Analysis of Convective Heat Losses in a Full-scale Compartment Fire Experiment

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ABSTRACT

This paper presents a methodology to quantify the convective heat losses in a full-scale compartment fire experiment, and presents an analysis of the key uncertainties in the experimental procedure, with the purpose of reducing uncertainty, and establishing robust error bars to the methodology. Uncertainties in the experimental flow measurements were assessed through a wind tunnel study of the bi-directional probe-transducer assembly at a range of Reynolds numbers. It was shown that at low Reynolds numbers, the probe-transducer assembly gains Reynolds dependence, for which the correction factor can be described as a piecewise function over the voltage of the differential pressure transducer used, with a minimum correction factor of 0.60. Numerical modelling in FDS is used to validate the assumptions driving the proposed methodology, and to provide the upper bound to the methodology. Due to the limited resolution of bi-directional probes in the experiment, the flow profile was mapped using a high resolution of thermocouples positioned near the opening, which allowed for the definition of the thermal interface height within the range (0.51-0.56)H_0, which was in proximity to the numerically determined neutral plane height within the range (0.61-0.64)H_0. Based on the numerical study and the literature, static inflows are assumed, and thus the convective heat losses through the opening are calculated. Similar to the underpinning work of the Compartment Fire Framework for well-ventilated fires, convective heat losses through the opening are calculated to account for 80% of the input fire. Sensitivity analyses of the hot layer temperatures and the interface height reveal a key dependency of the total convective heat losses to an accurate estimation of the interface height. The work presented herein will be utilised to characterise the fire dynamics of a set of full-scale experiments in order provide an accurate representation of thermal boundary conditions, either to the structure or out of the compartment, into adjacent spaces or windows.

KEYWORDS: Compartment fire, full-scale experiments, energy balance.

INTRODUCTION

The modern built-environment has experienced rapid development in the past few decades, with a key focus on coupling the need for new materials, architectural innovation, higher energy efficiency, and cost optimisation. Balancing these elements against fire safety considerations relies on an adequate understanding of fire dynamics derived from decades of research in compartment fires. Pioneering work undertaken from the 1940s to the 1980s conceived the Compartment Fire Framework; coupling fire dynamics with the characteristic parameters of the enclosure (i.e., geometry and ventilation factor) to quantify the maximum steady-state temperature and burning duration of a fire [1-3]. The applicability of the Compartment Fire Framework is limited to small cubic compartments with small openings, labelled by Thomas [4] as Regime I (ventilation-controlled fire). Beyond this condition, ventilation openings are sufficiently large such that they do not govern the thermal fields; this domain was defined as Regime II (fuel-controlled fire).

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As architecture evolved over the same period, the notion of small cubic compartments rapidly transformed to large open-plan volumes and interconnected spaces [5]. While the Compartment Fire Framework is still used today, its realms of applicability do not align with modern architecture. Experimental data gathered from full-scale fire tests have shown that the assumptions governing the Compartment Fire Framework do not hold validity for open-floor plan compartments as, to date, there is no theoretical link between Regime II and the gas-phase temperature or ventilation conditions [3, 6].

MOTIVATION

Experimental studies into Regime II fires have shown lower temperatures within the enclosure [2], hence Regime I is typically considered to be more conservative. The burning rate and thermal boundary condition can be quantified using available empirical data [1, 6] provided that the openings in a compartment are small enough to guarantee Regime I. Prior experimental studies into Regime II have not examined large open-floor plan compartments, for which that heat moving from the fire source may still be retained within the large compartment, continuing to heat the structural elements in the far-field, away from the fire source.

Recent full-scale Regime II fire experiments conducted over the last two decades [7-9] have demonstrated a divergence of the characteristic fire dynamics to the theory underpinning the Compartment Fire Framework. Severe gradients in spatial temperature distributions are evident, and the flow behaviour within the compartment can no longer be described as hydrostatic. To explore the characteristics of Regime II compartment fires, a series of full-scale tests was conducted at the BRE Burn Hall in Watford, UK in 2013. These experiments are referred to as the Edinburgh Tall Building Fire Tests (ETFT) and included over 2,200 sensors in a full-scale compartment and simulated a range of controlled fire modes (growing fire, travelling fire, and fully-developed fire) under different ventilation conditions [10], thereby representing the range of potential fire scenarios within an open-floor plan compartment typical of the contemporary architecture of tall buildings.

