CFD study of fire-induced pressure variation in a mechanically-ventilated air-tight compartment

Junyi Li^{a,*}, Tarek Beji^a, Sylvain Brohez^b, Bart Merci^a

^a Ghent University, Department of Structural Engineering and Building Materials, Ghent, Belgium

^b University of Mons, Département de Génie des Procédés Chimiques, Mons, Belgium

ARTICLE INFO

Keywords

CFD modeling
Fire-induced pressure
Duct reverse flow
Passive house fire

ABSTRACT

Computational Fluid Dynamics (CFD) simulations of fire-induced pressure and ventilation flow in a mechanically-ventilated air-tight compartment, representative of a passive house, are presented. Experiments conducted by University of Mons are used to assess the results. The simulation heat release rate is prescribed based on experimental measurements. The experimental duct flow rate is about 80 m³/h. Two methods are used to meet the aforementioned value in simulations: 1) by modeling the actual fan curve and dampers, and 2) by modifying the fan curve to avoid simulations of dampers. The combination of real fan curve and dampers provides better results. The simulations reproduce well the pressure profile (maximum 433 Pa) and the duct flows. The simulated reverse inlet flow and increased outlet flow rates reach values of 157 m³/h and 165 m³/h, respectively. Besides, the fire-induced pressure is high enough to hinder evacuation and fire rescue operations due to the impossibility of opening inward-open doors. Moreover, the adjacent room pressure also reaches a high level. Reducing the gap area between rooms significantly reduces the adjacent room pressure, but leads to an increase of the fire room pressure. Temperature deviations are observed and are improved by modeling a more realistic fire source.

Nomenclature

A	area (m ²)
c_p	specific heat of fluid (kJ/(kg·K))
D^*	characteristic length scale (m)
g	gravitational acceleration (m/s ²)
K	loss coefficient (–)
n	leakage pressure exponent (–)
Δp	pressure difference (Pa)
$\dot{Q}$	heat release rate (kW)
T_∞	ambient air temperature (K)
$\dot{V}$	volume flow rate (m ³ /s)

Greek

ρ_∞	ambient air density (kg/m ³)

Subscripts

d	diaphragm
eff	effective
L	leakage
$minor$	minor losses
ref	reference

1. Introduction

In order to achieve the goal of low energy consumption in buildings, air-tight constructions are widely used nowadays. The present paper focuses on a configuration, representative of a modern type of low-energy building, called ‘passive house’, which can significantly reduce the total energy demand over the life cycle of the buildings [1]. Many strategies are applied in passive houses to minimize the consumption of energy, such as boundary insulation, heat recovery, passive house windows, airtightness and thermal bridge reduction. The influence of the building airtightness on the pressure build-up in the course of fires is the focus of the present study.

While bringing benefits of energy savings, the airtightness poses additional fire risks to such buildings. Due to the small leakage area, the hot combustion products cannot easily escape from the enclosure, resulting in a high pressure during fires. It has been reported that a fire in a passive house has led to a pressure that is sufficiently high to prevent the occupants from opening the door and evacuating [2]. A fire-induced over-pressure of 100 Pa is mentioned in the literature [3,4] as a criterion to assess whether it is impossible for people to open the inward opening doors. Hence, the evacuation of occupants and rescue operations of firefighters could be hindered. In addition to this, if the pressure inside a passive house reaches up to 1000 Pa, the structural in-

* Corresponding author. UGent Campus UFO, Technicum Blok 4, Sint-Pietersnieuwstraat 41, B-9000, Ghent, Belgium.

E-mail address: junyi.li@ugent.be (J. Li)

tegrity could be compromised. For example, in Ref. [5], a window was shattered due to the high internal pressure.

Experimental studies demonstrated that the pressure variation in an air-tight compartment fire can result in a dangerous situation. In a series of apartment fire tests carried out by Aalto University, the recorded over-pressures were between 100 Pa and 1650 Pa within 50 s from ignition [5]. They also illustrated that inward-opening doors could not be opened from inside the fire room during the over-pressure period [5]. Besides, damage in the structure caused by high pressure was observed in one of their fire tests. The issue of pressure build-up in air-tight compartments has also been examined in the context of nuclear power plant safety. Within the PRISME (French acronym for "Fire Propagation in Elementary Multi-Room Scenarios") fire research program, several experimental campaigns have been carried out in confined compartments, and an over-pressure up to almost 3000 Pa inside the fire room was observed [6]. Prtre et al. [7] pointed out that the pressure peaks are an important issue for fire safety in confined and forced-ventilated enclosures. They also indicated that the pressure variation is mainly influenced by three aspects: the fire size, the ventilation conditions and the enclosure properties (thermal boundary conditions, geometry, etc.). Utiskul et al. [8] studied fire behaviour under limited ventilation in a small scale compartment and observed an oscillation phenomenon of pressure inside the compartment. Several experimental tests were also carried out by University of Mons on an experimental setup constructed by the Rgie Provinciale Autonome du Hainaut on their fire training center to illustrate the fire-induced pressure build-up in an air-tight compartment representing the passive house, producing the experimental data used in the present paper [9–12]. The highest recorded pressure peak value in one of the experiments reached 2000 Pa.

