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# Effect of cavity parameters on the fire dynamics of ventilated façades

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## Abstract

Narrow gaps and cavities present in modern construction systems have been identified as one of the key elements that may enable fire spread after an initial fire ignition. Although there are existing models to predict the exposure of the inner linings of a cavity when exposed to a fire, these models have a significant amount of error or are limited to small ranges of application. A parametric experimental study was performed to characterise the fire dynamics in a cavity using a medium-scale non-combustible parallel wall testing rig. Three different cavity widths, three different heat release rates and two air entrainment conditions at the base of the setup (i.e. closed, with no air entrainment at the base; open, with unrestricted air entrainment at the bottom) were varied. Measurements of the flame height and the total external heat flux on the cavity walls were performed. It has been shown that both the heat fluxes and the flame heights increased as the cavity width was reduced. It was found that the radiative component dominates the heat transfer at the bottom of the setup and is less relevant as the height increases, the cavity width increases and the burner output is reduced. Correlations were developed to quantify the heat exposure of a cavity as a function of the geometry of the system, the size of the fire and the ventilation condition. The results obtained can be further used as a baseline in the modelling of upward flame spread in confined spaces featuring combustible linings, and for assessing the potential risks of ignition of combustible elements in ventilated façade systems.

**Keywords:** fire safety engineering, fire dynamics, ventilated façade, heat flux, flame height.

## 1. Introduction & Background

Air cavities in façade systems are a common feature of the modern built environment. This void left between two façade elements allows drainage of moisture and acts as a ventilation for the system [1]. However, the presence of an air cavity poses multiple challenges to the fire safety strategy of high-rise buildings. This is because cavities have been identified as a key mechanism of fire spread after an initial fire event, serving as a path for smoke and fire to spread between multiple storeys [2]. This phenomenon was exhibited in previous high rise building incidents, such as the Grenfell Tower fire (2017) where cavity barriers were often either missing or not installed correctly, a fact which may have played a role in enabling the vertical spread of the flame through the cavity between the external cladding and the insulation in the façade system, as well as through small gaps present in the construction environment [3]. Other incidents involving ventilated façades where a cavity could have contributed to the fire spread have been previously reviewed [4].

The current knowledge of the fire behaviour of complex flammable façade systems is extremely limited [5]. Façades in high rise buildings are complex systems which implies the interaction of a number of phenomena that define the fire performance and fire safety outcomes of the façade. It is therefore essential to characterise the fundamental mechanisms that govern the fire response of the system to enable an analysis of the performance of these systems in fire. This requires a characterisation of the effect of varying different attributes of the cavity to provide tools that enable engineers to quantify such performance. This characterisation could be later combined with the available knowledge on the behaviour of different materials comprising the façade system [6].

### 1.1. Research significance

There is a dearth of knowledge on the vertical flame spread in confined spaces, although it has been identified as a common mechanism for fast fire growth [7]. Fully characterising fire spread

50 through combustible materials in such a configuration requires an understanding of the effect of  
51 the system geometry and knowledge on the parameters that govern the underlying fire dynamics  
52 in these systems. The study of fire dynamics in facades with non-combustible linings will shed  
53 light on how the configuration determines the incident heat flux in the cavity walls. Understanding  
54 and being able to predict the heat flux from fire plumes onto the surfaces will then allow a future  
55 assessment of the ignition and flame spread on combustible materials in these systems [8].  
56 Determining the distribution of the total external heat flux on the cavity walls is relevant since the  
57 energy transferred to the solid governs both the ignition and fire spread over combustible  
58 materials. These findings are intended to fit in a bottom-up approach fire safety engineers the  
59 tools that allow them to quantify the fire performance of a façade system.

60 The existing literature on cavity fires has demonstrated the relevance of the distance between the  
61 parallel walls as the governing geometrical parameter, as well as the intensity of the fire to explain  
62 the dynamics in a cavity fire. Although the differences between existing studies have been  
63 commented, there is a need to reconcile the existing results and to find the additional variables  
64 that might govern the fire dynamics in cavities.

### 65 *1.2. Aim and objectives*

66 This paper aims to fully characterise the fire dynamics in a non-combustible parallel wall cavity  
67 and explain the differences in behaviour with other non-combustible systems with similar  
68 cavities. A medium-scale non-combustible façade system was built to execute a parametric study  
69 that sheds light, and allow an investigation of the effect of different variables, on the fire dynamics  
70 of the system . The project was developed to quantify the effect of the size of the cavity, the  
71 energy source and the air entrainment on the incident heat flux and the flame heights of different  
72 enclosed fires. This will provide the necessary baseline for further research to characterise the  
73 flame spread over combustible cladding materials at an intermediate scale.

## 74 **2. Literature review**

### 75 *2.1. Nondimensional groups for a free burning flame*

76 A cavity comprises two surfaces which confine the flame in the event of a fire. To understand the  
77 influence of this geometric effect, it is important to first describe the behaviour of an unbounded  
78 fire plume. The geometry of turbulent diffusion flames has been found to scale with the square  
79 root of the Froude number [9]. The Froude number quantifies the ratio between the momentum  
80 forces and the gravitational or buoyancy forces that dictate the shape of a free burning fire (see  
81 Eq. 1)

$$Fr = \frac{u^2}{g \cdot D^2} \quad (1)$$

82 where  $u$  is the gas velocity,  $g$  is the acceleration due to gravity and  $D$  is the diameter of the flow  
83 source. Alternatively, the Froude number can be expressed in terms of the heat release rate (HRR  
84 or  $Q$ ) of the fire and its diameter. The heat release rate can be equated in terms or the burning rate  
85 ( $\dot{m}$ ) and the heat of combustion ( $\Delta H_c$ ) as  $Q = \dot{m}\Delta H_c$ . Further, the burning rate can be expressed  
86 as  $\dot{m} = \rho u$ . Hence, a dimensionless analysis for a circular burner of diameter  $D$  leads to the  
87 following relationship between Fr, the energy release rate, and the diameter of the source.

$$Fr \propto \frac{Q^2}{D^5} \quad (2)$$

88 The same analysis for a rectangular burner of width  $W_b$  and length  $L_b$  leads to a relationship of  
89 proportionality between the heat release rate per unit length of burner ( $Q'$ ), the burner width and  
90 Fr.

$$Fr \propto \frac{Q^2}{W_b^3 \cdot L_b^2} \propto \frac{Q'^2}{W_b^3} \quad (3)$$

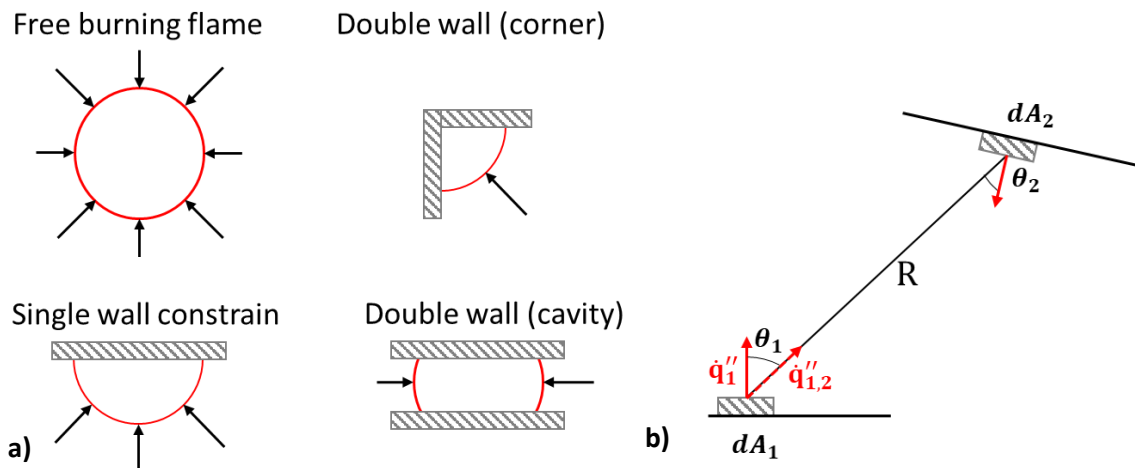
91 The flame geometry can be expressed as the flame height ( $L_f$ ) normalized by the source width  
 92 (see Eq. 4).

$$\frac{L_f}{W_b} \propto \frac{L_f}{Q'^{\frac{2}{3}}} \quad (4)$$

93 A further factor that influences the shape and behaviour of the fire plume is the geometry of the  
 94 burner (i.e. circular vs rectangular, aspect ratio of the rectangular burner). Hasemi and Nishihata  
 95 determined that there was a significant influence of the fuel shape on flame height and temperature  
 96 distribution in the plume above the flame, obtaining the shortest flames for a square burner and  
 97 the longest flames with burners with a large length to width ratio [10]. Other factors influencing  
 98 the behaviour of the plume include the type of burner used (sand bed vs gas) and the presence of  
 99 walls that confine the fire.

### 2.2. Interaction of the fire plume with walls and cavities

100 The presence of solid surfaces near a flame reduced the area through which air may be entrained  
 101 across the plume (see Figure 1 a). Previous research has investigated the effect of the presence of  
 102 one or more walls on the flame height [11]–[13]. Takahashi *et al.* studied the effect of a corner  
 103 wall on flame height (see Figure 1 a) and found flames were taller compared with free boundary  
 104 fire [11]. Sugawa *et al.* described the characteristics of flame geometry obtained experimentally  
 105 from multiple fire sources as a function of the heat release rate and the mixing factor for the  
 106 flames. The authors observed this factor controls the flame height due to the air entrainment  
 107 effects. Also, a larger dimensionless flame height was obtained for a burner placed next to a wall  
 108 compared to a free burning flame coming from a square burner [12]. Hu *et al.* found that flame  
 109 height decreases with an increase in the separation between the two walls comprising a cavity and  
 110 that it approaches the value of an unconstrained flame when the cavity width is beyond a critical  
 111 value, when the shorter side of the burner is perpendicular to the side walls. The authors also  
 112 found that the air entrainment into the buoyant non-premixed flames for line burners happens  
 113 mostly from the longer side and rarely from the shorter side of the burner [13].  
 114



115  
 116 **Figure 1.** a) Change of air entrainment to the fire plume caused by different configurations b) View factor between two  
 117 infinitesimal surfaces.