Recent efforts by Maluk et al. [11] focused on establishing an energy distribution analysis using data from the vast quantity of instrumentation in the ETFT. The purpose of this analysis was to demonstrate the link between the type of fire mode and the distribution of energy within the energy balance of a large open-floor plan compartment fire. Outcomes of this work showed that an appropriate calculation of energy distribution using experimental data was only possible for limited scenarios. These results found that, in a full-scale compartment fire experiment, regardless of ventilation conditions, convective heat losses through the outflow tend to dominate the characteristic fire dynamics. Irrespective of ventilation condition, proper quantification of the convective heat losses at the opening is an important consideration when considering the distribution of energy for compartment fires.

Maluk et al. [11] highlight that the quantification of the convective losses at the opening in compartment fires remains the greatest limitation to obtaining conservation of energy due to uncertainties in the flow measurements [11]. Flow measurements in full-scale experiments are typically taken using bi-directional probes [12] positioned at the opening, with the velocity determined using an associated differential pressure transducer (DPT). The use of bi-directional probes within the ETFT presents numerous uncertainties, which is attributed to the limited quantity of probes at each opening, inaccuracy of the DPTs, insufficient resolution of probes to capture the sharp velocity gradients in the outflow, and the positioning of the probes.

This work proposes a methodology to calculate the convective heat losses for a characteristic Regime II ETFT experiment, which is deemed to represent the simplest fire and ventilation mode out of the ETFT experimental programme. The approach presented will aim at reducing uncertainty in the quantification of convective heat losses, and establish robust error bars through a comprehensive analysis of flow fields and flow measurements that will be applicable for full-scale compartment fires. The goal of this work is to formulate an approach to calculate the convective heat losses leaving through the opening such that a refined energy distribution analysis can be conducted for the fire and ventilation modes from the ETFT programme [10].

EDINBURGH TALL BUILDING FIRE TESTS

Experimental description

A detailed description of the experimental compartment has been presented by Hidalgo et al. [10], however, a brief description is presented herein. The internal dimensions of the compartment were 17,800 mm x 4,900 mm x 2,000 mm. The dimensions of the compartment were selected to represent a scaled-down version of an open floor plan compartment typical in an office building. One side of the compartment was fully open with a 500 mm overhang, and fitted with a shutter system to control 15 independent segments of the opening, such that the opening factor could be varied. For this study, a scenario is taken with all the shutters left open, representing a well–ventilated compartment.

Key instruments used in the characterisation of the convective heat losses include over 1,800 Type K thermocouples (with a 1.5 mm bead), 30 bi-directional velocity probes [12], and 12 custom-built sand gas burners. The thermocouples were arranged into vertical arrays (also known as thermocouple trees), which were positioned in a grid spaced 600 mm along the depth of the enclosure and spaced 700 mm along the length of the enclosure. Each thermocouple array had thermocouples positioned at 300, 600, 900, 1200, 1400, 1600, 1800, and 1950 mm from the compartment floor. Further, thermocouple arrays were positioned at the centreline of each of the 15 opening segments at 180, 430, 680, 930, and 1180 mm from the compartment floor.

At each of the 15 opening segments, two bi-directional velocity probes were positioned at 220 mm and 1230 mm from the compartment floor. The lower probe was intended to measure the cold air flowing into the compartment, while the higher probe was intended to measure the hot gases leaving the compartment through the spill plume. The bi-directional probes were positioned parallel to the plane of the compartment floor, such that a characteristic horizontal gas flow is measured at each point.

The fire was controlled using six pairs of sand propane burners, uniformly distributed along the length of the compartment floor, with an associated pilot flame at each burner. The heat input was controlled using a mass flow controller, with a peak heat release rate of 2.5 MW for the characteristic experiment, with all burners turned on to simulate a fully-developed fire ($v_S \rightarrow \infty$).