Numerical studies have also been conducted to assess the capability of CFD simulations in predicting pressure variations. Chow et al. [13] used the Fire Dynamics Simulator (FDS), version 3.01, to investigate the fire-induced pressure variation in a confined chamber and pointed out that simulation results agreed reasonably well with experimental results. Numerical results reported by Brohez et al. [9–11] also validated the using of CFAST and FDS in modeling fire-induced pressure. Beji et al. [14] evaluated the capability of FDS, version 5.5.3, in the simulation of a large scale, well-confined and mechanically ventilated multi-room fire scenario. Their results showed that FDS can give a good first basis for a fire hazard analysis in forced-ventilated enclosure fires, provided that the HRR (heat release rate) is known from experiments or design calculation requirements. Wahlqvist et al. [15] simulated compartment fires with different leakage classes using FDS 6.1.1. It was shown that modeling leakage correctly is important to reproduce the pressure variation as well as ventilation system behavior. The obtained high fire-induced over-pressure of the tightest leakage class defined in Ref. [16] is about 25 times that of the loosest class.

Existing research has mainly focused on the fire dynamics inside well-confined and mechanically ventilated compartments, but the pressure build-up in passive houses and the influence on the pressure pro-

file of numerical settings, such as the leakage area and ventilation system, have received relatively less attention. This motivates the present research on fire-induced pressure variation in compartments with a high level of air-tightness. The present work constitutes a numerical study by means of CFD, more specifically using FDS version 6.5.2 [17], to investigate the influence of leakage area and ventilation settings on pressure build-up inside an air-tight compartment. This has been reported in Ref. [18], but the present paper extends to more configurations, including cases with reference leakage area of 0.0019 m², cases with different combinations of fan curve and damper, and cases with different settings on the designation of leakage area, i.e., 1) assigning the leakage area evenly to the two rooms, and 2) using a uniform distribution over the walls and ceiling surface area. The temperature profiles inside the compartment are also discussed in the present paper.

2. Experimental configuration

The experimental data, used in the present study, have been collected in a measurement campaign, performed by Brohez et al. at UMons (Belgium) [9–12]. Details can be found in Refs. [9–12], but the main features are repeated here for the sake of ease for the reader. The compartment consists of two rooms and has the same inner dimensions as a 40-foot shipping container [19]. The floor plan of the experimental facility is shown in Fig. 1.

A partition wall and a closed partition door separate the compartment into two rooms. A small gap of approximately 0.009 m² (0.9 m × 0.01 m) is located at the bottom of the partition door. The larger room was used as the fire room, where the fire source was positioned as shown in Fig. 1. The dimensions of the fire room and the adjacent room are 8 m × 2.4 m × 2.4 m and 4 m × 2.4 m × 2.4 m, respectively. The dimensions mentioned in Ref. [18] turned out to be not entirely correct and have been modified here. These slight modifications do not have a significant influence on the outcomes (not shown). The outer door, linked to the outside in the adjacent room, was closed and sealed during the experiments. Thermocouple trees and pressure measurement instruments were installed in the facility (see Fig. 1). There are 11 thermocouples in each thermocouple tree. The thermocouples are positioned with a spacing of 0.2 m and the first thermocouple is about 0.1 m from the floor. A pressure probe is positioned on the wall near the thermocouple tree, at 30 cm height from the floor.

Fifteen layers of overlapping wooden slats were used as a fire source in the experiments performed by UMons [9–11], as sketched in Fig. 2. The slats are 38 cm long and the section of a slat is 27 mm × 18 mm. A stainless-steel cup with a diameter of 9.5 cm containing 100 ml of heptane (used for ignition) was centered below the slats [9]. The slats and cup were put on a scale to measure the mass evolution. The heat release rate was calculated from the mass loss rate measured during the fire test, assuming an effective heat of combustion of 12 MJ/kg [9,12]. The result is presented in Fig. 3.

Fig. 1. Floor plan of the experimental facility.

Fig. 2. Sketch (unit: mm) of the fire source in the experiments of [9] (overlapping wooden slats). Left: top view; right: side view.

Fig. 3. Temporal evolution of heat release rate in the experiments (performed by UMons [9–12]) and as imposed in the numerical simulations.

The outer walls consist of three layers (from outside to inside): 0.2 m concrete brick, 0.05 m insulation layer and 0.026 m plasterboard. The ceiling is made of 0.15 m concrete slab and insulated by 0.013 m plaster boards and 0.05 m mineral wool. The partition wall is 0.12 m thick and consists of two layers of plaster boards with mineral wool in between. The partition door is 0.04 m thick and is composed a stratum of foam covered by two layers of steel plates. The thermal properties of the materials of the layers are taken from Ref. [4] and shown in Table 1.

The leakage of the compartment was estimated by UMons [9–12], using a fan pressurization test with a differential pressure of 58 Pa. During the leakage test, the partition door was open and the entire fa-

cility was treated as one pressure zone according to the ASTM standard [20]. It is noted that the partition door was closed during the fire test. Results shown in Refs. [9–12] illustrate that the leakage area from the blower door test at 58 Pa is 0.0019 m², considering a discharge coefficient equal to $C_d = 1$. When the discharge coefficient is taken to be $C_d = 0.7$, the leakage area is about 0.0026 m².