118 Besides the flame height, the temperature of the plume and the external heat flux on the surfaces  
 119 changes as the geometry of the system is modified. For example, the presence of a second wall  
 120 modifies the air flow pattern, affecting the radiative and convective heat transfer compared to a

121 single wall scenario by providing thermal exchange between the two surfaces [14]. The rate of  
122 heat transfer between two surfaces, from surface 1 to 2, can be expressed as:

$$\dot{q}_{1,2} = F_{1,2}A_1\varepsilon_1\sigma(T_1^4 - T_2^4) \quad (5)$$

123 where,  $\varepsilon_1$  is the emissivity of surface 1,  $A_1$  is the area of surface 1,  $\sigma$  is the Boltzmann constant  
124 and  $T_1$  is the temperature of surface 1. The term  $F_{1,2}$  is noted as the integrated configuration factor  
125 or view factor and can be expressed as:

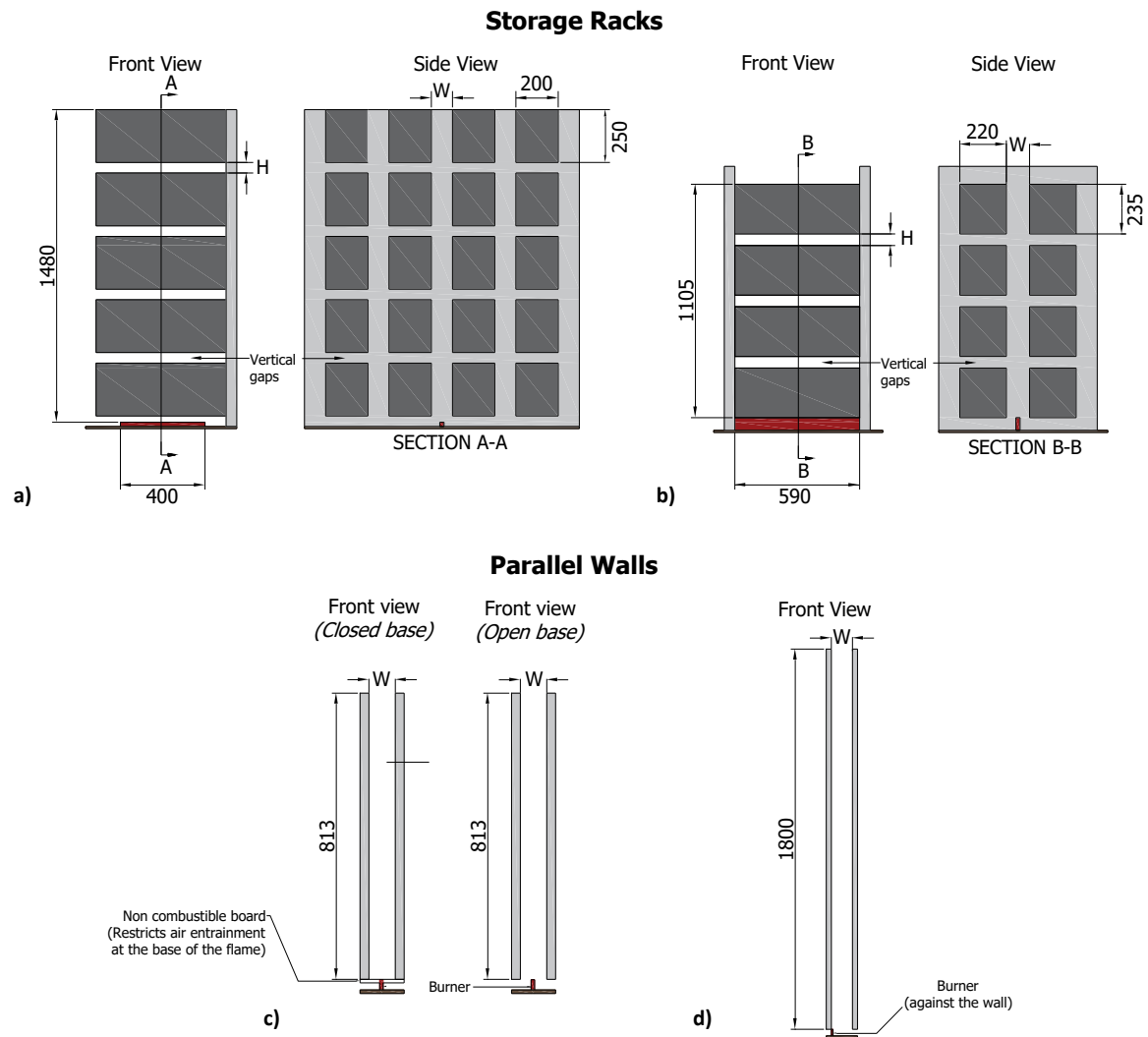
$$F_{1,2} = \frac{1}{A_1} \int_{A_1} \int_{A_2} \frac{\cos\theta_1 \cos\theta_2}{\pi R^2} dA_1 dA_2 \quad (6)$$

126 Where R is the shortest distance between the centre points of the surfaces and  $\theta$  is the angle  
127 formed between the vector normal to the surface and the segment R (see Figure 1). This indicates  
128 that the view factor (and hence the radiative heat exchange) diminishes as the distance between  
129 the surfaces increases. Furthermore, regions located further from the centreline of the surfaces  
130 will have smaller view factors.

131 Additional experimental studies have been performed on the effect on the temperature distribution  
132 above the burner [10], [15], and the heat transfer to the surfaces involved [16], [17]. Hasemi and  
133 Tokunaga [15] measured the temperature distribution on the fire plumes from square burners in a  
134 semi-infinite space (free plume), against a wall and in a corner wall configuration. The decrease  
135 of the difference between the plume temperature and the ambient temperature was slower for the  
136 case with a wall compared to the free burning flame. Back *et al.* measured the incident heat flux  
137 distributions on a wall placed adjacent to a propane burner fire. The heat release rate was found  
138 to have a strong impact on the peak heat fluxes [16]. Williamson *et al.* conducted several  
139 experiments to determine the effects of heat release rate and the position of the ignition source on  
140 the external heat flux distribution on lining materials in corner fire tests. The authors determined  
141 that relatively small differences in the ignition source stand-off distance - such as 5 cm - had large  
142 effects on the exposure conditions in a room fire. It was determined that a flame impinging the  
143 wall led to the most onerous heat flux exposure [17]. If this finding is extrapolated to a cavity  
144 scenario, a burner producing a flame that impinges both walls would lead to a more onerous  
145 scenario compared to a burner located in the centre of the cavity where the walls are sufficiently  
146 far away that there is no flame impingement. This was experimentally shown by Foley, who  
147 compared the effect of the ignition source on the heat flux received by parallel walls. The study  
148 found that the heat flux to the wall was greater with the burner against that wall, compared to a  
149 fire in the centre of the cavity, away from the walls. Foley concluded that moving the burner away  
150 from the wall affected the external heat flux by both allowing the convective cooling of the wall  
151 and reducing the convective heating from the flame [18], [19]. Further experimental studies have  
152 been conducted to characterise the fire dynamics of a cavity, mainly by exploring the influence  
153 of the geometry of the system on the flame height and the heat transfer to cavity walls exposed to  
154 a flame.

### 155 2.3. Previous research on cavity fires

156 Most research on fire plumes in cavities to date has focused on fires confined to parallel walls  
157 [19]–[21] and rack storage systems [22]–[25]. Storage racks were comprised by a main vertical  
158 flue and gaps between the racks in the vertical direction, whereas parallel wall systems just feature  
159 a main cavity. A diagram of the different setups is presented in Figure 2. All of these systems  
160 were comprised of non-combustible materials. The main investigated outputs were the flame  
161 height, the gas and solid phase temperatures and the heat flux profile to the surfaces of the cavity  
162 between walls or the racks.



163

164 **Figure 2.** Schematic representation of systems with cavities. a) Storage rack, air entrainment restricted in one direction  
 165 [22]. b) Storage rack, air entrainment restricted in two directions [23]. c) Parallel walls, different base configurations  
 166 [19]. d) Parallel wall setup, air entrainment restricted in two directions, but not at the base[20].

167 Karlsson *et al.* developed a correlation between flame height ( $L_f$ ) and cavity width ( $W$ ), the  
 168 spacing between racks ( $H$ ), and the heat input per unit length of the burner ( $Q'$ ) (see Eq. 7) [22].  
 169 The correlation was confined to the setup configurations where  $W$  and  $H$  had an influence on the  
 170 flame height. No influence of the geometry and  $Q'$  was observed for  $W/Q'^{2/3} < 0.007$ . The main  
 171 purpose of this study was to model the behaviour of flames between storage racks (See Figure 2.  
 172 a). The experimental setup had vertical separations between the racks, in addition to the horizontal  
 173 separation between the obstructions. Additionally, three sides of the storage rack were open to the  
 174 environment.

$$\frac{L_f}{Q'^{2/3}} = 0.00242 \left( \frac{W}{Q'^{2/3}} \right)^{-0.496} \cdot \left( \frac{H}{Q'^{2/3}} \right)^{-0.07} ; \left( \frac{W}{Q'^{2/3}} \right) < 0.007 \quad (7)$$

175 Ingason used a similar experimental setup where two or three sides of the setup were closed in  
 176 order to restrict the air inflow. A model for the flame height in terms of the heat release rate of  
 177 the burner and the cavity width was proposed (see Eq.2) [23]. Additionally, Ingason developed a  
 178 theoretical model for predicting flame heights and temperature profiles in a two-dimensional rack  
 179 storage system (See Figure 2. b) [25]. Karlsson indicated that the correlations obtained by him

180 and Ingason were significantly different due to the minor differences in the setups (e.g. restriction  
 181 of air entrainment from 1 of the 4 available sides vs restriction in just 3 of the sides). Ingason  
 182 stated that the developed models could be applied to other geometries with no lateral openings.  
 183 The models were limited to the study of the effect of the geometry on the flame height, and no  
 184 systematic heat transfer measurements were made in this research.

