Scaling of internal wall temperatures in enclosure fires

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Abstract

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The scaling of enclosure fires, including internal wall temperatures and heat fluxes, are thoroughly analyzed and a method of scaling internal wall temperatures is proposed. Two series of room fires were tested in three different scales, i.e. full scale (1:1), medium scale (1:2) and small scale (1:3.5), according to the theory. The fire source was either placed at the center or in the corner of the enclosures.

The measured internal wall temperatures, incident heat fluxes, gas temperatures, gas velocities and gas concentrations in different scales are compared and analyzed. The results show that the proposed scaling method is able to scale the internal wall temperature very well, especially in medium scale. The incident heat fluxes and gas velocities are also scaled very well. The gas temperatures and gas concentrations are scaled generally relatively well.

The calculation of upper layer gas temperatures in the compartment fires are investigated. The results show that both the maximum ceiling excess gas temperatures and average ceiling excess gas temperatures at the upper layer for corner fires are higher than those for center fires, and the ratio is 1.25 for both the maximum and the average ceiling excess gas temperatures. The maximum excess gas temperature normally exists right above the fire and is approximately 19 % higher than the average values for both center fires and corner fires. The widely used MQH equation cannot directly be used to estimate the gas temperatures in compartments covered with insulating materials. A correlation has been proposed to estimate the upper layer gas temperatures in compartments covered by similar insulating materials for both center fires and corner fires. The equation is expected to be valid for gas temperatures below 700 °C or slightly higher, but not over 1000 °C.

Key words: scaling; enclosure fire; internal wall temperature; wall material; gas temperature

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Preface

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Summary

A total of 7 tests were carried out in a full scale room, a medium scale room and a small scale room. The main focus of these tests is scaling of internal wall temperatures inside the wall materials.

The scaling of enclosure fires, including internal wall temperatures and heat fluxes, is thoroughly analyzed and a method for scaling internal wall temperatures is proposed. Two series of room fire tests were carried out in three different scales, i.e. full scale (1:1), medium scale (1:2) and small scale (1:3.5), according to the theory. The fire source was either placed at the center or in the corner of the enclosures.

The measured internal wall temperatures in different scales of enclosure fires were compared and analyzed. The results show that the proposed scaling method is able to scale internal wall temperatures very well, especially in medium scale. The gas temperatures, gas concentrations, gas flows through the door and incident heat fluxes generally are also scaled well.

The gas temperatures at upper layer are also investigated. The results show that both maximum ceiling excess gas temperatures and average ceiling excess gas temperatures for corner fires are higher than those for center fires, and the ratio is 1.25 for both the maximum and the average ceiling excess gas temperatures. The maximum excess gas temperature normally exists right above the fire and is approximately 19 % higher than the average values for both center fires and corner fires. For center fires, the widely used MQH equation can be used to estimate the maximum ceiling gas temperatures, but it significantly overestimates the average gas temperatures. For corner fires, the MQH equation can be used to approximately estimate the average ceiling gas temperatures, but it significantly underestimates the maximum ceiling gas temperatures. An correlation has been proposed to estimate the upper layer gas temperatures in compartments covered by similar insulating materials for both center fires and corner fires. The equation is expected to be valid for gas temperatures below 700 °C or slightly higher, but not over 1000 °C. The proposed equation could only be used for estimation of upper layer gas temperatures in compartments covered by similar insulating materials.

Nomenclature

a	thermal diffusivity (m^2/s)	<i>Greek symbols</i>	
A	area (m^2)	ρ	gas density (kg/m^3)
A_b	bounding area of hot gases (m^2)	β	expansion coefficient ($1/\text{K}$)
c	heat capacity ($\text{kJ}/\text{kg} \cdot \text{K}$)	δ	physical or thermal thickness (m)
C_d	flow coefficient	ε	emissivity
$C_{heat,\beta}$	lumped heat capacity ($\text{kJ}/\text{kg} \cdot \text{K}$)	σ	Stefan-Boltzmann constant ($\text{kW}/\text{m}^2 \cdot \text{K}^4$)
$=1/3$		κ	absorption coefficient ($1/\text{m}$)
C_2	coefficient	ϕ	ratio of the effective radiation area to the area of opening
D	burner diameter (m)		
E	energy content (kJ)	<i>Sup and subscript</i>	
g	gravity acceleration (m/s^2)	a	ambient
h	heat transfer coefficient ($\text{kJ}/\text{m}^2 \cdot \text{K}$)	b	bounding
H	height of an opening (m)	bw	backside of the wall
ΔH_c	heat of combustion (kJ/kg)	c	convection
k	thermal conductivity ($\text{kW}/\text{m} \cdot \text{K}$)	f	Fuel
K_{cond}	Correction factor ($\text{kW}/\text{m} \cdot \text{K}$)	F	full scale
l	length scale (m)	g	Gas
L_f	flame length (m)	i	ith step
L_m	mean beam length (m)	inc	incident heat flux
m	fuel mass (kg)	k	conduction
$\dot{m}_g''$	mass flow rate ($\text{kg}/\text{m}^2 \cdot \text{s}$)	m	mean
Nu	Nusselt Number	M	model scale
P	pressure (Pa)	net	net value
ΔP	pressure difference (Pa)	o	opening or reference
Pr	Prandtl number	PT	Plate thermometer
Q	heat release rate (kW)	r	radiation
$\dot{q}''$	heat flux (kW/m^2)	s	Solid
R	heat resistance ($\text{m}^2 \cdot \text{K}/\text{kJ}$)	t	Total
$\bar{R}$	lumped resistance ($\text{m}^2 \cdot \text{K}/\text{kJ}$)	w	Wall
Ra_l	Rayleigh number		
Re	Reynold Number	<i>Abbreviations</i>	
t	time (s)	BP	bi-directional probe
T	temperature (K)	G	gas analysis
ΔT	excess temperature (K)	PT	plate thermometer
u	velocity (m/s)	TP	thermocouple pile
ν	kinematic viscosity (m^2/s)	TS	thermocouple series
V	volume (m^3)		
V_b	volume of the hot gases (m^3)		
V_{is}	visibility (m)		
V_t	total volume of fuels (m^3)		
Y	gas concentration (kg/kg)		
z	depth below surface (m)		