A mechanical ventilation system was added to the compartment to study its influence on the fire behavior. In the scenario analyzed in this paper, fresh air flows into the compartment through ducts connected to the fire room, while the gas was extracted from the ducts connected to the adjacent room. The lengths of the inlet duct and the outlet duct are 2.5 m. The diameter and roughness of the ducts are 0.1 m and 0.0015 m [19], respectively. Each of the inlet and the outlet ducts is equipped with a ventilation fan and a diaphragm (damper), as shown in the sketch in Fig. 1. The ventilation fans have a stalling pressure of 145 Pa. The diaphragm was used in the experiments by UMons to control the maximum duct volume flow rate at about 80 m³/h, in line with Belgian requirements for residential ventilation.

3. Numerical setup

The Fire Dynamics Simulator (FDS) was employed in the numerical study. The default settings of FDS, version 6.5.2, are used unless specified otherwise. They can be found in the user's guide [17].

3.1. Fire source

In most of the simulations, the fire source was represented by a 0.4 m × 0.4 m × 0.4 m cube with prescribed HRR based on the experimental data (Fig. 3). During the first 60 s, there is no fire, so that the flow in the ducts can stabilize. The second peak in the HRR profile (at 570 s) was reported in Ref. [9] to be due to the fresh air flowing into the compartment again through the inlet ducts (see Fig. 8), enhancing the combustion to some extent [9].

In order to evaluate the influence of the fire source modelling, a spreading fire simulation has been performed in which the fire source is not a cube, but a 'crib' with a localized ignition point at the center of the bottom surface (which is more representative of reality), as shown in Fig. 4. A spread rate of 0.05 m/s has been selected after performing some trials to get the desired time history of the HRR (with very small deviations from the experimental profile) as suggested in the FDS user's guide [17]. It is important to note that when a constant spread rate is prescribed (as in the 'crib' simulation), neither heat-up nor pyrolysis

Table 1
Thermal properties of the materials of the layers.

Material	Density (kg/m ³)	Conductivity (W/(m·K))	Specific heat (kJ/(kg·K))
Concrete	2200	1.100	0.88
Brick	1600	0.690	0.84
Mineral wool	100	0.041	0.80
Gypsum	1440	0.480	0.84
Foam	800	0.028	1.45
Steel	7850	45.800	0.46

Fig. 4. Fire source in the FDS simulations. (a) Cubic; (b) Crib.

are modeled; the gas phase is fully decoupled from the solid phase (which is also the case for the cubic fire source simulations). Therefore, it is not necessary to specify the thermo-physical properties of the crib, e.g., thermal conductivity of wood.

3.2. Leakage settings

For passive houses, leakage may occur at gaps of doors, windows or wall joints. There are two methods to simulate the leakage in FDS: ‘Pressure Zone Leakage’ (Bulk Leakage) and ‘Localized Leakage’ [17]. The pressure zone leakage approach is intended to capture the bulk leakage through walls. The leakage flow is uniformly imposed over all surfaces that are designated as leak path. The localized leakage approach is suitable for simulating small cracks with a known location. For the pressure zone leakage approach, the volume flow rate through leakages is given by:

$$\dot{V}_L = C_d A_L \text{sign}(\Delta p) \sqrt{\frac{2|\Delta p|}{\rho_\infty}}, \quad (1)$$

where $\dot{V}_L$ is the volume flow rate (m^3/s), A_L is the leakage area (m^2), Δp is the pressure difference between the adjacent zones (Pa) and ρ_∞ is the ambient air density (kg/m^3). In FDS, the default choice for the discharge coefficient in Eq. (1) is $C_d = 1$ and the leakage area is assumed to be constant.

However, there is an option to deviate from Eq. (1). In Ref. [17], it is stated that the leakage area may grow when the room pressure increases as small gaps, cracks and other leakage paths open up. In order to simulate such a non-constant leakage area, FDS provides a ‘LEAK-PRESSURE_EXPONENT’ parameter, n , which can be used to reproduce the variation of the leakage area:

$$A_L = A_{L,\text{ref}} \left(\frac{\Delta p}{\Delta p_{\text{ref}}} \right)^{n-\frac{1}{2}}, \quad (2)$$

where $A_{L,\text{ref}}$ is the reference leakage area (m^2) at the reference pressure difference Δp_{ref} between the adjacent zones. By default, $n = 0.5$, which means that the leakage area is independent of the pressure rise, and $\Delta p_{\text{ref}} = 4$ Pa. However, as the reference leakage area has been determined at a pressure difference of 58 Pa here, Δp_{ref} has been set to 58 Pa. Different values of the exponent n (0.5, 0.6 and 0.7) are used below to study the influence of the exponent on the pressure variation which is discussed later in the present paper.

It is worth noting that another interpretation can be given, combining equations (1) and (2), namely that the leakage area remains constant, but the exponent for the pressure difference is modified from $n = 0.5$ (which is the value from the classical Bernoulli equation) to a higher value (e.g., 0.6 or 0.7), which is more representative for the flow through cracks, i.e., laminar flow ($n = 1$) or turbulent flow ($n = 0.5$) [21].

In the present work, the air leakage from the compartment to the ambient environment is simulated using the pressure zone leakage ap-

proach because the exact locations of the leakages are unknown. The walls and ceiling are set as leak path. Because the partition door was closed during the experiment, two pressure zones (one for each room) are prescribed in FDS, in addition to the ambient pressure zone. The leakage area is uniformly distributed over the walls and ceiling surface area, so the leakage area of the fire room is larger than that of the adjacent room. The case of evenly assigning the leakage area over the two rooms was also tested, and the results are not very different (see below, section 4.3). Furthermore, the leakage between the fire room and the adjacent room under the partition door has been prescribed using the localized leakage approach.