$$L_f = 0.307 + 6.15 \cdot 10^{-4} \cdot \left(\frac{Q'}{W}\right) \quad (8)$$

185 Foley and Drysdale performed a comprehensive set of tests of fires within a cavity, where the  
 186 cavity width, the heat release rate of the burner, the burner location (against the wall and at the  
 187 centre of the cavity), and the air availability (closed and open base) were modified (See Figure 2.  
 188 c) [19]. The study showed that the availability of air influenced the flame shape as well as the  
 189 heat flux on the wall. Furthermore, it was demonstrated that the total heat flux to the inner wall  
 190 increases as the cavity width is reduced. Several empirical correlations were proposed for the  
 191 incident heat flux as a function of the cavity width ( $W$ ), the non-dimensional heat release rate per  
 192 unit length of burner ( $Q'^*$ ), first introduced by Hasemi [26], and the height across the wall,  
 193 depending on the configuration of the system. The correlations follow the format of Eq. 9.

$$\dot{q}'' = K_1 \cdot \left( z \cdot \frac{\left(\frac{W}{L_b}\right)^{K_2}}{Q'^{* \frac{2}{3}} \cdot L_b} \right)^{K_3} \quad (9)$$

194 where  $\dot{q}''$  is the total external heat flux on the wall,  $z$  is the height above the burner,  $L_b$  is the length  
 195 of the burner,  $Q'^*$  is the non-dimensional heat release rate per unit length of burner (see Eq. 10)  
 196 and  $K_1$ ,  $K_2$  and  $K_3$  are constants of the model that vary according to the conditions of the system.

$$Q'^* = \frac{Q'}{\rho_\infty \cdot C_p \cdot T_\infty \cdot g^{\frac{1}{2}} \cdot L_b^{\frac{3}{2}}} \quad (10)$$

197 Livkiss *et al.* [20] obtained a correlation for flame height within a cavity based on experimental  
 198 data for ventilated façades (See Figure 2. d). The flame heights were measured from 30 photos  
 199 taken with 1 s intervals after an arbitrary time of 130 s. The authors commented that uncertainties  
 200 and errors in the measurement could be attributed to factors like the view angle of the camera and  
 201 the frequency of the flame oscillations relative to the camera shutter speed, among others. The  
 202 flame height was compared to the studies mentioned before. The discrepancies in the flame height  
 203 were attributed to the difference in the experimental setups, but these were not quantified.  
 204 Furthermore, a semi-quantitative analysis of the incident heat flux on the wall as a function of the  
 205 cavity width and the burner heat release was proposed. It was found that the incident heat fluxes  
 206 increased as the cavity was made narrower, which matches the conclusion of Foley and Drysdale  
 207 [19]. Additional research was conducted by de Ris and Orloff on a fire burning between two  
 208 parallel plates. The authors measured flame heights and flame heat-flux distributions for a wide  
 209 range of fuels and determined that the heat flux from the flames is sensitive to the flame sootiness.  
 210 The distance between the panels was kept constant at 300 mm and hence, it was not considered  
 211 as a variable for a theoretical model for the heat flux distribution along the walls [21]. The  
 212 outcomes of this last study are not included in the following comparison since studying the effect  
 213 of the cavity width was one of the main objectives of the present research.

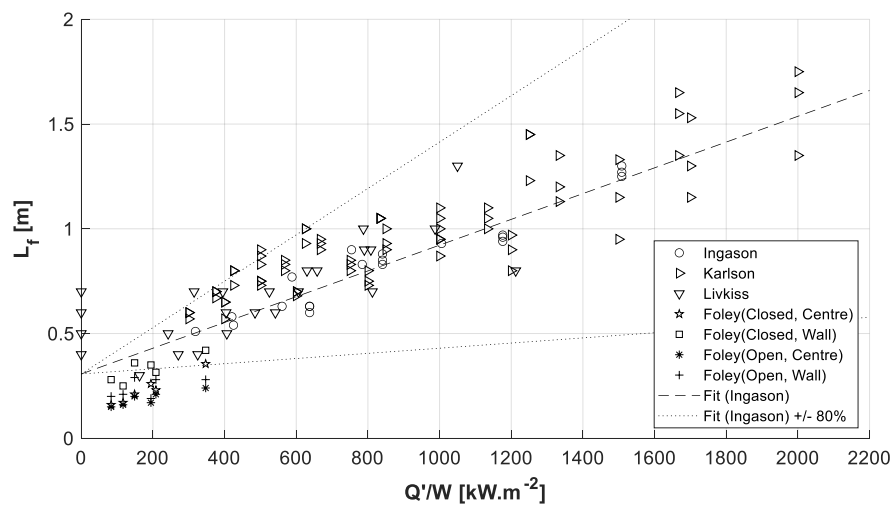
214 A comparison of the experimental setups and the range of the variables used in each of the studies  
 215 described is presented in Table 1.

**Table 1.** Experimental conditions for different authors that studied cavity flames.

	Base	Flame location	HRR per unit length of burner [kW.m <sup>-1</sup> ]	Cavity width (W) [m]	Vertical gaps width (H)[m]
<b>Foley &amp; Drysdale</b> [19]	Open	Centre of the cavity	11.6, 20.9	0.06, 0.1, one wall	N/A
	Closed	Near the wall			
<b>Karlsson et al.</b> [22]	Open	Centre of the cavity	60, 75, 85, 100, 125	0.05, 0.075, 0.1, 0.15	0.05, 0.1, 0.15
<b>Ingason</b> [23]–[25]	Open	Centre of the cavity	32, 42, 59, 75	0.05, 0.075, 0.1	0.05, 0.075, 0.1
<b>Livkiss et al.</b> [20]	Open	Near the wall	16.5, 24.8,	0.02, 0.03, 0.04, 0.05, 0.06, 0.1, one wall	N/A
			32.3, 40.4		
<b>Current study</b>	Open	Entire cavity	20, 35, 50, 74	0.05, 0.1, 0.15	N/A
	Closed				

217

218 The majority of these studies were not conducted with modern façade configurations in mind,  
 219 since burners with a fixed nozzle width were used, whereas in a real fire the flame may be  
 220 expected to fill the cavity or otherwise be proportional to the cavity width. Even if the behaviour  
 221 of the flame height was shown to follow similar trends as a function of the cavity width and heat  
 222 release rate, the models proposed for the flame height have not yet been reconciled between the  
 223 studies. This indicates the presence of extra variables that might influence the underlying fire  
 224 dynamics. A comparison of the flame height data against the correlation developed by Ingason  
 225 for flame height as a function of the heat release rate per unit area (Eq. 8) is presented in Figure  
 226 3. The equation proposed by Ingason cannot accurately predict the flame length for the different  
 227 setups. Hence, a further exploration of the variables of the system is necessary to find the  
 228 correlations that describe the fire dynamics for full-width burning cavities, as relevant to façades  
 229 today.



230

231 **Figure 3.** Experimental flame heights vs heat release rate per unit area from Livkiss *et al.*, Karlsson *et al.*, Ingason  
 232 and Foley [19], [20], [22], [23]. The correlation proposed by Ingason is presented as a dashed line.

233 Torero argued that cavity width has an important effect on the rate of flame spread. An extremely  
 234 large cavity will diminish radiative exchange and buoyancy-driven chimney effects, which will  
 235 generate a flame spread rate similar to the ones observed for a fire against an individual open  
 236 wall, i.e., when no cavity is present. Contrarily, an extremely narrow cavity will present a  
 237 blockage of the gas flow caused by the thermal expansion of the combustion gases. This will  
 238 ultimately cause the flame to cease to spread inside the cavity [27]. Some research has been  
 239 performed on cavities featuring combustible linings to determine the effect of the cavity on the

240 upward flame spread over the materials [28]–[31]. An extensive series of work has been  
 241 performed at FM Global using parallel plates setups to simulate the same radiative view factor as  
 242 in the corner configuration, ensuring an equivalent heat exposure [31]–[33]. This in order to create  
 243 a sufficiently strong exposure to evaluate full facade systems and predict behaviour in the large-  
 244 scale corner tests [32]. The work by Nam *et al.* provided an intermediate-scale testing  
 245 methodology as a screening tool to assess wall and ceiling assemblies for material flammability.  
 246 This research intended to deliver a testing methodology that allowed to predict full scale test  
 247 behaviour by using material properties determined at bench scale. Additional studies have been  
 248 performed to characterise the fire dynamics of ventilated façades featuring combustible elements  
 249 [34]–[36]. Garvey *et al.* proposed a methodology to isolate the effect of the individual materials  
 250 on the upward flame spread [34]. However, it was not possible to explore the effect of the cavity  
 251 width in any of these studies, since a fixed value was used for this parameter.

### 252 3. Methodology

253 The experimental setup consisted of two 600 mm wide, 1800 mm high, 25 mm thick, vermiculite  
 254 walls (Skamol V-1100 (375), see Table 2 for thermal properties), placed in parallel configuration  
 255 separated by an air gap, and mounted on an aluminium frame (see Figure 4. a) and b). The  
 256 dimensions of the non-combustible walls were set considering the dimension of experimental  
 257 setups used in previous studies [19], [20], [22], [23] and considering the standard size available  
 258 for construction materials.

259 One of the walls (denoted as Auxiliary wall in figure 2 a)) was mounted on movable elements  
 260 that allowed simple and rapid modification of the cavity air width for different experiments. The  
 261 non-combustible walls were replaced between the experiments after signs of deterioration were  
 262 detected e.g. cracks.

263 A sand methane burner was placed at the base of the parallel walls to generate fires with different  
 264 heat release rates per unit length. The heat released was controlled by using a Teledyne HFC-D-  
 265 303B mass flow controller. The sand was used to guarantee a uniform distribution of the fuel flow  
 266 and a uniform heat release rate per unit length of the burner. The width of the burner was modified  
 267 by partially covering the upper surface with a non-combustible material to obtain the desired  
 268 width to match the set cavity width. A rectangular burner with a modifiable width meant that it  
 269 was possible to have the flame present through the entire width of the cavity, instead of being  
 270 limited to a single fixed nozzle width for all tests, as in previous studies [19], [20], [22], [23].  
 271 This ensured that the flame impinged both cavity walls instead of a single lining thereby creating  
 272 a more onerous scenario [17]. This burner configuration ensures that the test conditions are  
 273 representative of a realistic scenario for a fire in ventilated façade compared to previous research.