1 Introduction

Due to the complexity of the fire phenomenon, full scale tests are still the main and the most credible tools while investigating any new fire related phenomenon or any new fire protection system. The full scale tests provide the most valuable data, however, are also very costly, especially in some scenarios at a huge scale. Model scale tests are therefore a tempting alternative to the full scale tests.

In the past few decades, the physical scaling method has been widely used in fire community. Its application nearly covers every field of fire research, from free plume to fire suppression in tunnel fires. Despite its simplifications in applications, the scaling technique has significantly improved our understanding of fire dynamics. Heskestad [1] reviewed the scaling techniques mainly including the pressure modeling and the Froude modeling that were used in the fire community. Quintiere [2] also reviewed the scaling applications in fire research on ceiling jets, burning rate, flame spread and enclosure fires. Perricone et al. [3] investigated the thermal response of steel tube covered by insulating materials, however, the scaling laws used for the thick insulating materials could be questioned. Cross and Xin [4] examined the scaling of wood crib fires and found good agreement between the different scales. Scaling of water-based fire suppression systems has also been conducted in open and enclosure fires. Heskestad [5,6] carried out a series of gas and pool fire suppression tests to investigate the credibility of the scaling of the interaction of water sprays and flames, and obtained a simple correlation for extinguishment of gas and pool fires using water sprays. Quintiere et al.'s work [7] shows that the scaling of pool and gas fires works well, however, the comparison of results in rack-storage fires between model and full scale doesn't show good correlation. Yu et al. [8-10] tested and investigated the scaling of suppression of gas fires and pool fires using water mist systems and obtained good agreement between model scale and full scale.

SP Fire Technology has carried out extensive model scale tunnel fire tests in the past decade. Ingason and Li investigated the key fire parameters and smoke control in model scale tunnels with longitudinal ventilation [11] and with point extraction ventilation [12]. Ingason [13] also carried out a series of 1:10 model scale railcar tunnel fire tests to investigate the effect of openings on the fire sizes. Li et al. [14-21] carried out several series of model scale tunnel fire tests to investigate the critical velocity, backlayering length, maximum ceiling gas temperature, smoke control in cross-passages, and smoke control in rescue stations in long railway tunnels. Lönnermark et al. [22] carried out a 1:3 model scale metrocar fire tests for the full scale fire tests in Brunsberg tunnel [23]. Ingason [24] tested the water spray system in tunnel fires using hollow cone nozzles and wood crib fires. Deluge system and water curtain system were also tested. Li and Ingason [25] investigated the automatic water spray system in tunnel fires using full cone nozzles and wood crib fires. Response time of individual sprinklers was modeled by a scaling theory.

Despite much work on the scaling of fires in the open and the enclosures, the credibility of scaling of the internal wall temperatures has not been explored. The internal wall temperatures have attracted our special attention partly due to the relation to fire resistance of the walls. For example, in case of a fire in a lightweight construction, the temperature at the load-bearing structure should not exceed a certain value, e.g. 140 °C, or else the structure will lose strength and collapse [26,27]. Further, the scaling of wall heat fluxes affects the gas temperatures in the enclosures.

Therefore, the scaling of internal wall temperatures needs to be investigated carefully in order to be able to trust experimental data from scaled enclosure fires, which is the main focus of this project.

In this report, the scaling of enclosure fires including internal wall temperatures and heat fluxes are thoroughly analyzed and a method for scaling of internal wall temperatures is proposed. According to the theory, compartment fire tests were carried out in different scales. Based on the tests data, the scaling of internal wall temperatures, incident heat fluxes, gas temperatures, gas velocities and gas concentrations are analyzed. Further, the calculation of upper layer gas temperatures in the compartment fires are investigated.