3.3. Ventilation system

The ventilation system is reproduced according to the experimental setup of UMons as described in Refs. [9,19]. Two methods were used to set the volume flow rate in the simulations (see Fig. 5): 1) by modeling the actual fan curve and dampers, i.e., fan curve 1 with damper; and 2) by modifying the fan curve in an attempt to avoid the inclusion of dampers in the simulations, i.e., fan curve 2 without damper. Fan curve 1 is the prescribed fan curve in FDS based on the actual fan curve as reported by UMons [9–11], instead of the commonly used quadratic fan curve. It interacts with dampers to meet the volume flow rate requirements. Fan curve 2 (see Fig. 5) is a simplified curve, with the maximum flow rate set to $80 \text{ m}^3/\text{h}$. The method of the simplified fan curve was used in Ref. [19] to meet the maximum duct flow rate without damper modelling in the simulations. The modified fan curve used in Refs. [18,19] was thus a simplification of the actual low-speed mode curve of the fan. Actually, in the UMons experiments, the fan was running in the high-speed mode (i.e., fan curve 1 in Fig. 5) [9], and thus a more realistic simplified curve (i.e., fan curve 2 in Fig. 5) is used in the present paper. A similar simplified fan curve was also used in Refs. [9,12] to simulate the compartment fire using CFAST, but the curve is not completely identical with the curve here.

During the simulations, when reaching a point where the flow in the duct is at the maximum flow rate, the fan will not add any additional pressure and the flow rate solely depends on the node pressures of the HVAC ductwork. If, on the other hand, the flow rate reaches zero, the fan adds the maximum fan pressure to the node pressure difference in the HVAC ductwork. If the pressure keeps increasing, there will be a reverse flow and the pressure rise by the fan is kept constant. It is noted that in the experiments there was a diaphragm in the ducts and the opening percentage was about 45% [9]. In FDS, for a partially open diaphragm, the loss of ducts can be specified to simulate dampers with variable position. The effective loss of the duct, K_{eff} , can be calculated as [9,17]:

Fig. 5. Fan curves as implemented in the simulations. Fan curve 1 has been reported by UMons [9–12].

$$K_{eff} = \left(\frac{A_{eff}}{A_d} \right) K_{minor}, \quad (3)$$

where A_{eff} is the effective area of the duct (m^2), A_d is the opening area of the diaphragm (m^2) and K_{minor} is the dimensionless ‘minor losses’ coefficient, which represents losses through components (the diaphragm here). Here, $A_{eff} = 78.54 \text{ cm}^2$, $A_d = 30.94 \text{ cm}^2$ and $K_{minor} = 7$, so the effective loss of the duct is 17.77 [9].

4. Results and discussion

4.1. Mesh sensitivity analysis

A mesh sensitivity analysis is performed. The required mesh cell size was first estimated using the D^* criterion [17]. The recommended ratio of $D^*/\delta x$ is between 5 and 10 [22]. In the present paper, the (cubic) mesh cell sizes, equal to 0.2 m, 0.1 m and 0.05 m, correspond to ratios calculated based on the peak HRR equal to 2.75, 5.5 and 11, respectively. The simulation results of pressure variation and duct flow rates are shown in Fig. 6. Only the results between 150 s and 750 s are shown in Fig. 6 for the sake of clarity. While the D^* criterion has been developed in the context of smoke plumes, Fig. 5 illustrates that application of this criterion also leads to meshes yielding grid insensitive results for global quantities like pressure and volume flow rates. Although there are some discrepancies between 220 s and 290 s and

for the peak pressure, the overall simulation results with mesh cell sizes of 0.1 m and 0.05 m are similar, while a clear difference is noted from the result with mesh cell size of 0.2 m. The HVAC system is linked to the computational domain through a vent, spanning over several mesh cells. On a finer mesh, this span is thus over more mesh cells than on a coarser mesh. This could result in a more accurate calculation of the flow properties at the junction between the computational domain and the HVAC system, which can explain the aforementioned discrepancies. The relative differences in maximum pressure, inlet duct volume flow rate and outlet duct volume flow rate between the 0.1 m mesh and the 0.05 m mesh are 5.4%, 3.6% and 2.1%, respectively. The aforementioned values are slightly different from the ones in Ref. [18], because the simulations were re-run, using the modified dimensions as mentioned in Section 2. The difference between cases with different cell sizes is small, but the computational time for the 0.05 m grid is about 16 times that of the 0.1 m grid. Therefore, the mesh cell size of 0.1 m is considered fine enough for this study.

4.2. Influence of the mechanical ventilation settings

Simulation results of the pressure variation in the fire room are shown in Fig. 7 for different combinations of fan curves and dampers. The results shown in this section are based on the reference leakage area of 0.0019 m^2 and the leakage pressure exponent of 0.7, which will be discussed in detail in the next section. When there is no damper, the resistance in the duct is much lower, resulting in more gas flowing

Fig. 6. Simulation results with different mesh cell sizes. Temporal evolution of: (a) Pressure variation in the fire room; (b) Duct flow rate.