274 The width of the cavity was also modified as one of the parameters of the tests. The length of the  
 275 burner was kept constant at 480 mm. It was set to be shorter than the wall width (600 mm) to  
 276 ensure that flames would not escape the cavity. The combinations of cavity width and HRR were  
 277 set so the flames did not extend above the top of the walls or outside from the edges of the walls.

278 **Table 2.** Temperature dependant thermo-physical properties of the non-combustible boards and TSC materials

Material	Thermal conductivity [W.m <sup>-1</sup> .K <sup>-1</sup> ]	Density [kg.m <sup>-3</sup> ]	Specific heat capacity [J.kg <sup>-1</sup> .K <sup>-1</sup> ]	Emissivity [-]
Vermiculite (SKAMOL V-1100(375) [37])	0.12 @ 200°C 0.15 @ 400°C 0.16 @ 600°C	375 @ 20°C	0.94	Not used

		0.19 @ 800°C			
Inconel	15 @ 200°C	7800	$450 + 0.28T - 2.91 \cdot 10^{-4}T^2 + 1.34 \cdot 10^{-7}T^3$	0.44 to 0.36	
	18 @ 400°C	@ 20 < T < 1000 °C	@ 20 < T < 1000 °C	@ 216 < T < 490°C	
		(T in °C)	(T in °C)		

279 Two arrangements were introduced to study the effect of the air entrainment on the behaviour of  
 280 the fire (see Figure 4. c). In one of them, the vermiculite boards were located in contact with the  
 281 burner to prevent air entrainment at the base, this configuration will be called “closed base”. In  
 282 another configuration , the experimental rig was lifted up 80 mm from the position in the closed  
 283 base configuration in order to allow the air entrainment from the bottom of the walls, this  
 284 configuration is denoted “open base”. As the opening size was not varied the influence of this  
 285 parameter – beyond the presence or absence of an 80 mm opening - on the fire dynamics cannot  
 286 be determined from the results of these tests . The experimental configurations which were  
 287 investigated are presented in Table 3.

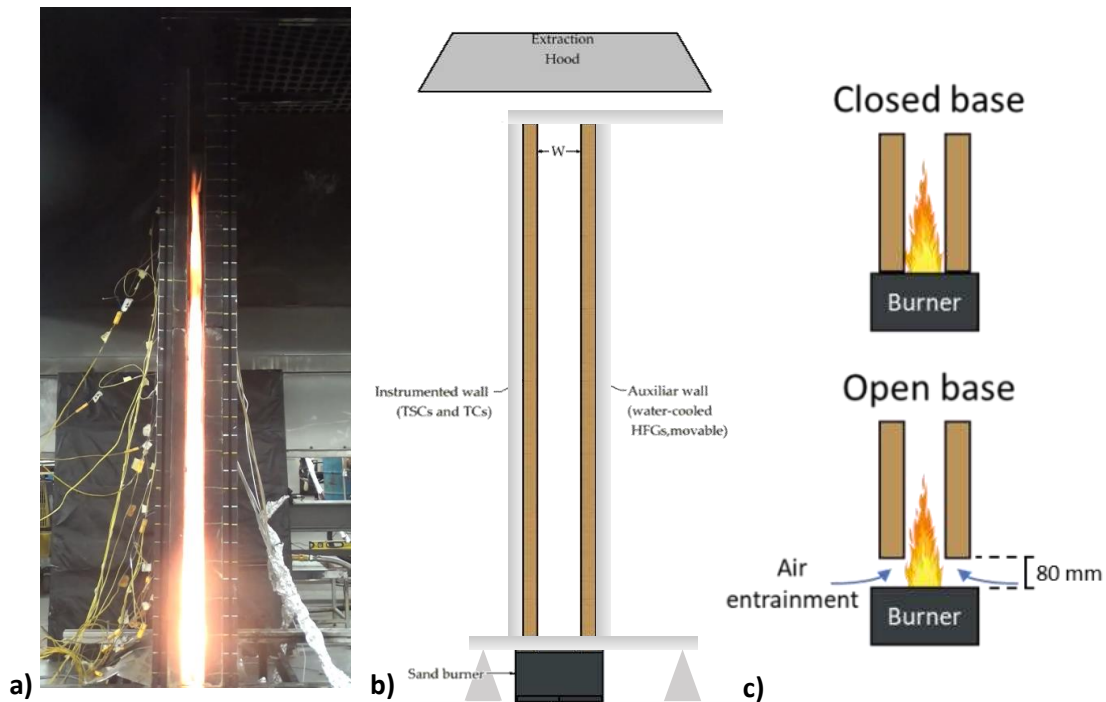
288 For each cavity width, three burner heat release rates were tested sequentially one after another  
 289 within the same experiment. This was to increase the number of tests that could be performed  
 290 rather than waiting for the apparatus to cool between each test which would be highly time-  
 291 consuming. For each heat release rate, the test was run until a quasi-steady state was achieved as  
 292 measured by the TSCs, which took a minimum of 10 minutes and up to 15 minutes. The data was  
 293 then extracted from a 90 s window within those steady state conditions, and those are the results  
 294 presented in this manuscript. Every test was carried out twice in order to demonstrate the  
 295 repeatability of the flame height and external heat flux when following the same experimental  
 296 procedure.

297

**Table 3.** Experimental campaign conditions.

Test	Cavity width (W) [mm]	Q' (Nominal) [kW.m <sup>-1</sup> ]	Q' (Real) [kW.m <sup>-1</sup> ]	Base configuration
C1	50	20, 35, 50	20.8,36.4,51.56	Closed
C2	50	20, 35, 50	20.8,36.4,52.03	Closed
C3	100	20, 35, 50	20.8,36.4,52.03	Closed
C4	100	20, 35, 50	20.8,36.4,52.06	Closed
C5	150	35, 50, 74	36.41,52.06,79.73	Closed
C6	150	35, 50, 74	36.46,52.06,77.37	Closed
O1	50	20, 35, 50	20.8,36.5,52.06	Open
O2	50	20, 35, 50	20.8,36.6,52.03	Open
O3	100	20, 35, 50	20.8,36.6,52.12	Open
O4	100	20, 35, 50	20.8,36.5,52.08	Open
O5	150	35, 50, 74	36.44,52.05,77.25	Open
O6	150	35, 50, 74	36.44,52.05,78.32	Open

298



299

300 **Figure 4.** a) Image of the experimental setup. B) Schematic representation of the main components of the experimental  
 301 setups. c) Base configurations for air entrainment. The blue arrows represent air entrainment at the base of the walls.

302 The burner heat release rate was calculated as the product of the mass flow rate of the natural gas  
 303 and its effective heat of combustion (approximately  $52.5 \cdot 10^3 \text{ kJ} \cdot \text{kg}^{-1}$ ). Additionally, the testing  
 304 rig was placed under an extraction hood in order to verify the heat release rate of the burner using  
 305 the species evolution approach (both Oxygen consumption [38] and Carbon Dioxide Generation  
 306 calorimetry [39]). All the tests were recorded with a video camera, located at a height of 1.3 m  
 307 and a distance of 2 m from the setup (see Figure 5. e), in order to extract the visual flame height  
 308 of the fire generated by the burner.

### 309 3.1. Flame height determination

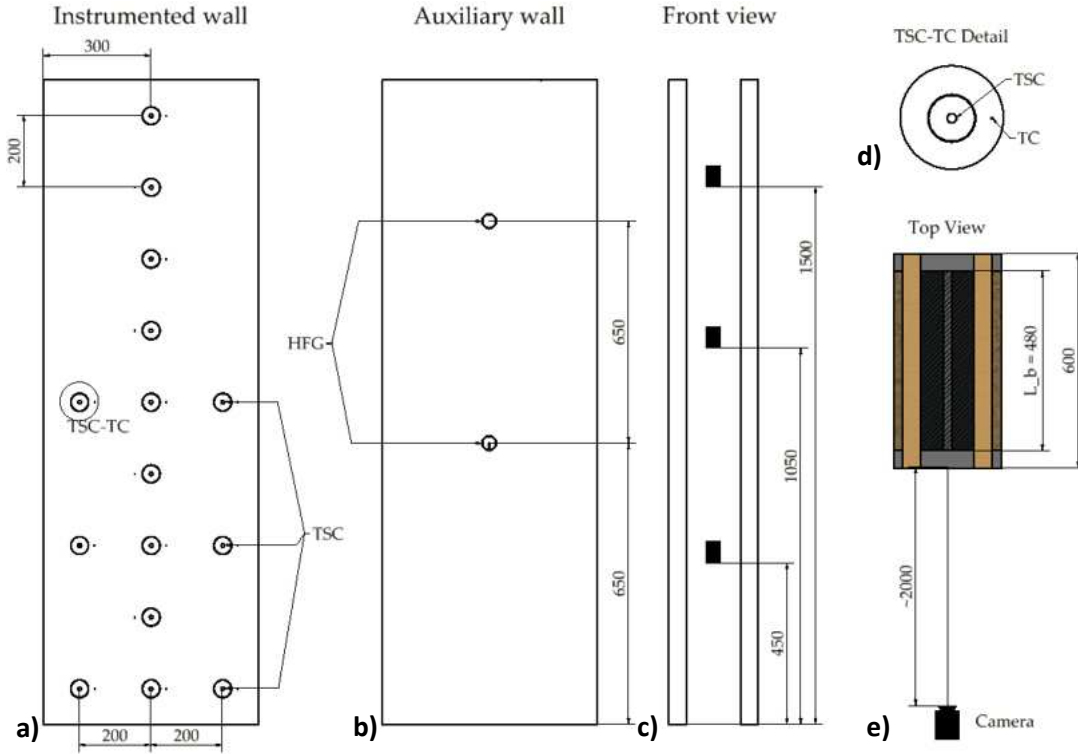
310 Flame heights were measured as the tip of the continuous flame from 2250 photos corresponding  
 311 to all the frames of the 90 s span. A height reference was incorporated in the walls of the setup in  
 312 order to calibrate the height of the camera. The flame height was initially calculated as the average  
 313 of the measurements and then corrected to account for the view angle for the camera. To do so,  
 314 the location of the flame tip was assumed to be in the middle of the burner total length. This is  
 315 supported by observations made during the experimental campaign and from the external heat  
 316 flux distribution on the walls (see Section 5). In order to verify the reliability of the flame height  
 317 data, a repeatability test was performed to measure the relative deviation of the flame height for  
 318 the sets of experiments (i.e. a test and its repetition).