Energy distribution analysis

Maluk et al. [11] detail an approach to conducting a temporal analysis of energy distribution for the experiments within the ETFT programme, through achieving global energy conservation, which can be defined as:

$$\dot{Q}_{fire} + \dot{Q}_{in,opening} - \dot{Q}_{out,opening} - \dot{Q}_{boundaries} = dQ_{CV}/dt \,, \quad (1)$$

where $\dot{Q}_{fire}$ is the heat release rate of the fire inside the compartment, $\dot{Q}_{in,opening}$ is the enthalpy of the inflow, $\dot{Q}_{out,opening}$ encompasses both the enthalpy of the outflow, $\dot{Q}_{out}$, and the radiation losses through the opening, $\dot{Q}_{rad}$, $\dot{Q}_{boundaries}$ is the net heat delivered to the solid boundaries (i.e., ceilings, walls, and floor), and dQ_{CV}/dt is the variation of energy within the control volume. Radiation losses through the opening were neglected, and the net heat lost through the shutters was assumed negligible due to the low emissivity and thermal conductivity of the material used for the shutters.

UNCERTAINTIES IN THE QUANTIFICATION OF THE CONVECTIVE HEAT LOSSES

The source of uncertainty in the quantification of the convective heat losses in the energy distribution analysis by Maluk et al. [11] is attributed to two aspects; uncertainty in the experimental methodology, and uncertainty in the measurement of flow.

Experimental methodology uncertainties

Maluk et al. [11] attribute a large portion of uncertainty to the use of two horizontal bi-directional probes positioned at the opening, providing poor resolution to capture the spillage of the hot outflow gases, as sharp velocity gradients are typically noted within Regime II outflows [13].

A single bi-directional probe was assigned to the inflow and outflow respectively, and a linear flow profile was assumed in order to estimate the neutral plane height, H_N. This assumption represents a source of error since there is inherent bias for the neutral plane height to be lower, as the outflow probe measures significantly higher velocities than the inflow probe. The estimation of the convective heat losses through the opening was determined using trapezoids to discretise the mass flows of the outflow and the associated gas-phase temperatures. Therefore, an underestimation of the neutral plane height results in higher convective heat losses as the volumetric flow rate of hot gases exiting the compartment is higher than the volumetric inflow of cold gases.

Flow measurement uncertainties

The use of bi-directional probes designed by McCaffrey and Heskestad [12] is commonplace and has been successfully adopted in a range of compartment fire experiments; typically to measure the speeds of ceiling jets or flows through an opening, where the flow direction is generally known. The determination of the local velocity using the bi-directional probe follows that of a pitot tube by measuring the pressure differential between the front and rear side of the probe.

$$V = C\sqrt{2\Delta P/\rho}\,, \tag{2}$$

where C is the probe constant as a function of the Reynolds number, and ρ is the local gas density determined as a function of the local temperature measured with the associated thermocouple. The probe constant acts as a correction factor for the design of the probe head, and has been shown to asymptotically converge to 0.92 for flows ranging from $40 < Re < 3800$, with a relative accuracy of ±5%. The probe constant corresponds to the error associated with the head of the probe, which has been the subject of many studies, with numerous improvements to designs presented, e.g. [14].

McCaffrey and Heskestad [12] note that at very low speeds, difficulties in velocity measurements can be associated with the resolution of the DPT, as the voltage differential readings become very small. This presents a significant source of error within the calculation of velocities as the field DPTs used (Gems 5266 model) in the experiment are specified to operate to ±50 Pa, with an accuracy of ±1% full-scale error. Differential pressures at compartment openings have been shown in numerous experiments to be small [15, 16], irrespective of the ventilation condition, and therefore it is likely in some cases that the error embedded within the DPT can exceed the actual pressure differentials at the inflow and outflow.

METHODOLOGY OF THE STUDY

A revised methodology to calculate the convective heat losses is presented herein. This analysis is based on understanding the key sources of experimental uncertainties governing the calculation, with the uncertainties explored as they arise.

The methodology used to estimate the net enthalpy loss through the opening of a well-ventilated compartment fire is based upon the following key assumptions:

(1) Stratification effects at the inflow are negligible, and thus inflow velocities are treated as constant [13].

(2) The height of the thermal discontinuity from hot to cold, also known as the thermal interface height (H_I), is close to the location of the neutral plane height (H_N).

(3) Gas temperature measurements at the opening present a misleading value for the location of the thermal interface height within the compartment as the temperatures at the opening are not stratified. The opening plane lies within the shear region [17], where the cold gases flowing into the compartment can shear gases from the hot layer, thus mixing some hot gases within the cold layer. Further, thermocouples at the opening are subject to convection losses to cold air entrainment, and radiation losses to the ambient surroundings.