Fig. 7. Pressure variation as function of time in fire room for different combinations of fan curves and dampers. (a) Fan curve 1; (b) Fan curve 2. The dashed red lines correspond to the critical value of pressure which could hinder evacuation [3,4] and the duration of the hindrance. Experimental data reported by UMons [9–12]. (For interpretation of the references to colour in this figure legend, the reader is referred to the Web version of this article.)

out (see Fig. 8 b and Fig. 9 b). Consequently, the pressure rise in the compartment is underestimated. The experimental peak value of the fire-induced over-pressure in the fire room is about 421 Pa [9]. Modelling the damper (and more specifically, the effect of the damper on the resistance) in the simulations is essential to reach the maximum pressure level as measured. On the other hand, the type of fan curve has very little impact on the pressure profile when using the same damper settings. During the decay period, an under-pressure of about -140 Pa is observed in the experiments due to the cooling down of gases in the compartment [9]. The reduction in pressure seems exaggerated in the simulations during the first part of the decay period. As a consequence, the experimentally observed second pressure peak, due to the second peak in HRR (see Fig. 3), is under-predicted. Nevertheless, the pressure difference between the first minimum (at $t = 549$ s) and the second maximum (at $t = 578$ s), is well captured (230.9 Pa in the experiment [9], vs. 233.1 Pa in the simulations). The decay in pressure during the second part of the decay period in HRR is less pronounced in the simulations. When considering the critical pressure mentioned in the literature [3,4], i.e., 100 Pa, the fire-induced over-pressure in the present case is much higher than this value during a period of about 200s, as shown by the dashed lines in Fig. 7. This means that the opening of inward-open doors could be impossible at the early stage of the fire. It can be seen that the pressure peaks are different with different ventilation settings, but the period during which the over-pressure is larger than 100 Pa is more or less the same.

The duct flow rates are shown in Figs. 8 and 9 for different combinations of fan curves and dampers. During the first 60 s, the flow in the duct reaches a steady state. The volume flow rates generated by

fan curve 1 with damper, fan curve 2 with damper and fan curve 2 without damper, all meet the requirement of $80 \text{ m}^3/\text{h}$. As such, the three mentioned combinations are a good representation of reality prior to the fire. However, the combination of fan curve 1 without damper generates higher flow rate in the ducts due to the lower resistance in the ducts without dampers. When the fire occurs, the inclusion of the damper again leads to better agreement with the experimental results. When the room pressure exceeds the stalling pressure of the ventilation fan (see Fig. 5), it causes reverse flow in the inlet duct, causing smoke spread into the inlet duct. As mentioned above, the reverse flow solely depends on the pressure difference between nodes in the simulations. The configurations of fan curve 1 without damper and fan curve 2 without damper have a smaller pressure difference and lower duct resistance, while the configurations of fan curve 1 with damper and fan curve 2 with damper have higher pressure difference and higher duct resistance. This results in the similar reverse flows in the simulations. In the period of 220 s–300 s, differences are visible. Fig. 7 reveals that with dampers, the pressure rise inside the compartment just exceeds the stalling pressure (i.e., 145 Pa), while this does not happen without dampers. As for the outlet flow, the configurations without dampers have lower duct resistance, yielding higher flow rates in the ducts. The maximum simulated reverse volume flow rate, occurring at approximately 330 s, is $157 \text{ m}^3/\text{h}$, vs. $147 \text{ m}^3/\text{h}$ in the experiment [9] (Figs. 8 a and Fig. 9 a). On the contrary, the volume flow rate in the outlet duct increases due to the pressure rise inside the enclosure, up to $165 \text{ m}^3/\text{h}$, vs. $173 \text{ m}^3/\text{h}$ in the experiment [9] (Figs. 8 b and Fig. 9 b). It can be seen from Figs. 8 and 9 that the combinations of fan curve 1 with damper and fan curve 2 with damper are better in

Fig. 8. Evolution of volume flow rate with time for the combinations of fan curve 1 and dampers. (a) Inlet duct; (b) Outlet duct. Experimental data reported by UMons [9–12].

Fig. 9. Evolution of volume flow rate with time for the combinations of fan curve 2 and dampers. (a) Inlet duct; (b) Outlet duct. Experimental data reported by UMons [9–12].

line with experimental results. This confirms that the damper has more influence on the results than the fan curve. Although the simplified fan curve can give reasonable flow rates without fire, it is also illustrated that leaving out the dampers from the simulations, the experimental results with fire cannot be reproduced accurately. All elements of the experiment should thus be included in order to gain reasonable simulation results, particularly with respect to pressure peaks and duct flow rates.

4.3. Influence of different leakage settings

As mentioned in Refs. [9–12], the leakage area is 0.0019 m^2 considering a discharge coefficient $C_d = 1$, which is the default value in FDS [17]. Another leakage area of 0.0026 m^2 is also used in Refs. [9,10] considering a discharge coefficient of C_d

= 0.7 when doing simulations with zone model CFAST. In the pre-

sent paper, both leakage areas are used in FDS simulations in order to reveal the sensitivity of leakage area, keeping the discharge coefficient fixed as $C_d = 1$. The simulation results are shown in Fig. 10, using fan curve 1 with damper, with different leakage areas and exponents. The influence of the leakage area and the pressure exponent on the pressure rise is small when there is a ventilation system. Due to the much lower pressure rise, compared to the configuration without ventilation, the difference between the actual pressure difference Δp and the reference pressure difference Δp_{ref} ($= 58$ Pa, as mentioned above), is smaller. This could reduce the influence of leakage pressure exponent on the pressure rise. It can be seen from Fig. 10 that the leakage area has little influence on the pressure, except for the peak value. The maximum pressures obtained with different leakage areas and exponents are shown in Fig. 11. The discrepancies between experimental and numerical results for the peak pressure are listed in Table 2. When the area is set to 0.0019 m^2 , the maximum pressure obtained is closest to the experimental value with leakage pressure exponent $n = 0.7$. This is in line with the experimental observation of leakage area and exponent in Ref. [11]. The exponent value $n = 0.5$ gives the best agreement with the experiment when the area is set to 0.0026 m^2 .