2 Scaling theory

The widely used and well known Froude scaling technique has been applied in this project. Although it is impossible and in most cases not necessary to preserve all the terms obtained by scaling theory simultaneously, the terms that are most important and most related to the study are preserved.

The Froude scaling has been used widely in enclosure fire research and results from model scale tests could fit large scale data well, see references [2,5]. A large amount of model scale tunnel fire tests carried out at SP also show that there is good agreement between model scale and large scale test results on many focused issues [11-21,28].

However, in most of the model scale tests that have been carried out worldwide, the thermal properties of the involved material were not taken into account. One reason could be that for those studies the internal wall temperatures were not the focus, however, the wall materials have strong influence on the scaling of the fires, especially at small scale.

In this project, a theoretical scaling study on the heat conduction inside the walls and heat fluxes has been conducted, see Appendix A. The scaling laws for internal wall temperatures are proposed and are investigated in the following based on the data obtained from two series of fire tests carried out in different scales of enclosures.

The rooms were built in a scale of 1:1, 1:2 and 1:3.5 respectively, which means that the sizes of the rooms are scaled geometrically according to the scaling ratios. The scaling of the key parameters is presented in the following. The detailed information about scaling of heat fluxes and internal wall temperatures can be found in Appendix A.

It is shown in Table 1 that gas temperatures and internal wall temperatures should be the same in all scales. However, it should be kept in mind that the correlations shown in Table 1 only correspond to the perfect scaling. In reality, there are always some parameters that cannot be scaled well and thus induce errors to the other related parameters. The key error in this type of scaling is produced by the scaling of heat fluxes.

Note that the scaling of these parameters is obtained based on the assumption of the same heat of combustion. If different fuels are used in model scale, some parameters will not be scaled as shown in Table 1.

There are two source terms in the controlling equations for the mass transfer and heat transfer in an enclosure fire, that is, the heat release rate and the mass loss rate, respectively. If we focus on the scaling of the mass loss rate, the gas concentration can still not be scaled well due to the failure of scaling of the buoyancy force and gas temperature. In such cases, the only solution is to focus on the scaling of the heat release rate, regardless of the species production. As a consequence, the heat release rate, energy content, velocity, time, temperature, and pressure will approximately scale as shown in Table 1. However, the overall gas concentration cannot scale well, and according to the scaling of the energy content, the fuel mass will scale as:

$$\frac{m_M}{m_F} = \left(\frac{\Delta H_{c,F}}{\Delta H_{c,M}} \right) \left(\frac{l_M}{l_F} \right)^{5/2} \quad (1)$$

It should be kept in mind that different fuel types may affect the heat radiation due to the difference in soot yield and species production.

Table 1 A list of scaling correlations for the model tunnel.

Type of unit	Scaling	Equation number
Heat Release Rate (HRR) (kW)	$\dot{Q}_M / \dot{Q}_F = (l_M / l_F)^{5/2}$	(2)
Velocity (m/s)	$u_M / u_F = (l_M / l_F)^{1/2}$	(3)
Time (s)	$t_M / t_F = (l_M / l_F)^{1/2}$	(4)
Energy (kJ)	$E_M / E_F = (l_M / l_F)^3$	(5)
Temperature (K)	$T_F / T_M = 1$	(6)
Pressure (Pa)	$P_M / P_F = (l_F / l_M)^1$	(7)
heat flux (kW)	$\dot{q}_M'' / \dot{q}_F'' = (l_M / l_F)^{1/2}$	(8)
thermal inertia	$(k\rho c)_{s,M} / (k\rho c)_{s,F} \propto (l_M / l_F)^{3/2}$	(9)
thickness	$(k / \delta)_{s,M} / (k / \delta)_{s,F} \propto (l_M / l_F)^{1/2}$	(10)

Assuming $\Delta H_{c,M} = \Delta H_{c,F} \cdot l$ is the length scale. Index M is related to the model scale and index F to full scale.

3 Test set-up

A total of 7 tests were carried out in three rooms with a scaling ratio of 1:1, 1:2 and 1:3.5 respectively. Photos of fully developed fires in the different scales are shown in Figure 1.

(a) 1:1

(b) 1:2

(c) 1:3.5

Figure 1 Photos of the room fires in different scales. The heat release rates were 1.2 MW at full scale.

3.1 Room geometry and structure

All the three rooms were constructed using non-combustible 13 mm thick gypsum boards, mounted in wooden frames. The geometry of the room in full scale is 2.4 m (W) $\times$ 2.4 m (H) $\times$ 3.6 m (L) and the door is 2 m high and 0.8 m wide, see Figure 2. In other words, the full scale room has the same geometry as in room corner test.

