cdot m^{-2} \cdot s^{-1}$
- A_f : Horizontal area of the fuel pan, m²
- V_f : Velocity of the fluid near the wall $m \cdot s^{-1}$
- \overline{V}_{f} : Mean velocity of hot gas near the wall $m \cdot s^{-1}$.

Data Availability

No data were used to support this study.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this article.

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