### 319 3.2. Heat flux measurement

320 One of the walls was instrumented with 15 thin skin calorimeters (TSCs) to measure solid phase  
 321 temperature used in the calculation of the incident radiant heat flux. These were made of 10 mm  
 322 diameter, 1.3 mm thickness Inconel discs mounted on 50 mm diameter vermiculite cores inserted  
 323 in the wall. Additionally, 15 gas-phase thermocouples (TCs, 1.5 mm diameter, type K mineral-  
 324 insulated metal-sheathed (MIMS)) were installed in this wall to measure the gas-phase  
 325 temperature at vicinity of the solid surface. These measurements were used in conjunction with  
 326 the temperature of the TSCs to calculate the radiative incident heat flux on the instrumented wall  
 327 ( $\dot{q}''_r$ ). The position of the instrumentation is depicted in Figure 5. The opposite wall was

328 instrumented with two Hukseflux water cooled heat flux sensors at heights of 0.65 m and 1.3 m  
 329 above the edge of the wall. The measurements from the heat flux gauges were compared against  
 330 the trends for the calculated total external heat fluxes, obtained from the TSC and TC  
 331 measurements (see Heat flux calculation). This comparison is made to validate the precision of  
 332 the TSC results and to be able to compare the magnitude of the total external heat fluxes in the  
 333 wall with those obtained by Foley [19], who used water-cooled heat flux gauges for the  
 334 measurements. The heat flux measurements reported correspond to the average of 90  
 335 measurements in a steady state, corresponding to the same time span when the flame heights were  
 336 determined.

337



338

339 **Figure 5.** a) Location of the Thin skin calorimeters and gas phase thermocouples in the instrumented wall. b) Heat  
 340 flux gauges in the Auxiliary wall c) Front view and location of the velocity probes. d) Detailed position of the gas  
 341 phase thermocouples. e) Camera positioning from top view.

### 342 3.3. Heat flux calculation

343 Heat transfer from flames is mainly attained by radiation and convection. The relative significance  
 344 of these heat transfer mechanisms under different scenarios is of interest, both to increase  
 345 understanding of heat transfer from flames to the linings of a cavity and to suggest appropriate  
 346 fire protection measures. The total external heat flux ( $\dot{q}_T''$ ) on the wall was defined as the sum of  
 347 a radiative and a convective component:

$$\dot{q}_T'' = \dot{q}_r'' + \dot{q}_c'' \quad (11)$$

348 The convective component was defined as:

$$\dot{q}_c'' = h_c(T_s - T_{gas}) \quad (12)$$

349 where  $T_{gas}$  is the temperature of the gas in the cavity measured with the gas-phase thermocouples,  
 350  $T_s$  is the solid temperature measured by the TSC), and  $h_c$  is the convective heat transfer  
 351 coefficient. The convective heat transfer coefficient ( $h_c$ ) was calculated using Eq.13.

$$h_c = k_g \left( \frac{Nu}{L_c} \right) \quad (13)$$

352 where  $k_g$  is the gas conductivity and  $L_c$  is the characteristic length. The Nusselt number was  
 353 calculated using two different approximations. One for the natural convection regime ( $Nu_{NC}$ , see  
 354 Eq. 14) [40], [41] and for the forced convection regime ( $Nu_{FC}$ , see Eq. 15) [42].

$$Nu_{NC} = 0.68 + \frac{0.67Ra^{\frac{1}{4}}}{\left( 1 + \left( \frac{0.492}{Pr} \right)^{\frac{9}{16}} \right)^{\frac{4}{9}}} \quad (14)$$

$$Nu_{FC} = 0.037Re^{\frac{4}{5}}Pr^{\frac{1}{3}} \quad (15)$$

355 Where the Reynolds number ( $Re$ ) was calculated from the velocities obtained using bidirectional  
 356 velocity probes with a 0.015 m diameter. The three velocity probes were located in the centreline  
 357 of the wall and in an equidistant point between the walls. The probes were located 450, 1050 and  
 358 1550 mm above the burner (see Figure 5. d).

359 It was estimated that using either of the approximations would lead to a deviation not higher than  
 360 1 kW.m<sup>-2</sup> in the convective heat flux component, which is negligible considering the order of  
 361 magnitude of the incident radiant heat flux (see Eq. 16). Hence, the natural convection  
 362 approximation was used, since it provided a greater number of locations for the characterisation  
 363 of the convective and total convective heat flux [43].

364 The radiative heat transfer flux to the wall was then calculated using the methodology proposed  
 365 by Hidalgo *et al* [44], as described by Eq.16:

$$\dot{q}_r'' = \frac{1}{\alpha_{TSC} \cdot (1 - C)} \left[ \rho_{TSC} \cdot \delta_{TSC} \cdot C_{pTSC} \cdot \frac{dT_s}{dt} + \varepsilon_{TSC} \cdot \sigma \cdot T_s^4 + h_c \cdot (T_s - T_{gas}) \right] \quad (16)$$

366 where  $\alpha_{TSC}$  is the absorptivity of the TSC metal disc,  $C$  is a correction factor for the heat transfer  
 367 by conduction,  $\rho_{TSC}$  is the density of the TSC metal disc,  $\delta_{TSC}$  is the thickness of the disc,  $C_{pTSC}$   
 368 is the specific heat capacity of the disc,  $\varepsilon_{TSC}$  is the emissivity of the disc,  $\sigma$  is the Stefan-  
 369 Boltzmann constant,  $T_s$  is the solid-phase temperature measured by the TSC.

370 For the measurement of the incident radiant heat flux in a semi-steady state it can be assumed that  
 371 the term for the variation of the temperature of the solid can be approximated to 0 (see Eq. 18)  
 372 and the energy stored by the metal disc can be neglected.

$$\frac{dT_s}{dt} \approx 0 \quad (17)$$

373

#### 374 4. Flame height

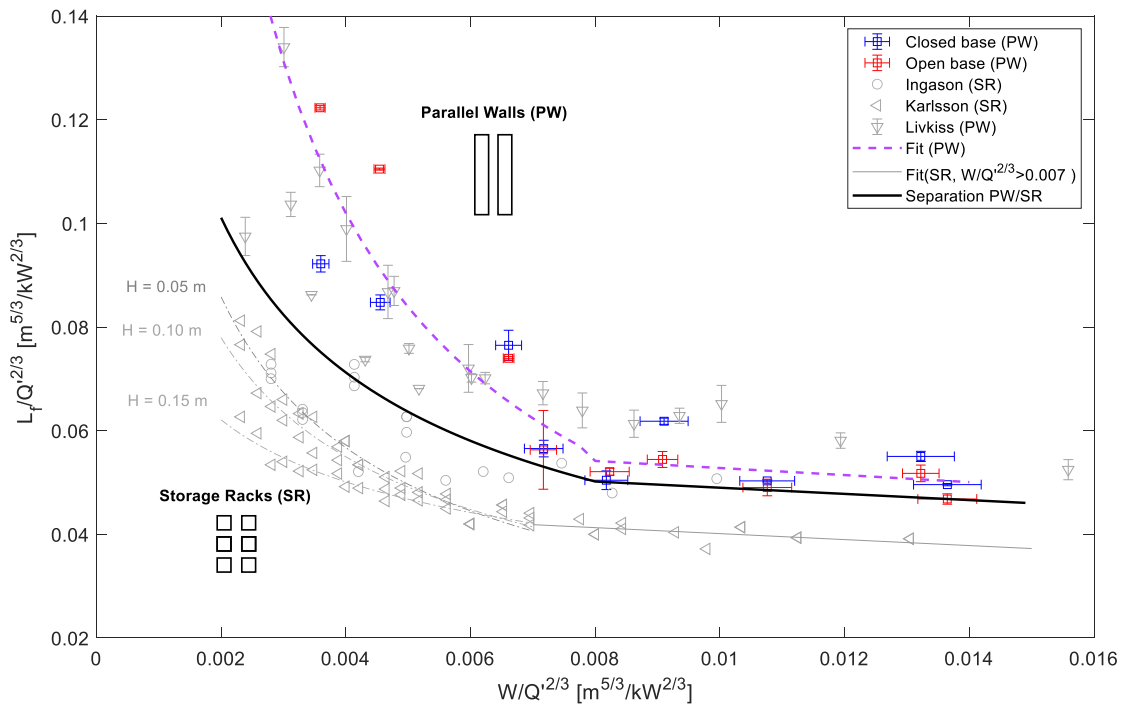
375 The results of the average flame tip heights are shown in Figure 6 as a function of the cavity  
 376 width. These parameters are both normalised by the heat release rate as part of a dimensional  
 377 analysis for rectangular gas burners based on the work developed by Thomas *et al.* [45] and later  
 378 used by Quintiere *et al.* [46]. This takes the dimensionless Froude number as the base to compare  
 379 burners with different sizes and heat release rates. It is clear that for both open and closed based  
 380 configurations, the flame height dramatically increases with narrower cavity widths and large  
 381 HRRs. This aligns with previous findings by Karlsson *et al.* [22], who generated a power law  
 382 model (see Eq. 7) and hinted to the existence of two different regimes, one where the flame height

383 is drastically influenced by the cavity width and heat release rate per unit length burner and one  
 384 where the influence is less significant. Two different regions were defined for this study  
 385 considering that the flame height drastically increases when  $W/Q^{2/3} < 0.008$  (narrower cavity  
 386 widths and large HRRs), whereas the effect is less evident pronounced for the rest of the cases.  
 387 A piecewise function was therefore proposed to predict the normalised flame height (Eq. 18 and  
 388 Eq. 19).

$$L_f/Q^{2/3} = 8.0 \cdot 10^{-4} \left( W/Q^{2/3} \right)^{-0.878}; \quad W/Q^{2/3} < 0.008 \quad (18)$$

$$L_f/Q^{2/3} = -0.68 (W/Q^{2/3}) + 5.96 \cdot 10^{-2}; \quad W/Q^{2/3} > 0.008 \quad (19)$$

389 A significant difference is observed between the open and closed base configurations when the  
 390 narrowest cavity and the two highest heat release rates were combined ( $W/Q^{2/3} < 0.006$ ). In those  
 391 cases, the open base configuration led to higher flames. This might indicate that air entrainment  
 392 from the base helps push the flame towards the centre of the cavity, away from the walls. The  
 393 influence of air entrainment on the elongation of the flame does not seem to be significant for the  
 394 rest of the configurations. The results of the current study are compared with previous studies  
 395 developed by Livkiss *et al.* [20], Ingason [23] and Karlsson *et al.* [22]. The nondimensional flame  
 396 heights for the study by Foley and Drysdale are not included since the intention of that research  
 397 was not to characterise the flame height but the heat transfer [18], [19]. In all cases, the general  
 398 trend is for flame length elongation as either the cavity width is reduced or as the burner heat  
 399 release rate is increased. This effect is most pronounced at small cavity widths, and a seemingly  
 400 asymptotical value is reached as the setup tends to open burning at very large cavity widths. This  
 401 similarity indicates that the behaviour of the cavity flame is controlled by similar physical  
 402 phenomena in spite of the experimental setup differences.