Note that the size of the room is scaled geometrically according to the scaling ratio. Therefore, in the 1:2 scale room, the geometry was 1.2 m (W) $\times$ 1.2 m (H) $\times$ 1.8 m (L) and the door had dimensions of 0.4 m (W) $\times$ 1 m (H). In the 1:3.5 scale room, the geometry was 0.69 m (W) $\times$ 0.69 m (H) $\times$ 1.03 m (L) and the door had dimensions of 0.23 m (W) $\times$ 0.57 m (H), see Figure 2.

Figure 2 Front view of the rooms in different scales.

3.2 Insulating wall materials

The interior walls were covered with mineral wool (stone wool) in all tests. During the tests, the material was directly exposed to the hot gases. However, different types of mineral wools were used in different scales based on Eq. (4) and Eq. (5), see Table 2. In other words, more thermal resistant materials need to be used in a smaller scale. Note that the values of heat of capacity and conductivity in Table 2 are quite close to each other for these three mineral wools, however the density varies. For the scale ratio of 1 to 3.5, it is difficult to find a mineral wool with the thickness of 109 required by the scaling laws, therefore the thickness of 120 mm were used instead. However, the redundancy in thickness was compensated for when measuring the temperatures inside the wall as the measurement points for the thermocouples inside the wall were situated based on the thickness of 109 cm as required by the scaling laws, rather than 120 cm. Therefore it is expected that the influence of the difference in the thickness could be ignored.

Note that all the lining materials were attached to the 13 mm gypsum board mounted to the wooden frames. This means that the materials outside of the minerals wools were not scaled, and it may have some influence on the temperature distribution inside the wall

after the heat penetrates the mineral wools. However, the temperature gradient at this depth is much smaller compared to in mineral wools. Further, it can be expected that the heat has not penetrated the mineral wools before 20 min at full scale. Therefore, it is reasonable to consider the influence of the gypsum boards as being insignificant, at least at the beginning of the tests. This effect will be investigated based on tests data.

The inner thick materials are the most important. The temperature gradients inside can be expected to be much greater than the other layers. Further, the temperatures at the backside of the insulating materials in the tests are expected to be low before 20 min. Therefore, only the inner layer (fire resistant part) is scaled in the tests.

Further, note that the thermal conductivity could vary slightly with the temperature. However, it is expected that the conductivities for these three similar materials show similar dependences on the temperature and thus this effect is implicitly considered.

Table 2 The chosen materials based on the scaling Eqs. (9) and (10).

Scale ratio	Materials	k (W/m·K)	ρ (kg/m ³)	c (J/kg·K)	Thermal inertia $k\rho c$ (J ² /m ⁴ ·K ² ·s)		Thickness (mm)	
					Scale	actual	scale	actual
1:1	PAROC FPS 17	0.038	170	750	4845	4845	60	60
1:2	PAROC ROX COS 10	0.036	65	750	1713	1755	80	80
1:3.5	PAROC UNS	0.037	28	750	740	777	109	120

3.3 Fire sources

Propane burners were used in the tests due to its simplicity and convenience to control the heat release rate. Note that the flame length can be expressed in such a simple form [29]:

$$L_f = 0.235Q^{2/5} - 1.02D \quad (11)$$

Note that $Q \propto l^{5/2}$. We obtain:

$$D \propto l, \quad L_f \propto l \quad (12)$$

This suggests that the geometry of the fire source should scales as the length scale. Cubic fire sources were used in the tests. The side lengths of the fire sources are 30 cm, 15 cm and 8.6 cm in different scales, respectively.

The propane burner on the floor was either placed at the center of the room or at the room corner opposite to the door, see Figure 3. The top surfaces were 30 cm, 15 cm and 8.6 cm above the floor in full scale, medium scale and small scale, respectively.

3.4 Measurements and instrumentation

The instrumentation of the full scale room is shown in Figure 4. In other scales, the geometrical dimensions are directly scaled by the scaling ratio. The layout of

instrumentation in 1:2 and 1:3.5 model scales can be found in Appendix B. Here the measurements in full scale are described.

Various measurements were conducted during each test. Figure 3 shows the layout and identification of instruments in the full scale room.

One thermocouple tree was placed at the center of the room, i.e. T1 to T6. Another thermocouple tree was positioned at the centerline of the door, i.e. T7 to T11. Four thermocouples, i.e. T12 to T15 were placed 0.2 m below the ceiling, see Figure 3. All the thermocouples have a diameter of 0.25 mm.

Five plate thermometers[30,31], i.e. PT1 to PT5, were either placed on the walls or on the floor of the room center. The incident heat fluxes are calculated by the following equation:

$$[q''_{inc}]^{i+1} = \frac{\varepsilon_{PT} \sigma [T_{PT}^4]^i + (h_{c,PT} + K_{cond})([T_{PT}^4]^i - T_g) + C_{heat,\beta=1/3} \frac{[T_{PT}]^{i+1} - [T_{PT}]^i}{t^{i+1} - t^i}}{\varepsilon_{PT}} \quad (13)$$

where the conduction correction factor $K_{cond} = 8.43 \text{ W/m}^2 \cdot \text{K}$, and the lumped heat capacity coefficient $C_{heat,\beta=1/3} = 4202 \text{ J/m}^2 \cdot \text{K}$, the surface emissivity of Plate thermometer $\varepsilon_{PT} = 0.8$.