As mentioned in section 3.2, an additional simulation has been carried out, assigning the leakage area evenly to the two rooms, rather than using a uniform distribution over the walls and ceiling surface area (the latter leading to a larger leakage area in the fire room, compared to the adjacent room). The result is based on the case with discharge coefficient $C_d = 1$, reference leakage area of 0.0019 m^2 , leakage pressure exponent $n = 0.7$ and fan curve 1 with damper. Fig. 12 illustrates that the effect of leakage area distribution over the rooms on the pressure results is small.

4.4. Influence of the gap area on the pressure difference between rooms

The pressure profiles shown in this section are based on the reference leakage area of 0.0019 m^2 and the leakage pressure exponent $n = 0.7$, in accordance with the measured values. Fan curve 1 with damper is applied. The contour plots of the pressure in the compartment at different times are shown in Fig. 13. The gap area below the door, separating the fire room and the adjacent room, is set to 90 cm^2 (see section 2). It can be seen from the contour plots that the pressure inside the compartment is quite uniform and the pressure differences between the two rooms are very small. A visible difference can only be observed around the time of the peak value (see Fig. 14). The pressure variation inside the adjacent room is shown in Fig. 15 a. The pressure in the adjacent room also reaches a high value (exceeding 100 Pa) that can render safe evacuation impossible. In order to reduce the pressure rise in the adjacent room, the gap area between rooms can be reduced. Simulating the fire scenarios with $1/2$, $1/5$ and $1/10$ of the original gap area between rooms, i.e., reducing the gap area from 90 cm^2 to 9 cm^2 , the pressure rise in adjacent room reduces significantly, see Fig. 15 a. For the gap areas of 18 cm^2 and 9 cm^2 , the adjacent room is even in a negative pressure state during the initial stage because the amount of air extracted through the outlet duct exceeds inflow rate through the gap from the fire room. As the gap area at the bottom of the partition door decreases, the pressure difference between the two rooms becomes larger and larger (see Fig. 14). Table 3 summarizes how the maximum pressure difference between the fire room and the adjacent room increases as the gap area decreases. The maximum pressure difference between the rooms has a 19-fold increase for a 10-fold reduction in gap area (from 90 cm^2 to 9 cm^2). Therefore, reducing the gap area between rooms might significantly reduce the pressure rise in the adjacent room, thereby reducing the associated risk with respect to opening doors in the adjacent room. It is worth noting, though, that the reduction of gap area will also result in higher pressure rise in the fire room, see Fig. 15 b. The maximum pressure in the fire room increases from about 433 Pa to 850 Pa as the gap area reduces from 90 cm^2 to 9 cm^2 . For the sake of clarity it is mentioned that this section is a nu-

Fig. 10. Pressure variation as function of time in fire room. (a) Leakage area of 0.0019 m²; (b) Leakage area of 0.0026 m². Experimental data reported by UMons [9–12].

Fig. 11. The maximum pressure obtained with different leakage pressure exponents. Experimental data reported by UMons [9–12].

Table 2
Deviation of the simulated maximum pressure from the experimental value.

Leakage pressure exponent	Deviation from the experimental value	
	Leakage area 0.0019 m ²	Leakage area 0.0026 m ²
0.5	31.08%	0.38%
0.6	16.46%	8.88%
0.7	2.93%	23.64%

merical parameter study, not meant for use in practice (e.g., velocities become very high through small gaps, or it may become impossible to open other doors).

4.5. Influence of fire source modeling on temperature and pressure evolutions

The temperature profiles shown in this section are obtained using a leakage area of 0.0019 m², an exponent $n = 0.7$, and fan curve 1, including the effect of the damper. The temperature profiles in the fire room at heights 0.1 m (near the floor) and 2.1 m (near the ceiling) from the floor level are shown in Fig. 16. Two thermocouples with different bead diameters (0.5 and 1 mm) are set in the experiment [12]. It can be seen from Fig. 16 that the experimental results with bead diam-

Fig. 12. Fire room pressure variation based on different settings on the designation of leakage area. Experimental data reported by UMons [9–12].

eters of 0.5 mm and 1 mm are close to each other, but still slightly different. The simulation profiles at a given position are nearly identical, regardless of the bead diameter. The numerical temperature contour plots at different moments in time in the vertical plane through the thermocouple tree and the center of the burner are shown in Fig. 17. The vertical dashed line denotes the position of the thermocouple tree. It is noted that the maximum value on the y-axes in Fig. 16 is different in the 2 plots, which illustrates that both experimental data and numerical results reveal temperature stratification in the compartment. The stratification can also be seen from Fig. 17, but the temperature distribution over height shows differences between experimental and numerical results.