403

404 **Figure 6.** Average normalised flame heights ( $L/Q^{2/3}$ ) from this study and the studies by Livkiss *et al.*, Karlsson *et*  
 405 *al.*, and Ingason [20], [22], [23]. Error bars correspond to the minimum and maximum point from the repetitions for  
 406 each experiment.

407 The trend in all the data is similar, and the current study follows the data from Livkiss *et al.*,  
408 which had the most similar setup focused on a wall cavity scenario (above the black line in Figure  
409 6). Conversely, although the data for the other two studies (Ingason and Karlsson, both focused  
410 on rack storage systems scenario, below the black line in Figure 6) follow a similar trend and the  
411 overall flame heights are significantly lower. Specifically, the current study shows approximately  
412 52% higher flames for the open base configuration compared to Karlsson *et al.*, 24.5% higher  
413 compared to Ingason, and 2.2% lower compared to Livkiss *et al.* A shorter flame length is  
414 expected for the studies by Ingason and Karlsson due to the presence of gaps in the vertical  
415 direction in the rack storage setups – as described in the Introduction section – which enabled the  
416 air entrainment in several heights of the gap. The results presented here therefore appear to  
417 confirm that flame heights are consistently different between rack storage system (with additional  
418 vertical cavities) and ordinary cavity flame heights where the two walls are parallel with no  
419 vertical cavities. The presence of these cavities in the vertical direction has a considerable effect  
420 and decreases the flame height significantly. This is expected to be because these cavities enhance  
421 the cold air entrainment, which cools down the plume, generating a decrease on the buoyancy and  
422 leading to a shorter plume. Additionally, the higher availability of air at lower heights due to the  
423 vertical cavities implies that the combustible gas can be mixed with the oxidiser at lower heights  
424 and is consumed at a lower height from the burner. Last, the geometry of the storage rack,  
425 especially the sharp edges might enhance flow shearing, promote the air-combustible mixing  
426 which allows the reaction to occur at lower heights, leading to shorter flames if compared to the  
427 parallel wall setup.

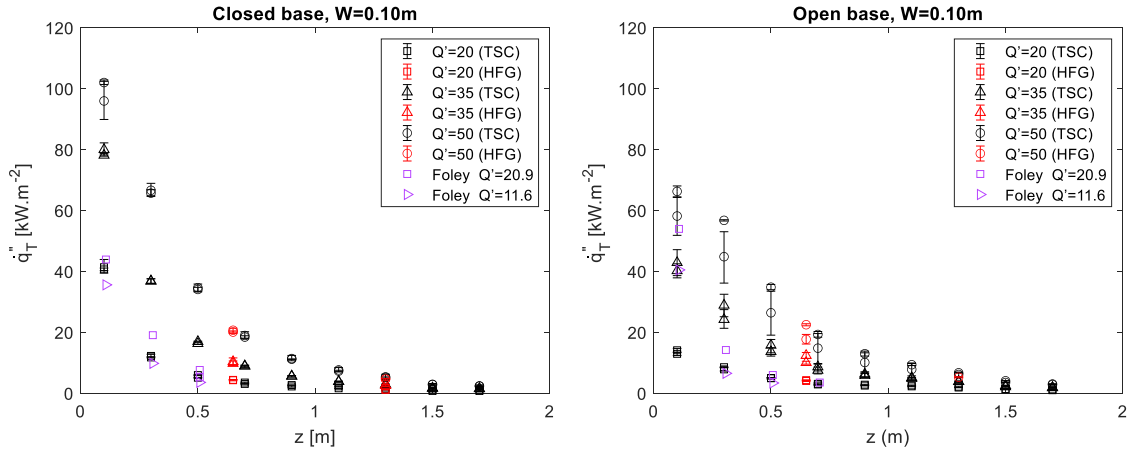
428 There are still some differences in the cavity flame heights presented here and those of Livkiss *et*  
429 *al.*, particularly at large cavity widths. This is likely because of the difference in the type of burner  
430 used in both studies. A flame produced by a sand burner (in this study) should be affected less by  
431 the momentum of the gases from the burner compared to a flame produced by a line burner (in  
432 Livkiss' study). Additionally, having the burner next to the wall would generate taller flames  
433 compared to the burner used in this study, which filled the whole cavity, due to the difference in  
434 the aspect ratio of the burners. Further discrepancies could be attributed to the variation within  
435 the techniques for the measurement of the flame, since Livkiss and the current study used different  
436 frequency for the flame height acquisition and different treatment for the effect of the view angle  
437 of the camera.

438 From all the above, it is clear that the flame height is highly sensitive to changes in the geometry  
439 of the system, the heat release rate of the fire source, and the restriction of the air supply. The  
440 results here suggest that parallel plates setups with no vertical gaps have consistently larger flame  
441 heights compared to storage rack systems. The exact impact of the vertical gaps has not been  
442 quantified here given that this is not particularly relevant for façade scenarios, since the inclusion  
443 of these additional gaps leads to the entrainment of cold air and to a lower heat exposure of the  
444 walls, thereby representing a less onerous scenario. Furthermore, air entrainment through vertical  
445 gaps appears to influence the system but has not been included in the current scope. To explore  
446 some of the differences in the results further, the total external heat flux helps describe some of  
447 the dynamics within the cavity.

## 448 **5. Total external heat flux**

449 The total external heat flux (refer to Eq. 11) as a function of height is presented in Figure 7. As  
450 with the flame heights, there are clear distinctions for different burner HRR, with higher HRR  
451 consistently giving higher total external heat flux across the height of the wall. The data from  
452 TSCs and heat flux gauges is plotted to enable a comparison of the results from the two  
453 instruments. It can be seen that the calculations for the centreline total heat flux follow a trend  
454 and that the values do not deviate significantly from the measurements obtained by the water-  
455 cooled heat flux gauges. This is because the thin skin calorimeters are calibrated using the heat

456 flux gauges and both the calculation and measurement are linked by using the correction factor  
 457 (C) in Eq. 16 . This allowed a comparison of these results with external heat flux measurements  
 458 performed by Foley & Drysdale [19]. A comparison of closed base (left) and open base (right)  
 459 also shows that a higher external heat flux is registered in the wall for the closed base scenario  
 460 compared to the open scenario condition, and the highest external heat flux is close to the burner  
 461 as would be expected.

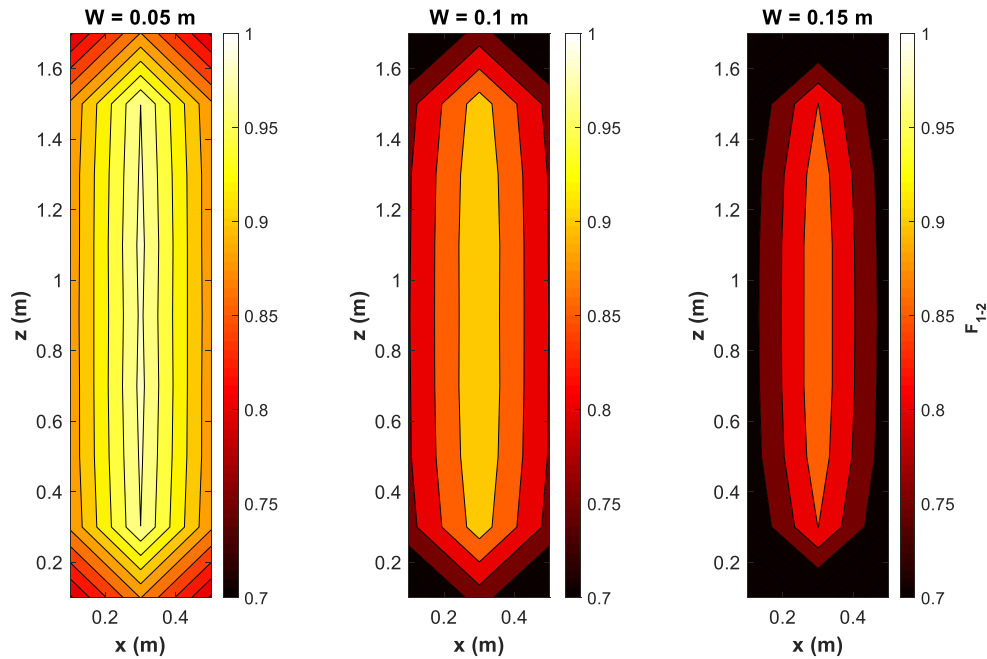


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463 **Figure 7.** Centreline total external heat flux measured by heat flux gauges(HFG) and calculated from TSCs. Error  
 464 bars correspond to the 20<sup>th</sup> and 80<sup>th</sup> percentile of the measurements for the steady state span.

465 The difference observed between the closed and open scenarios is because the flame fills the  
 466 whole cavity and impinges both walls in the case of the closed base. When the base is open, cool  
 467 air flows upwards from the base and comes between the flame and the walls, preventing direct  
 468 flame impingement and reducing the total external heat flux. Flame impingement increases the  
 469 temperature of both walls and generates a larger radiative exchange between the two surfaces.

470 The cavity width also had influence on the heat transferred to the walls of the cavity. The increase  
 471 of the cavity diminishes the view factor between the two surfaces. The effect on the view factor  
 472 was quantified using Eq. 6 and the spatial variation for the view factor along the plate for the three  
 473 different cavity widths was calculated and is presented in Figure 8. This change in the view factor  
 474 ultimately leads to an enhancement of the thermal exchange between the surfaces.