Hot gas flow velocity through the door, i.e. BP1, was measured using a bi-directional tube [32] placed beside the centerline of the door and 0.2 m below the upper edge of the door in full scale. The pressure difference was measured with a pressure transducer with a measuring range of $\pm 30 \text{ Pa}$.

Gas concentrations through the door, including CO_2 , CO and O_2 , i.e. G1 to G3, were sampled by one probe consisting of open copper tube beside the bi-directional probe, i.e. 0.2 m below the upper edge of the door.

Figure 3 Layout of instrumentation in the full scale tests (Dimensions in: mm).

The thermocouple series inside the walls were placed at 5 positions in the room, i.e. TS1 to TS5, see Figure 4. At each position, a thermocouple series consisting of 5 thermocouples was placed in different depths below the interior wall surface, see Figure 5. The five depths from the interior wall surfaces are 10%, 20%, 40%, 60% and 80% of the thickness (60, 80 and 109 mm in 3 scales respectively). Note that in different scales, the positions of corresponding thermocouples are different since the total thicknesses are different.

All the thermocouples, pressure transducers, gas analysers, flux meters, flow meter, and activation equipments were all connected to IMP 5000 KE Solotron loggers. The data were recorded by a laptop computer at a rate of approximately one scan per second.

Figure 4 Locations of thermocouples inside the walls (Dimensions in: mm).

4 Test procedure

In total, 7 tests were carried out in different scales, see Table 2. The main variable is the room geometry, lining materials, fire sources location and fire curves. During each test, the heat release rate was increased in a stepwise manner. In tests 1 to 3, the heat release rate was 100 kW for the early 10 min at full scale (0 min to 10 min), and 300 kW for another 10 min at full scale (10 min to 20 min), and then immediately turn the burner off. In tests 4 to 7, the heat release rate was 1.2 MW for further 7 min at full scale (20 min to 27 min) and then immediately turn the burner off. In other scales, the transition time was determined based on scaling law, i.e. Eq. (2) in Table 1. The gas burners were placed at the corner towards the door in tests 4 to 6 and placed at the center of the room in other tests. The burners were always set at floor level.

Table 3 HRR in different scales (Center and corner fires).

Test nr.	Fire location	Scaling ratio	1st stage		2nd stage		3rd stage	
			HRR kW	t ₁ Min	HRR kW	t ₂ min	HRR kW	t ₃ min
1	Center	1:1	100	10	300	20		
2	Center	1:2	18	7.1	53	14.1		
3	Center	1:3.5	4.4	5.3	13.1	10.7		
4	Corner	1:1	100	10	300	20	1200	27.0
5	Corner	1:2	18	7.1	53	14.1	212	19.1
6	Corner	1:3.5	4.4	5.3	13.1	10.7	52.4	14.4
7	Center	1:2	18	7.1	53	14.1	212	19.1

In each test, a cube of fibreboard was used as the ignition source. It was soaked in heptane and then placed beside the gas burner. This cube was ignited at 2 minutes from starting the logging system, and then moved after the burner was ignited. The ambient temperature was approximately 20 °C and the humidity is around 50 %.

Some photos of the tests are presented in Appendix C.

5 Summary of tests data

A short summary of tests data are presented here for the further analysis. These data include gas temperatures measured beneath the ceiling and at the thermocouple trees, heat fluxes, gas flow and gas concentration measured at the door and temperatures inside the walls. Note that in the tables presented in this chapter, the time in model scales have been scaled up for comparison, but not for the other parameters.

All the detailed tests data can be found in Appendix D.

5.1 Gas temperatures

The gas temperatures measured in all the tests are given in Table 4. T1 to T6 correspond to the thermocouple tree located at the center of the room, and T7 to T11 correspond to the thermocouple tree located along the centerline of the door. T12 to T15 are the gas temperatures measured 20 cm beneath the ceiling.

Note that in each test, the fire heat release rate can be divided into several stages. In tests 1 to 3, the heat release rate was 100 kW for the early 10 min in full scale (0 min to 10 min), and 300 kW for another 10 min in full scale (10 min to 20 min). In tests 4 to 7, the heat release rate was 1.2 MW for 7 more min in full scale (20 min to 27 min). Note that the gas temperatures increase with time at each stage, therefore, the maximum values generally correspond to the values at the last moment of the corresponding stage.

5.2 Incident heat flux

Incident heat fluxes measured by the plate thermometers are presented in Table 5. The maximum values generally correspond to the value at the last moment of a stage.

5.3 Gas flow and concentration

Gas flow velocity measured by bi-directional pressure tube and gas concentrations, including CO₂, CO and O₂, are presented in Table 5. The maximum values generally correspond to the value at the last moment of a stage.