Fig. 16 shows that the simulated temperature rise is slower and reaches a lower peak when the fire source is modeled as a crib instead of a cube. As a result, somewhat better agreement is obtained with the experimental data, particularly during the growth phase of the fire. It is worth noting that in the UMons experiments, during the fire growth phase, the wooden crib was ignited from the bottom and the fire developed gradually from a localized position to the whole crib [9]. This is clearly better captured with the ‘crib’ simulation where a spreading fire was prescribed, as described in section 3.1. Nevertheless, there are still some discrepancies between the experimental and numerical temperature profiles, particularly in the decay period.

The improvement in the temperature predictions (especially during the growth phase of the fire) by simulating fire spread in the crib led to improved pressure predictions between 200 s and 300 s, see Fig. 18. However, the peak pressure is better predicted when the cubic fire source is used. This goes hand in hand with differences in tempera-

Fig. 13. Contour plots of pressure distribution at different times.

Fig. 14. Pressure difference between rooms as function of time for different gap areas. Experimental data reported by UMons [9–12].

ture (lower temperatures when the fire is modeled as a crib). Since the difference between the simulations is the fire source, it is considered that more heat is transferred and confined within the crib structure. This deserves more investigation in the future, but also indicates it is better to work with more controllable fire sources (liquid, or even better, gas fuel).

5. Conclusions

Fire-induced pressure variation in an air-tight compartment, representative of passive houses, was studied numerically in the present work. A mechanical ventilation system, including inlet and outlet ducts, was applied in well-confined rooms and a reverse flow in the sup-

ply duct was observed due to the high fire-induced pressure rise. The simulated fire-induced pressure in the fire room reached up to 433 Pa (vs. 421 Pa in the experiment [9]), which is high enough to impede occupant evacuation and firefighter rescue operations. The high pressure inside the confined compartment caused a maximum of 157 m³/h (vs. 147 m³/h in the experiment [9]) reverse flow in the supply duct, while it increased the flow in the extraction duct up to 165 m³/h (vs. 173 m³/h in the experiment [9]).

A mesh sensitivity study suggested that the mesh cell size of 0.1 m is sufficient for the CFD simulations of the compartment fire at hand. The trends and magnitudes of the experimental pressure variation and ventilation flow rates were well captured by the FDS simulations, while deviations were observed in the smoke temperature evolutions inside the fire room. This suggests that prescribing the heat release rate, in combination with accurate settings for leakage area and mechanical ventilation, is sufficient for the relatively uniform global quantity that pressure is, while the way in which this HRR is imposed can have a strong impact on local quantities with strong gradients, such as temperature in the case at hand.

It is important to include the dampers, present in the experimental set-up [9], in the simulations in order to obtain good agreement for the temporal evolution of pressure inside the compartment. Replacing their effect through a modified fan curve can reproduce the duct flow without fire, but is not sufficient when the fire is applied.

The influence of the leakage pressure exponent on the maximum fire-induced pressure rise in the simulations has been illustrated, with, as expected, a lower maximum pressure rise for higher pressure exponent (i.e., $n > 0.5$). Similar peak pressure levels have been obtained with different combinations of leakage area and leakage pressure exponent. The leakage pressure exponent $n = 0.7$ provides the maximum pressure in closest agreement with the experiments when the leakage area of 0.0019 m² is applied, which is in line with the experimental observations as reported in Refs. [9–12]. The value $n = 0.5$ seems optimum for the case in combination with the value 0.0026 m² for the leakage area.

The observed pressure rise in the adjacent room also reached such a high level that evacuation becomes impossible. Reducing the leakage area at the bottom of the door between the fire room and the adjacent room reduces the pressure levels in the adjacent room, but leads to even higher pressure levels in the fire room.

CRediT authorship contribution statement

Junyi Li: Conceptualization, Methodology, Software, Validation, Formal analysis, Investigation, Writing - original draft, Visualization. **Tarek Beji:** Conceptualization, Formal analysis, Writing - review & editing, Supervision. **Sylvain Brohez:** Validation, Investigation, Resources, Writing - review & editing. **Bart Merci:** Conceptualization, Formal analysis, Writing - review & editing, Supervision.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgments

The authors would like to thank the China Scholarship Council (CSC) for the financial support of the first author (Grant No. 201706420076), as well as the co-funding from Ghent University through BOF (Grant No. 01SC2018).

Fig. 15. Pressure variation as function of time for different gap areas. (a) Adjacent room; (b) Fire room. The red dashed lines correspond to the critical value of pressure which could hinder evacuation [3,4]. Experimental data reported by UMons [9–12]. (For interpretation of the references to colour in this figure legend, the reader is referred to the Web version of this article.)

Table 3

Maximum pressure differences under different gap areas between fire room and adjacent room.

Gap area (cm ²)	90	45	18	9
Maximum pressure difference (Pa)	42.43	142.67	488.99	798.22

Fig. 16. Temperature variations as function of time at heights 0.1 m (left) and 2.1 m (right) from floor level inside the fire room. Legends: Sim refers to simulation results. Exp refers to experimental results. TC and number refer to the thermocouple and its bead diameter. Experimental data from UMons [12].

Fig. 17. Numerical contour plots of temperature distribution in the vertical plane through the thermocouple tree (dashed line) and the center of the burner at different times. The maximum value is chosen for the visualization of smoke temperature. The fire temperature is much higher than this value.

Fig. 18. Fire room pressure variation using different fire source. Experimental data reported by UMons [9–12].