Studies on pressure measurements in medium-scale enclosures by McCaffrey and Rockett [13] and Steckler et al. [17] have shown that the inflow tends to be characterised by static pressures, and therefore the velocity gradient of the inflow is assumed to be negligible. Drag induced flow emanating from the entrainment of cold air to the plume is not expected to be significant as the burners are not located close to the openings as observed. The flow rate can be lumped volumetrically as follows:

$$\dot{m}_{in} = \rho_{air}(T_1)\dot{V}_{in}, \tag{3}$$

$$\dot{V}_{in} = H_I v_{in} w_0, \tag{4}$$

where $\rho_{air}(T_1)$ is the density of air assumed at the temperature of the lowest thermocouple, $\dot{V}_{in}$ is the volumetric flow rate, H_I is the interface height, u_{in} is the inflow velocity given by the lower bi-directional probe, and w_0 is the width of each opening (1.1 m).

Through the principle of conservation of mass, and under the assumption that $\dot{m}_{in}, \dot{m}_{out} \gg \dot{m}_{fuel}$, the mass flow rate of the outflow must be equal to the inflow,

$$\dot{m}_{in} = \dot{m}_{out}. \tag{4}$$

The enthalpy of the hot and cold gases leaving and entering the compartment can be defined as:

$$\dot{Q}_{out} = \dot{m}_{out} c_p \left(T_H\right) T_H, \tag{6}$$

$$\dot{Q}_{in} = \dot{m}_{out} c_p \left(T_1\right) T_1, \tag{7}$$

where c_p is the specific heat capacity of the gases as a function of temperature (kJ.kg^{-1}.K^{-1}), T_1 is the temperature of the lowest thermocouple (K), and T_H is the hot gas layer temperature (K).

In order to estimate the total convective heat losses, the inflow velocity, interface height, and the temperature of the hot layer must be determined using the instrumentation within the experiments.

Measurement correction for gas flow experimental data at the opening

Fire-induced gas flows at the openings for large compartment fires are typically very low-speed, ranging from 0.2 to 3 m/s [13, 16]. The bi-directional probes used in the experiments by Hidalgo et al. [10] were initially calibrated individually with an accurate pressure transducer, resulting in similar correction factors to those presented by McCaffrey and Heskestad [12].

To determine the influence of error associated with the field DPT, the bi-directional probe and the DPT are treated as an assembly, and a wind tunnel study is undertaken to calculate the revised correction factor for the "probe-transducer" assembly. The wind tunnel speed is calculated using a high-resolution manometer and pitot tube. The probe-transducer correction factor is defined as:

$$C = v_{pitot} \big/ v_{assembly} \, . \tag{8}$$

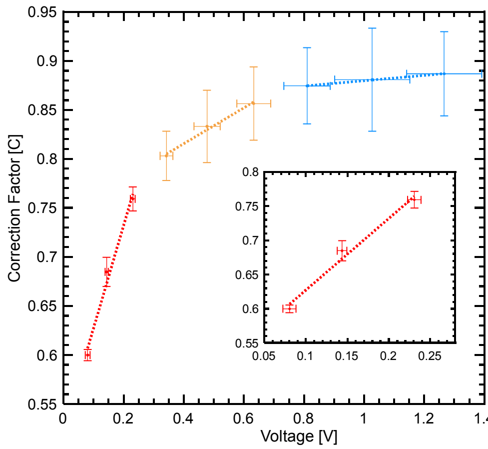

Fig. 1. Wind tunnel experiments of the probe-transducer assemblies conducted under a range of Reynolds numbers ranging from Re = 890 to Re = 5100, with the correction factor presented as a function of voltage.

Table 1. Summary of best-fit regressions for the probe-transducer assembly

890 < Re < 2400	2401 < Re < 3480	3481 < Re < 5100
C = 0.85V + 0.56	C = 0.18V + 0.74	C = 0.03V + 0.85

[a] Reynolds number as a function of the probe head diameter and wind tunnel speed: $\mathrm{Re} = \rho v D / \mu$

As shown in Fig. 1, at the lowest Reynolds number (Re = 890), the correction factor is C = 0.60, indicating that the DPT records higher pressure differentials than the pitot tube. This correction factor is a significant departure from the probe constant reported by McCaffrey and Heskestad [12]. While the correction factor can be expressed as a linear regression, the wind tunnel experiments demonstrate that the correction factor can be described as a piecewise function over the Reynolds number. The linear regressions to determine the correction factors as a function of the voltage response of the DPT are presented in Table 1. It is important to note that these results only apply to the Gems 5266 DPT used, and the specific design of the bi-directional probe reported by McCaffrey and Heskestad [12].