5.4 Temperatures inside the wall

The internal wall temperatures measured using thermocouples embedded inside the insulation materials at different positions are presented in Table 6 and Table 7. Only the data corresponding to the end of each stage are listed here. Each thermocouple series (TS), includes 5 thermocouples, named 1 to 5 in the tables. Position 1 corresponds to the thermocouple closest to the interior wall surface, i.e. 10 % of thermal wall thickness below the interior wall surface, and position 5 is the deepest, i.e. 80 % of thermal wall thickness below the interior wall surface.

Table 4 Test results relevant to gas temperatures.

Test no.	Full-scale time t (min)	T1	T2	T3	T4	T5	T6	T7	T8	T9	T10	T11	T12	T13	T14	T15
		°C	°C	°C	°C	°C	°C	°C	°C	°C	°C	°C	°C	°C	°C	°C
1	10	265	304	430	625	700	87	196	109	52	30	28	210	216	215	209
	20	554	636	750	837	829	295	390	289	139	59	56	453	446	447	466
2	10	266	290	403	616	841	113	212	182	56	36	31	225	229	227	226
	20	504	512	637	695	811	213	386	343	143	76	54	466	421	435	462
3	10	208	213	227	273	327	71	187	115	38	24	22	202	203	198	197
	20	376	372	408	277	374	154	332	225	63	40	31	393	374	354	368
4	10	289	231	113	94	76	62	237	73	50	30	27	267	274	297	269
	20	558	462	328	274	233	189	422	184	130	65	59	498	521	672	523
	27	981	1034	977	843	801	649	957	730	529	427	376	1003	1131	1181	1021
5	10	312	236	110	97	80	68	245	84	57	34	24	290	307	335	300
	20	547	467	320	268	225	203	459	214	155	69	46	511	551	635	554
	27	923	925	885	827	686	701	873	708	518	310	179	914	1023	1049	956
6	10	306	219	134	113	91	83	219	71	37	25	24	265	293	336	283
	20	495	388	297	256	210	191	351	147	68	37	34	430	484	590	460
	27	892	830	739	661	616	559	728	519	239	134	120	804	901	978	834
7	10	274	370	498	594	807	102	214	173	54	37	33	229	227	225	226
	20	507	593	642	729	825	267	383	340	133	79	71	441	413	419	458
	27	896	888	866	784	902	623	983	861	598	463	397	962	857	857	933

Table 5 Test results relevant to heat flux, gas flow velocity and gas concentration.

Test no.	Full-scale time t (min)	PT1	PT2	PT3	PT4	PT5	u	CO ₂	CO	O ₂
		kW/m ²	kW/m ²	kW/m ²	kW/m ²	kW/m ²	m/s	%	%	%
1	10	3.7	2.3	3.4	4.7	3.7	3.1	1.05	0.0025	19.4
	20	16.2	12.9	17.6	18.3	11.1	5.1	2.45	0.022	17.1
2	10	4.2	2.0	3.8	5.7	3.7	1.6	1.13	0.0043	19.1
	20	13.0	9.8	14.4	15.8	11.0	2.5	2.20	0.049	17.2
3	10	3.4	1.8	2.9	4.0	2.9	0.9	-	0.0033	19.5
	20	9.2	5.6	6.9	9.7	7.0	1.4	-	0.016	17.8
4	10	4.7	14.2	5.2	3.6	2.7	2.3	1.04	0.0077	19.2
	20	20.2	87.6	32.0	18.5	11.4	4.6	2.12	0.037	17.6
	27	197.2	291.8	207.6	198.0	131.5	8.5	7.43	0.14	8.5
5	10	3.4	11.7	3.8	3.1	2.6	1.0	1.05	0.014	18.8
	20	17.1	67.5	23.0	16.6	12.4	2.0	2.37	0.14	16.6
	27	120.7	200.1	139.4	123.3	100.1	4.1	9.52	0.53	5.5
6	10	3.1	6.3	3.2	1.9	2.9	0.5	0.67	0.011	19.4
	20	10.4	32.2	13.1	8.7	9.9	1.1	1.42	0.044	18.0
	27	71.6	121.5	81.4	71.6	61.2	2.6	8.91	0.86	5.5
7	10	4.3	2.1	4.6	6.0	3.4	1.7	1.31	0.0036	19.0
	20	11.9	9.3	12.9	15.0	11.5	3.0	2.49	0.017	17.1
	27	107.8	84.9	106.2	110.6	89.3	5.1	-	0.39	3.1

“-” measurement error or out of range.

Table 6 Test results relevant to internal wall temperatures (TS1 to TS3).