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**Figure 8.** Spatial variation of the view factor between the two parallel walls.

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It was determined that for the widest cavity, each wall is receiving only 80 % as much energy from the opposite wall as it did in the narrowest cavity. Also, a significant drop in the view factor is observed away from the centre points of the walls, due to the geometric configuration of the walls. This implies a lower radiative exchange between the two walls. It is possible to observe that although the flame heights are similar for both base configurations, the behaviour of the total external heat flux is different. A deeper study considering the influence of the air entrained on the heat flux distribution was conducted to explore the mechanisms underlying the discrepancies to explain this behaviour.

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### 5.1. Influence of air entrainment

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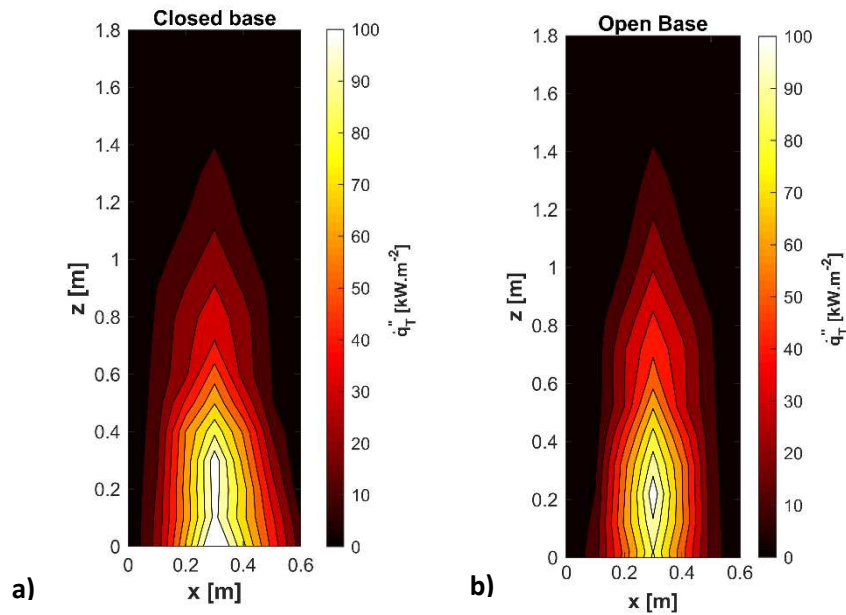
The TSCs located distant from the centreline of the wall (see Figure 5 (a)) were used to quantify the spatial variation of the total heat flux over the wall. A subset of the results is presented in

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Figure 9. It is evident that for both open and closed based configurations, the flame is symmetrically distributed and that the flame tip is present at the middle of the centreline, where the highest heat flux value is present. A decay of the heat flux towards the edges of the wall can be observed. This can be explained by the change in the view factor and by the air entrainment from the open sides of the setup. The view factor at the lateral edges is lower than at the centreline of the wall, which implies a lower radiative heat transfer. As for the air entrainment, this process could only happen laterally, via the cavity when the base was closed. The entrainment decreased as the cavity width was reduced. This causes the flame to be pushed towards the centreline of the walls ( $x=0.3$  m) by the incoming air. This effect would also happen in a similar but wider setup. Conversely, when the base is open, air is entrained both vertically from the bottom of the burner and horizontally through the cavity between the walls. Also, it was observed during the experiments that for the closed base, the flame behaved as a uniform sheet filling the entire cavity, whereas for the open scenario, the flame impinged the walls for a shorter distance. A more detailed understanding of the heat transfer behaviour was sought through the study of the heat transfer mechanisms to determine the influence of the different factors on the external heat flux and propose adequate design measures that diminish the level of exposure from the dominant heat transfer component.

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**Figure 9.** Heat flux spatial distribution a) HRR = 35 kW.m<sup>-1</sup>, W = 0.05 m, closed base b) HRR = 35 kW.m<sup>-1</sup>, W = 0.05 m, open base

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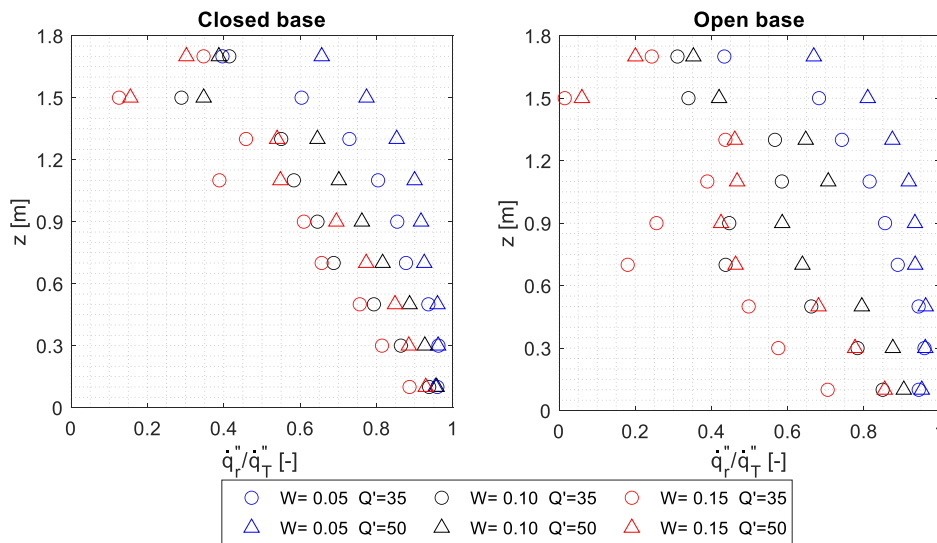
### 5.2. Dominant heat transfer mechanism

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The magnitude of the radiative component and its dependence on the cavity width and height from the fire source was characterised as the ratio between the radiative heat transfer and the total heat transfer. This ratio is presented for different configurations in Figure 10.

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**Figure 10.** Contribution of the radiative heat flux component to the total external heat flux vs setup height

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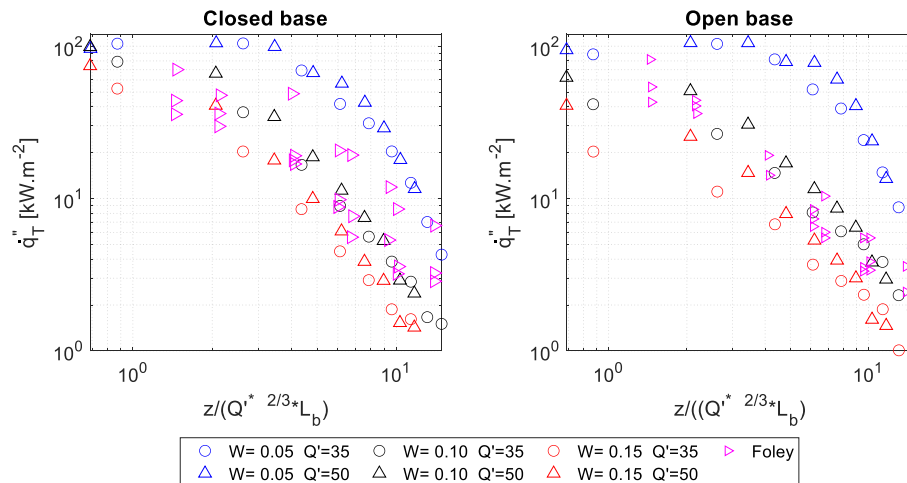
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A decrease of the contribution of the radiative component can be observed as the cavity width increases, due to the reduction of the view factor and hence the radiative exchange between the two surfaces. Additionally, a decrease in the significance of the radiative component can be observed for all the configurations as the height above the burner increases. This could be explained by the absence of the flame at the upper region of the walls. The significance of the radiative component follows a similar trend for both the open and closed configurations. However, it can be noticed that this ratio decreases more significantly in the open base configuration. The increase in radiation in the closed base scenario arose from restricting air

522 access for combustion, causing a thickening of the flames. This phenomenon can also explain the  
 523 increase in the radiative component when the cavity width is reduced. It was also expected for  
 524 radiative exchange between the walls to decrease with increased separation. The generalisation  
 525 of this behaviour would require experiments in a wider range of conditions. However, the results  
 526 obtained highlight the importance of radiative heat transfer between surfaces that restrict a fire.  
 527 A system where heat transfer is dominated by radiative heat transfer is more easily scalable, since  
 528 the effect of the heat losses by convection can be neglected.

529 *5.3. Dimensionless correlations for the external heat flux as a function of the system variables*  
 530 Correlations were sought for the calculated total external heat fluxes and their dependence on the  
 531 system variables (i.e. cavity width, heat release rate per unit length of burner, aspect ratio of the  
 532 burner, height from the base of the burner). First, the dependence on the spatial variation and the  
 533 dimensionless height,  $z/(Q^{*2/3} \cdot L_b)$ , based on previous work by Hasemi [26] and Thomas *et al.*  
 534 [45], was studied. A subset of the data consisting of the cases with the same heat release rates per  
 535 unit length burner and data from Foley and Drysdale [19] are presented in Figure 11.



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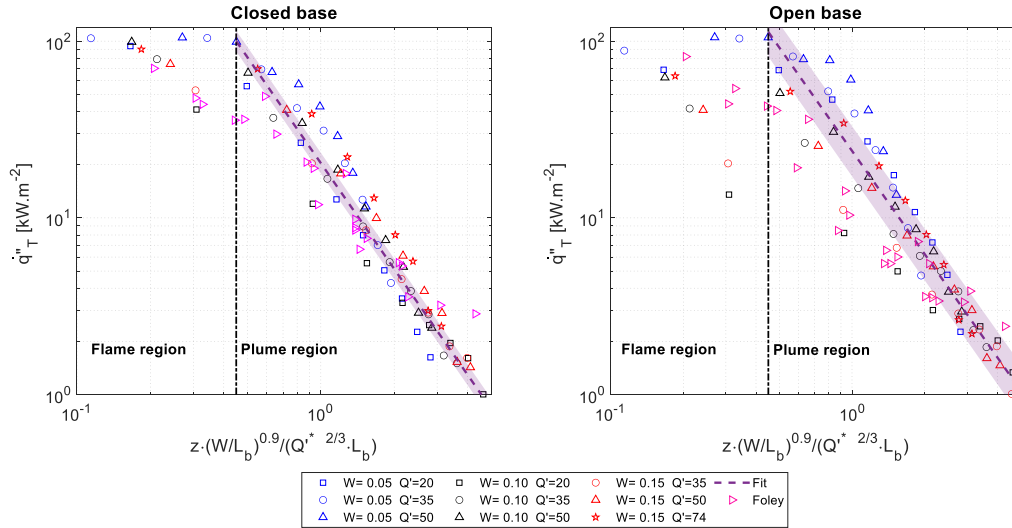
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**Figure 11.** Heat flux as a function of the dimensionless height

538 Figure 11 shows the influence of the cavity width and the heat release rate on the total external  
 539 heat flux. For both open and closed base configurations, the cavity walls are exposed to a greater  
 540 external heat flux as the cavity width is decreased. Also, it is possible to observe that a higher  
 541 heat release rate of the burner led to higher heat fluxes, as expected. A considerable dispersion  
 542 between the data series corresponding to different cavity widths was observed, which indicated  
 543 the inclusion of an extra parameter accounting for the geometry of the burner was necessary to  
 544 model the behaviour of the total external heat flux.