At high Reynolds flows (3480 < Re < 5100), the probe-transducer correction factor plateaus, and at higher Reynolds numbers than 5100, the correction factor asymptotes towards 0.92; the probe constant reported by McCaffrey and Heskestad [12]. These results indicate that the correction factor is Reynolds dependent at lower Reynolds numbers, and the error at low Reynolds numbers is attributed to the DPT, rather than the design of the probe head.

Estimation of the thermal interface height

Instead of using bi-directional probes to determine the interface height, thermocouple arrays positioned in-depth from the opening plane of the compartment are used. In each vertical thermocouple array, eight discrete points (i.e., thermocouples) are used to map the direction of the flow.

Flow fields for Regime I compartments are typically characterised through a hydrostatic model, for which the pressure differential is determined using temperature differentials from the thermocouples positioned at the plane of the opening. Due to the aforementioned issues with using thermocouples located at the opening plane, the interface height is defined using the thermocouple row positioned in-depth, y = 0.4 m from the opening plane. The temperature readings were benchmarked against other thermocouple rows in-depth at y = 1.1 m, and y = 1.8 m with no major differences in temperatures, thereby defining this region within the stratified region. The thermocouple profiles between the opening plane and the stratified region located at y = 0.4 m are shown in Fig. 2a.

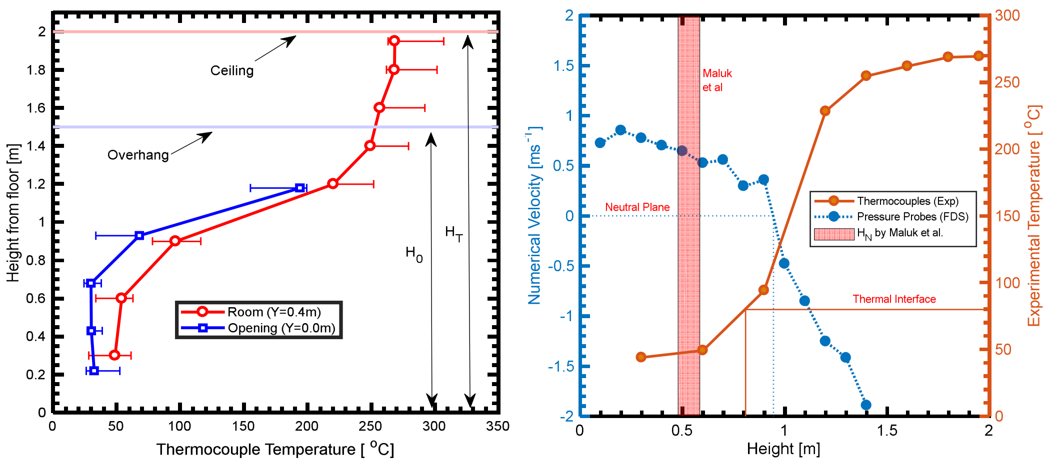

Fig. 2. Temperature and velocity profiles at opening segment 7 (i.e., the centreline of the compartment. (a) Thermocouple arrays located at the opening plane, and in-depth respectively at t = 1000 s ($\dot{Q}_{fire}$ = 2.5 MW). (b) Comparison of the time-averaged experimental thermal interface height at y = 0.4 m, and the numerical neutral plane height, H_N, is determined using the custom horizontal component velocities measurements in FDS. Positive velocities indicate inflow, while negative velocities indicate outflow.

The interface height for each vertical thermocouple array is determined using the procedure detailed by Janssens and Tran [18], which has been shown to consistently estimate the interface height for large compartments [9]. The hot gas layer temperature is then determined by averaging the integral of the temperature profile from the interface height to the ceiling as per Eq. 9. Likewise, the cold gas layer temperature is determined through averaging the integral of the temperature profile from the floor to the interface height as per Eq. 10.

$$T_H = \frac{1}{(H_T - H_I)} \int_{H_I}^{H_T} T(z)dz \,, \tag{9}$$

$$T_C = \frac{1}{H_I} \int_0^{H_I} T(z)dz \,. \tag{10}$$

Numerical modelling comparisons of the thermocouples and gas-phase temperatures, which are not presented herein, show a 1-2% deviation in the hot gas layer temperatures, and therefore the thermocouples were not corrected for radiation. Interface heights for the thermocouple arrays located at the opening plane, and in-depth displayed in Fig. 2a are calculated to be 0.81 and 0.91 m respectively.