Test no.	Full scale time t	TS1 (Positions 1 to 5)					TS2					TS3				
	min	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5
1	10	81	66	29	22	20	97	64	34	27	21	102	58	38	25	21
	20	235	200	83	41	26	271	191	101	71	30	294	185	119	56	30
2	10	79	55	33	24	22	95	69	36	26	22	116	89	43	26	23
	20	228	168	91	40	27	248	195	100	51	30	311	263	139	61	33
3	10	88	60	28	19	17	42	63	27	19	18	101	63	34	26	21
	20	242	187	94	37	23	122	169	81	40	29	244	180	106	77	55
4	10	95	76	30	21	19	302	193	83	53	23	123	67	41	24	20
	20	272	234	99	46	27	778	669	489	393	119	408	287	195	84	38
	27	883	813	460	279	182	–	–	–	–	–	–	–	–	–	–
5	10	100	67	26	21	19	472	410	214	56	31	76	47	25	20	20
	20	343	283	96	53	30	688	641	510	324	112	341	237	64	38	28
	27	803	698	375	225	58	–	–	–	–	–	–	–	–	–	–
6	10	123	84	37	23	21	–	381	194	70	38	152	95	50	42	30
	20	298	233	121	48	28	–	588	474	336	234	378	302	209	182	135
	27	706	630	448	192	61	–	754	633	508	391	760	681	571	532	450
7	10	79	54	31	22	21	92	68	34	24	21	115	81	42	25	22
	20	212	158	86	37	25	237	187	96	48	28	289	224	128	57	31
	27	675	556	356	121	40	683	597	378	174	60	781	673	500	238	80

“–” measurement failure.

Table 7 Test results relevant to internal wall temperatures (TS4 to TS5).

Test no.	Full scale time t (min)	TS4					TS5				
		1	2	3	4	5	1	2	3	4	5
1	10	121	69	33	24	21	54	45	26	20	19
	20	334	214	99	46	28	163	136	63	30	22
2	10	117	94	49	30	-	66	45	28	22	21
	20	314	270	154	72	-	177	123	63	32	24
3	10	76	50	29	19	18	47	32	20	18	17
	20	199	147	81	34	25	128	89	43	26	20
4	10	129	68	32	21	19	443	367	158	40	25
	20	342	216	100	43	27	733	672	466	215	65
	27	977	735	420	202	175	1040	963	771	469	196
5	10	104	71	31	22	20	507	427	222	58	28
	20	326	254	119	51	32	638	584	447	270	80
	27	765	541	257	74	45	859	787	634	463	203
6	10	90	55	32	22	21	423	314	109	36	22
	20	246	172	94	38	28	528	493	343	192	72
	27	644	534	360	127	59	720	685	581	452	268
7	10	121	91	53	28	24	62	42	26	21	20
	20	293	239	155	61	34	172	119	60	30	22
	27	697	531	356	105	44	733	653	486	171	42

“—” measurement failure.

6 Discussion of results

Firstly, the measured gas temperatures, gas flows, gas concentration, and incident heat fluxes are discussed. Then the internal wall temperatures and energy balance are analyzed. Further, the calculation of upper layer gas temperatures are investigated. Note that for comparison, the time in model scales has been scaled up in most of the figures.

6.1 Scaling of gas temperature

6.1.1 Ceiling gas temperature

Figure 5 shows the measured ceiling gas temperature as a function of the full-scale time for center fires. In other words, the time in medium scale and small scale has been scaled up for easy comparison.

Figure 5 Ceiling gas temperatures for the center fires.

Clearly, it is shown in Figure 5 that the ceiling gas temperatures correlate well in different scales of center fires, especially in medium scale tests where a perfect match can be found.

Note that in small scales, the ceiling gas temperatures are slightly lower, especially at the 2nd stage when the gas temperatures are around 400 °C. Further, the ceiling gas temperatures measured by the four ceiling thermocouples in full scale are closely of the same values, however, in model scales, small differences can be found for different thermocouples. In general, the ceiling gas temperatures are scaled very well in this series of tests.

Figure 6 shows the measured ceiling gas temperatures for corner fires. Note that for comparison, the time in model scales has also been scaled up.

Clearly, it is shown in Figure 6 that the ceiling gas temperatures in different scales correlate very well before 20 min, and after 20 min, the ceiling gas temperatures are highest in full scale, and slightly lower in medium scale and lowest in small scale. The maximum ceiling gas temperatures are 1181 °C in full scale, 1049 °C in medium scale and 978 °C in small scale. Note that these temperatures correspond to flame temperatures in the continuous combustion region and thus they partly indicate the combustion intensity. Another reason for the variations could be that the sooty smoke blocks the heat loss in larger scales.

Note that before 10 min, all the temperatures measured at T12 to T15 are almost the same, however, after 10 min, the temperature at T14 is much higher compared to other positions and the temperatures measured by the other thermocouples are very similar. This is due to that after 10 min the continuous flame region reached the ceiling and T14 directly measured the flame temperature. Also, note that the highest temperatures were registered at T14, then T13, T15 and T12. This is mainly due to that in the fully developed fires, more heat was lost at T15 and T12 which were close to the openings, and T12 is the farthest thermocouple from the fire source. This indicates that after the flame impinges the ceiling, the ceiling gas temperatures are not uniform, but depend on its location relative to the fire source and the opening.