545

546 A term considering the aspect ratio between the cavity width and the burner length was then  
 547 included to explore the effect of changing this variable. The exponent for the aspect ratio was set  
 548 to 0.9 after an optimization process that aimed to minimise the dispersion of the data in the plume  
 549 region. Data obtained by Foley [18] for the same dimensional groups were adjusted to the format  
 550 of the obtained correlations (Eq. 20) and are presented along the experimental data for this study  
 551 in Figure 12. The delineation between the flame and the plume that is shown in Figure 12 is based  
 552 on the variation of the incident heat flux on the wall, which is minimal in the flaming region as  
 553 opposed to in the plume region where a sharp decay of the incident heat flux to the wall can be  
 554 observed. This approach is consistent with the work by Back [16].



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**Figure 12.** Heat flux as a function of the dimensionless height. The shaded areas represent the 95% confidence bounds

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The correlations were limited for the region of the abscissa, where a sudden decrease of the value for the external heat flux was noticed. The correlations obtained are more conservative than the ones obtained by Foley since, in both open and closed scenarios, the value for the prediction of the model is higher than the data obtained in that study. This can be in part attributed to the use of a lower heat release rate for one of the configurations used by Foley. A greater scatter can be observed for the open base configuration. This could imply that the variables used for the correlation might not be enough to predict the behaviour of the total external heat flux, and further studies with different air entrainment conditions should be conducted to obtain a more accurate model.

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A set of correlations following the structure presented in Eq. 20 was obtained for the ranges corresponding to the plume region.

$$\dot{q}''_T = C_1 \cdot \left( z \cdot \left( \frac{W}{L_b} \right)^{0.9} / (Q'^{* \frac{2}{3}} \cdot L_b) \right)^{C_2}, \quad \left[ z \cdot \left( \frac{W}{L_b} \right)^{0.9} / (Q'^{* \frac{2}{3}} \cdot L_b) \right] > 0.45 \quad (20)$$

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where  $C_1$  is the model preexponential factor and  $C_2$  is the dimensionless heat flux decay exponent.

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A single expression containing the data for both base configurations could be proposed, however the correlation with the combined experimental data is significantly weaker than when the two configurations were treated separately. The coefficients for the two cases contained in Eq. 20, their error margins for the 95% confidence bounds and their respective correlation coefficient are presented in table 4.

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**Table 4.** Error margins corresponding to 95% confidence bounds for the coefficients used in the external heat flux correlations and correlation coefficients.

	Preexponential factor ( $C_1$ )	Exponent ( $C_2$ )	Correlation coefficient ( $R^2$ )
<b>Closed</b>	$20.42 \pm 0.96$	$-1.99 \pm 0.11$	0.95
<b>Open</b>	$23.76 \pm 2.4$	$-1.94 \pm 0.37$	0.87

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It is possible to tell although the correlations present the same structure, the coefficients are different for both open and closed base configurations. A greater scatter is observed for the open base scenario which can be observed in larger error margins for the coefficients. The total external

581 heat flux depends on  $\left(\frac{W}{L_b}\right)^{-1.76}$  for the open base scenario, whereas it depends on  $\left(\frac{W}{L_b}\right)^{-1.79}$  for  
582 the closed base scenario. This means a greater influence of the cavity width for the later  
583 configuration. This could be attributed to the additional availability of air at the bottom of the  
584 flame and its subsequent effect on the cooling of the walls and the prevention of flame  
585 impingement, both leading to a sharper decay on the external heat flux measured at the surface.

586 It was not possible to determine the underlying causes for the degree of variability of the  
587 coefficients for the open and closed base scenarios. This is because in this study only two levels  
588 of air entrainment were included and this variable was not quantitatively characterised. The use  
589 of CFD tools as well as a detailed quantification of the magnitude of the air entrained in all the  
590 directions of the plume might be beneficial to explore the underlying physics in more detail.

## 591 **6. Conclusions**

592 This paper has investigated the differences in flame heights and heat flux for various cavity fire  
593 setups by different authors, and has generated new data for further comparison with these existing  
594 datasets. The differences in the behaviour of the flame between the different configurations has  
595 been explained by an analysis of the underlying mechanisms that govern the fire and its interaction  
596 with the surroundings.

597 The obtained data for the normalised flame heights follows a similar trend to previous studies,  
598 which point to similar governing mechanisms for the fire dynamics in cavities. The differences  
599 among setups can be mainly attributed to the presence of additional vertical gaps in rack storage  
600 systems that cause a decrease in the elongation of the flame. Other sources of variability can be  
601 the position of the flame in the cavity and the type of burner as well as the use of different methods  
602 to record the flame height.

603 An increase of the total external heat flux on the walls was observed when the separation between  
604 the walls was decreased because of different reasons. First, the heat exchanged among the surface  
605 comprising the cavity increases because a relative increase of the view factor. Besides the cavity  
606 size, air entrainment at the bottom of the walls was shown to have an influence on the external  
607 heat flux. The flow of air from the base reduces the flame impingement on the walls reducing the  
608 energy transferred from the flame to the wall. This phenomenon is more significant in the  
609 configurations with the 0.10 m and 0.15 m separation. The findings of this research in regards to  
610 the influence of cavity width and air entrainment on the external flux received by the walls of a  
611 cavity have direct relevance to problems of ignition and upward flame spread in confined spaces  
612 which have inward-facing combustible surfaces, such as ventilated facades.

613 It was found that radiation generally dominates the heat transfer, especially near the bottom of the  
614 system. A decrease in both convective and radiative heat transfer is noticed when the cavity width  
615 is increased. For heights further from the fire source, the heat flux trends indicate that the air  
616 entrainment from the sides might be more relevant than the entrainment from the base. However,  
617 further studies or measurements on air entrainment would be required to confirm this and obtain  
618 more accurate models for both scenarios.

619 The results of this work are only applicable to the ranges studied, and no consideration has been  
620 given to extensive scaling or extrapolation. The conclusions for the “open base” configuration  
621 apply to an 80 mm gap at the bottom and the influence of the size of this opening could be the  
622 subject of further study. Furthermore, the basic setup is intended to explore the relevant  
623 phenomena and their influence on fundamental fire dynamics. The correlations and data obtained  
624 for the external heat flux as a function of the cavity width, the heat release rate of the fire and the  
625 height of the walls can be used as a design tool if a maximum allowable heat transfer exposure to  
626 the combustible linings is set as a test input. Understanding the behaviour in complex façade

627 systems for real buildings requires additional work and careful consideration, since the research  
628 presented in this article was executed in a simplified set up with reduced dimensions to focus on  
629 the characterisation of the fundamental principles governing the system. Future work could also  
630 include further numerical work to verify the applicability of the proposed scaling laws, studying  
631 the impact of the addition of a combustible lining in the cavity, varying the aspect ratio between  
632 the burner and the cavity, as well as the position of the burner to better understand the implications  
633 of these variables in the fire hazard of the system.

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640

641 **Nomenclature**

$C$	Thin skin calorimeter correction factor (—)
$C_1$	model preexponential factor( $\text{kW} \cdot \text{m}^{-2}$ )
$C_2$	dimensionless heat flux decay exponent(—)
$C_p$	specific heat capacity ( $\text{kJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$ )
$Fr$	Froude number (—)
$g$	gravitational constant ( $\text{m} \cdot \text{s}^{-2}$ )
$h_c$	convective heat transfer coefficient ( $\text{kJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$ )
$H$	vertical cavity size (m)
$k_g$	gas conductivity ( $\text{kW} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$ )
$K_1$	model constant ( $\text{kW} \cdot \text{m}^{-2}$ )
$K_2$	dimensionless burner aspect ratio constant (—)
$K_3$	dimensionless heat flux decay constant (—)
$L_b$	burner length (m)
$L_c$	characteristic length (m)
$L_f$	flame height (m)
$Nu$	Nusselt number (—)
$Pr$	Prandtl number (—)
$Q$	heat release rate (kW)
$Q'$	heat release rate per unit length of burner( $\text{kW} \cdot \text{m}^{-1}$ )
$Q'^*$	dimensionless heat release rate (—)
$\dot{q}''_c$	convective heat flux ( $\text{kW} \cdot \text{m}^{-2}$ )
$\dot{q}''_r$	incident radiative heat flux ( $\text{kW} \cdot \text{m}^{-2}$ )
$\dot{q}''_T$	total external heat flux ( $\text{kW} \cdot \text{m}^{-2}$ )
$R$	shortest distance between two surfaces (m)
$Ra$	Rayleigh number (—)
$Re$	Reynolds number (—)
$T_{gas}$	gas phase temperature (K)
$T_s$	solid phase temperature (K)
$W$	cavity width (m)
$W_b$	burner width (m)
$z$	height (m)

$\alpha_{TSC}$	absorptivity (—)
$\delta$	thickness (m)
$\varepsilon_{TSC}$	emissivity (—)
$\rho$	density ( $\text{kg. m}^{-3}$ )
$\rho_{\infty}$	ambient density ( $\text{kg. m}^{-3}$ )
$\sigma$	Stefan-Boltzmann constant ( $\text{kW. m}^{-2}. \text{K}^{-4}$ )
$\theta$	surface angle (rad)

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