The average steady-state interface height for the centreline opening segment 7 is shown in Fig. 2b, although the average steady-state upper and lower bounds range from 0.51 H_0 to 0.56 H_0 (i.e., 51% and 56% of the opening height), within the ranges reported by [19]. The neutral plane height generally lies above the interface height [20], and therefore will serve as the upper bound to the proposed methodology.

Numerical model description

A CFD model was formulated using the Fire Dynamics Simulator version 6.6.0 (FDS) [21] to study the flow fields of the experiment in detail. The objective of the numerical modelling work was to bound the error bars for the proposed methodology, by validating the assumptions used in the methodology and quantifying the neutral plane height. The neutral plane height is a key parameter dictating the volumetric exchange of convective gases and will be used to provide the upper bound to the methodology.

Table 2. Specification of the simulation setup constructed in FDS v.6.6.0

Simulation parameter	Simulation Values
Domain Range [*x0, xf, y0, yf, z0, zf*] (m)	Four meshes [-2.0, 20, -3.0, 5.3, 0.0, 6.0]
Grid Size [Length, Width, Height] (m)	0.1 x 0.1 x 0.1 (1,320,000 cells)
Simulation Time (s)	1491
Fuel	Propane (C_3H_8)
Heat of Combustion (kJ/g)	46.45
Soot Yield (-)	0.024

The compartment geometry, ventilation factor, material composition of the walls, and positioning of the burners is identical to the experimental setup. The fire is modelled using propane burners with an input burning rate that is set to follow the progression of propane flow recorded by the mass flow controllers in the experiment. The Froude number is sufficiently low to treat each burner as a small pool fire, and therefore the radiative fraction is calculated to be 0.21. The instrumentation mirrors the experiment, although additional component flow velocity devices are positioned at the centreline of each opening in vertical arrays spaced equally at intervals of 0.1 m.

Relatively accurate hot gas layer temperature predictions for the Regime II characteristic experiment are shown in Fig 3. The hot gas layer height and temperature are determined using thermocouple outputs in FDS, and the latter is normalised against the experimentally determined hot gas layer temperature.

While the modelling of energy transport is validated for these types of scenarios, the aforementioned experimental uncertainties noted with probe-transducer assembly can make a direct validation of velocities difficult to achieve.

The temperature profiles within the compartment in Fig. 4a highlight the sharp temperature gradients between the opening plane and the in-depth compartment temperatures, especially near the opening, indicating that thermal stratification of the opening plane is not well defined. The regions in-depth from the opening plane, aside from the locations of the backward tilting fire plumes appear to be a good representation of the stably stratified region. Observations of the velocity in Fig. 4b show high vertical velocity components just underneath the overhang, with peak flows of 3 m/s. This contrasts the actual location of the probe, which was positioned at $0.54\,H_0$, where velocities are lower, with lower vertical components. The effect of cold air entrainment and mixing between the hot-cold layer transitions are also observed towards the opening plane. Surprisingly, vertical components at the bottom of the opening are quite high, likely driven by the entrainment of the plume. Inflow velocities of relatively constant magnitude (as dictated by the length of the arrows), and a well-defined neutral point (zero-velocity vector) is observed around $0.61\,H_0$.

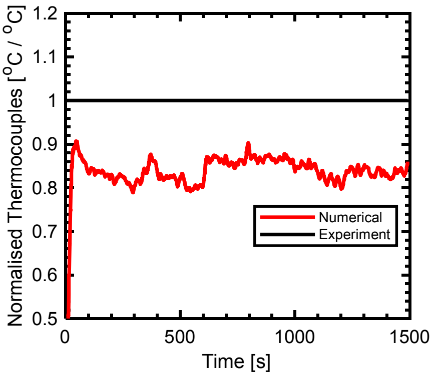

Fig. 3. Normalised plot of the numerically determined hot gas layer temperature to the experimental hot gas layer temperature. The numerical model generally underpredicts the experimental results by 10-15%.