References

- [1] I Sartori, A G Hestnes, Energy use in the life cycle of conventional and low-energy buildings: a review article, *Energy Build.* 39 (2007) 249–257, doi:10.1016/j.enbuild.2006.07.001.

- [2] S Brohez, P Duhamel, Inwards doors blocked by fire induced overpressure in airtight apartment: a real case in Germany, *Chem. Eng. Trans.* 67 (2018) 25–30, doi:10.3303/CET1867005.
- [3] S Hostikka, R K Janardhan, U Riaz, T Sikanen, Fire-induced pressure and smoke spreading in mechanically ventilated buildings with air-tight envelopes, *Fire Saf. J.* 91 (2017) 380–388, doi:10.1016/j.firesaf.2017.04.006.
- [4] B Karlsson, J Quintiere, *Enclosure Fire Dynamics*, CRC press, 1999.
- [5] R K Janardhan, S Hostikka, Experiments and numerical simulations of pressure effects in apartment fires, *Fire Technol.* 53 (2017) 1353–1377, doi:10.1007/s10694-016-0641-z.
- [6] L Audouin, L Rigollet, H Pretrel, W Le Saux, M Rowekamp, OECD PRISME project: fires in confined and ventilated nuclear-type multi-compartments - overview and main experimental results, *Fire Saf. J.* 62 (2013) 80–101, doi:10.1016/j.firesaf.2013.07.008.
- [7] H Pretrel, W Le Saux, L Audouin, Pressure variations induced by a pool fire in a well-confined and force-ventilated compartment, *Fire Saf. J.* 52 (2012) 11–24, doi:10.1016/j.firesaf.2012.04.005.
- [8] Y Utiskul, J G Quintiere, A S Rangwala, B A Ringwelski, K Wakatsuki, T Naruse, Compartment fire phenomena under limited ventilation, *Fire Saf. J.* 40 (2005) 367–390, doi:10.1016/j.firesaf.2005.02.002.
- [9] I Caravita, *Fire Development in Passive Houses: Experimental Analysis and Validation of Theoretical Models* Master thesis, Department of civil, chemical, environmental and materials engineering, Università di Bologna, Bologna, Italy, 2017.
- [10] S Brohez, I Caravita, Overpressure induced by fires in airtight buildings, *J. Phys. Conf.* (2018), doi:10.1088/1742-6596/1107/4/042031 042031.
- [11] S Brohez, I Caravita, Fire induced pressure in passive houses: experiments and FDS validation, Ninth International Seminar on Fire and Explosion Hazards, 2019, pp. 524–533, doi:10.18720/spbpu/2/k19-132.
- [12] L Piret-Gérard, *Quid Des Dangers Pour Les Occupants Lors D'un Incendie Dans Un Milieu Confiné?* Master thesis, Faculté Polytechnique, Université de Mons, Mons, Belgium, 2017.
- [13] W K Chow, G W Zou, Numerical simulation of pressure changes in closed chamber fires, *Build. Environ.* 44 (2009) 1261–1275, doi:10.1016/j.buildenv.2008.09.016.
- [14] T Beji, F Bonte, B Merci, Numerical simulations of a mechanically-ventilated multi-compartment fire, *Fire Saf. Sci.* 11 (2014) 499–509, doi:10.3801/IAFSS.FSS.11-499.
- [15] J Wahlqvist, P van Hees, Evaluating methods for preventing smoke spread through ventilation systems using fire dynamics simulator, *Fire Mater.* 41 (2017) 625–645, doi:10.1002/fam.2404.
- [16] P J DiNenno, D Drysdale, C L Beyler, W D Walton, R L P Custer, J R Hall, J M Watts, *SFPE Handbook of Fire Protection Engineering*, third ed., National Fire Protection Association, 2002.
- [17] K McGrattan, S Hostikka, R McDermott, J Floyd, C Weinschenk, K Overholt, *Fire Dynamics Simulator User's Guide*, sixth ed., NIST and VTT, 2017, doi:10.6028/NIST.SP.1019.
- [18] J Li, T Beji, B Merci, CFD study of fire-induced pressure variation in a mechanically-ventilated air-tight compartment, Ninth International Seminar on Fire and Explosion Hazards, 2019, pp. 574–583, doi:10.18720/spbpu/2/k19-25.
- [19] D C O Cruz, *Numerical Simulations of Pressure Effects in the Event of Fire in Passive Houses*, Department of Flow, Heat and Combustion Mechanics, Ghent University, Ghent, Belgium, 2018.
- [20] ASTM Standard test method for determining air leakage rate by fan pressurization (E779-10) Standard Test Method for Determining Air Leakage Rate by Fan Pressurization (E779-10). United States <https://doi.org/10.1520/E0779-102010>
- [21] I S Walker, D J Wilson, M H Sherman, A comparison of the power law to quadratic formulations for air infiltration calculations, *Energy Build.* 27 (1998) 293–299, doi:10.1016/S0378-7788(97)00047-9.
- [22] K McGrattan, Verification and validation of selected fire models for nuclear power plant applications volume 7: fire dynamics simulator (FDS), U.S. Nuclear Regulatory Commission, Office of Nuclear Regulatory Research (RES) and Electric Power Research Institute (EPRI), Rockville, MD, Palo Alto, CA, 2007 Verification and Validation of Selected Fire Models for Nuclear Power Plant Applications Volume 7: Fire Dynamics Simulator (FDS).