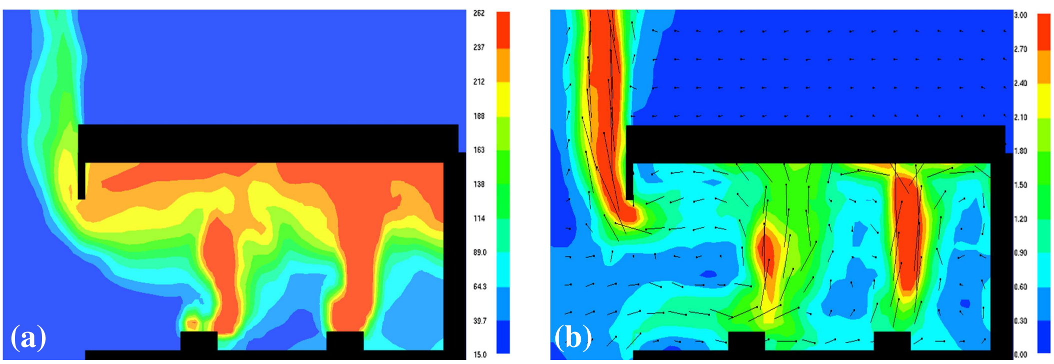

Fig. 4. Results of the numerical simulation of flow velocities in FDS for opening segment 7 (centreline of the enclosure) at t = 1000 s ($\dot{Q}_{fire}$ = 2.5 MW), and time-averaged over 5-second intervals.

(a) Cross-section view of the gas-phase temperatures; (b) Cross-section view of the velocity vectors.

Using the additional velocity measurements collected numerically in FDS, the flow is mapped out with a high degree of resolution in order to identify the neutral plane height, which occurs at the point at which the horizontal velocity component is zero. The neutral plane heights are lower at both ends of the compartment, giving the flow of the compartment the distinct "U" shape observed by Hamins [16]. The maximum deviation of the neutral plane height between the opening segments ranges from $0.60\,H_0$ to $0.64\,H_0$, representing little deviation when compared to the interface heights. The maximum deviation of the numerical inflow and outflow speeds in Fig. 4b is 0.55 m/s and 1.41 m/s, respectively. Below the neutral plane, the variation of inflow velocity is significantly lower than the outflow. Therefore, the assumption of a static mass inflow rate, $\dot{m}_{in}$ can be seen to be valid and will be used to establish a mass balance in the compartment.

Calculation of the total convective heat losses

Using the corrected velocities, and the interface height, the total convective heat losses for all 15 opening segments in the compartment is shown in Eq. 11, and the results are presented in Fig. 5.

$$\sum_{1}^{15} \dot{Q}_{net,loss} = \dot{Q}_{in} - \dot{Q}_{out} \,. \tag{11}$$

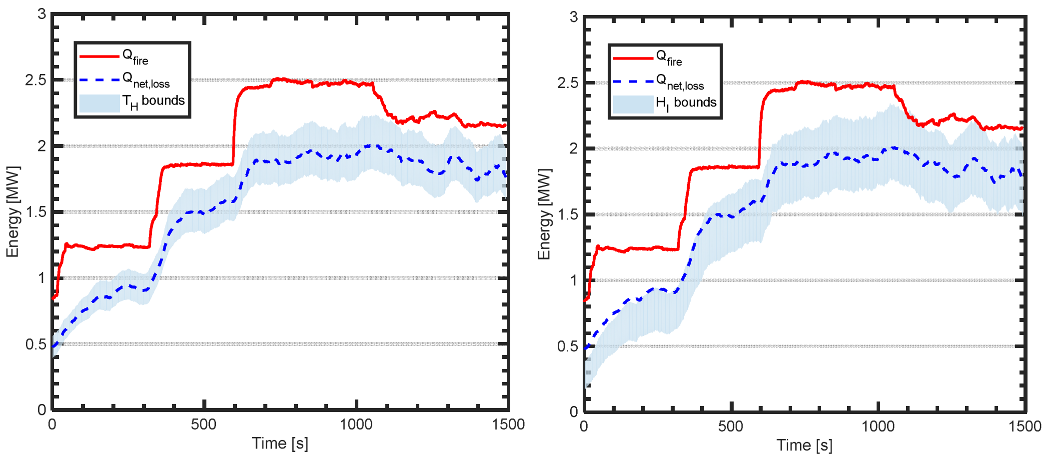

Fig. 5. Integrated convective heat losses across all fifteen opening segments. (a) the error bars show the response of the convective heat losses to the maximum and minimum thermocouple reading within the region of T_H . (b) the error bars show the response of the convective heat losses to the maximum and minimum bounds of the interface heights averaged across all of the opening segments during the steady-state portion of the experiment.

Figure 5 shows the application of the methodology to determine the net convective heat losses through all of the opening segments in the experiment. It can be seen that the convective heat losses account for 80% of the total energy input from the burners, higher than the estimate of 64% by Maluk et al. [11]. This estimate is consistent with the Regime II experiments with similar ventilation opening factors (ϕ = 4.1) by Harmathy [22] and Majdalani [15], who estimate convective heat losses representing 79%, 80%, and 85% of the total energy input respectively.

The error bars in Fig. 5a represent the sensitivities of the applied methodology to the hot gas layer temperatures used in the calculation of the convective heat losses, and to the sensitivity to the interface height. The maximum deviation from the hot gas layer temperature is 1.1 T_H and 0.92 T_H . This result demonstrates relative insensitivity of convective heat losses to the hot gas layer temperature. Therefore, it is not expected that radiation errors in the hot gas phase temperatures will influence the results significantly, as radiation errors have been shown to bias cold layer temperatures more [23]. Figure 5b highlights the sensitivity of the methodology to the deviations in the interface height, with the maximum deviation from the average interface height over a time average period of the steady-state portion of the experiment, ranges from 0.50 H_0 and 0.56 H_0 , which is below the numerical ranges of the neutral plane. It is clear that the methodology is relatively sensitive to the small ranges of interface heights imposed, therefore the need for an accurate estimate of the interface height is confirmed. The accuracy of the numerical method used to estimate the thermal interface height is linked to the accuracy of the temperature profile of the vertical thermocouple arrays.

CONCLUSIONS

Recent full-scale experiments have shown that the fire dynamics of Regime II can be defined through the convective heat losses that are produced by the fire. While the characterisation of the thermal fields is generally achievable using thermocouples and thin-skin calorimeters, quantification of the flow fields remains a key limitation in the quantification of the convective losses, and therefore the full characterisation of full-scale compartment fire dynamics. This paper has proposed a methodology to estimate the total convective heat losses by using (1) gas inflow measurements using the bi-directional probes and assuming equal to gas outflow and (2) by determining the thermal interface height and the gas temperature of the outflow using measurements of thermocouples located within the temperature stratified region of the compartment. Through a series of numerically validated assumptions, conservation of mass can be applied in order to determine the total convective heat losses with reduced uncertainty, and quantified error.

The wind tunnel study of the bi-directional probes and DPTs demonstrates that, at low Reynolds flows (Re = 890) typical of full-scale compartment fires, the probe correction factor shifts from the probe-head to the probe-transducer assembly, with the correction factor shifting to C = 0.60 at Re = 890. As the Reynolds number increases, the correction factor is determined through a piecewise function using the voltage response of the DPT.

Using the high density of thermocouples instead of the two bi-directional probes to map the shape of the flow, the average thermal interface height during the steady-state period of the experiment is defined within the range of (0.51–0.56) H_0, consistent with the literature. This was found to be lower, but still within close proximity to the numerically determined average neutral plane height which ranges from 0.61 H_0 to 0.64 H_0 , thereby representing the upper bound to the problem.

Applying the corrections of the experimental techniques, and validated assumptions, the total convective heat losses were found to account for up to 80% of characteristic fire dynamics within the compartment, demonstrating an agreement to the literature underpinning the Compartment Fire Framework. The sensitivity analysis of the hot gas layer temperature shows ±10% variation of heat losses within the range of (0.92–1.1) T_H , whilst the sensitivity of the interface height shows ±13% variation within the shorter range of (0.51–0.56) H_0 , indicating that the interface height can significantly alter the ratio of convective heat loss from the compartment, and the estimation should be refined such that proximity to the neutral plane height is attained.

Ongoing work carried by the authors focuses on applying this methodology to a series of full-scale experimental datasets collected under the ‘ETFT’ programme, in order to provide full quantification of the energy distribution within a range of compartment fires; varying in ventilation condition and fire modes. Through the characterisation of the fire dynamics governing full-scale compartment fires, it is expected that an adequate representation of the thermal boundary conditions for describing the structural response during a fire can be made, irrespective of the ventilation condition.

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