In summary, for center fires, the upper layer gas temperatures correlate well in different scales, especially in medium scale tests where a perfect match can be found. In small scales, the ceiling gas temperatures are slightly lower, especially at the 2nd stage. For corner fires, the ceiling gas temperatures are scaled very well before 20 min but are slightly underestimated in model scales after 20 min at a heat release rate of 1.2 MW at full scale.

(a) Full scale

(b) Medium scale

(c) Small scale

Figure 6 Ceiling gas temperature in small scale (corner fire).

6.1.2 Thermocouple tree at the room center

Here the gas temperatures measured by the thermocouple trees placed in the centers of the rooms in different scales, i.e. T1 to T6, are compared.

Figure 7 shows the vertical temperature distribution in the center of the room for center fires. Note that for easy comparison, the time in model scales has been scaled up. Since the fire was placed in the center and on the floor, it was strongly affected by the air flows through the doors induced by the hot smoke flows. The scatter has been eased by averaging the measured values, i.e. 10 s average values in full scale, 7 s average in medium scale and 5 s in small scale according to the scaling of time. It can be seen that the measured temperatures at T4 and T5 are always instable and differ significantly in different scales. This is mainly due to the strong influence of the air flows through the openings. However, it can also be seen that there is good agreement in different scales for the measured temperatures at the other positions.

(a) Full scale

(b) Medium scale

(c) Small scale

Figure 7 Temperature distribution in center of the room for center fires.

Figure 8 shows the vertical temperature distribution in the center of the room for corner fires. Note that for comparison, the time in model scales has been scaled up. Clearly, the measured temperatures in different scales show high similarity. Before 10 min, there was a strong stratification and the smoke layer height lay between T2 and T3. After 10 min, the smoke layer slightly descended. After 20 min the floor temperature was over 600 °C and there was no clear stratification.

(a) Full scale

(b) Medium scale

(c) Small scale

Figure 8 Temperature distribution in center of the room for corner fires.

In summary, the vertical temperature distribution at the center of the room is scaled well for both corner fires and center fires, except some data points in center fires scattered due to the strong influence of the air flows through the openings on the flame.

6.1.3 Thermocouple tree at door

Figure 9 shows the vertical temperature distribution at the doors for center fires in different scales. For comparison, the time in model scales has been scaled up. The smoke flow height can be estimated from the temperature distribution. It can be seen that the temperatures are scaled well in medium scale except at T8. This could be due to that the

temperature gradient is very large and thus the gas temperature is sensitive to even slight deviation of the position of the thermocouple.

It can also be seen that in small scale the smoke layer height are slightly higher. This could be partly due to the larger roughness of the door and all the surfaces. Also, note that the temperature is lower in model scales. One reason could be that more heat is lost through the opening and the radiation is overestimated at this location in model scales. Note that for the sooty smoke inside the carriage and far away from the openings, the openings could be considered as invisible, and thus the gas temperatures can be scaled well. However, for the gases close to the opening, the radiation heat loss is significant

(a) Full scale

(b) Medium scale

(c) Small scale

Figure 9 Temperature distribution at the doors for center fires.

Figure 10 shows the vertical temperature distribution at the doors for corner fires in different scales. For comparison, the time in model scales has been scaled up. The profiles of the temperatures have high similarities, especially between medium scale and full scale before 20 min. However, after 20 min, the gas temperatures at the top and at the bottom in model scales are lower than in full scale, especially in small scale.

(a) Full scale

(b) Medium scale

(c) *Small scale*

Figure 10 Temperature distribution at the doors for corner fires.

In short, the vertical temperature distribution at the door is relatively scaled well before 20 min, however, after 20 min, the gas temperatures at the top and at the bottom in model scales are lower in model scales compared to in full scale. This should be mainly explained by the relatively higher heat loss through the openings in model scales.

6.2 Scaling of gas flow and concentration

6.2.1 Gas flow

Figure 11 shows the gas velocities through the doors in different scales of center fires. Note that for easy comparison, the time in model scales has been scaled up. Clearly, it shows very good correlation in different scales. Note that after 10 min, the gas velocity is slightly lower in small scales. This could be explained by the finding from the temperature distribution that the smoke layer height is slightly higher in small scale. An interesting phenomenon is the high similarity in the gas flow after 20 min when there was no fire source. Therefore, this process was a cooling process by natural ventilation, contrary to the heating process at the two early stages.

Figure 11 Full-scale gas velocity through the doors for center fires (data scaled up).

Figure 12 shows the gas velocities through the doors in different scales of corner fires. Also, there is good correlation between the scales. In small scale, the gas velocity is also slightly lower during 10 min to 20 min, and the same reason can be used to explain this.

In summary, the gas velocities are scaled very well for both center fires and corner fires.

Figure 12 Full-scale gas velocity through the doors for corner fires (data scaled up).

6.2.2 Gas concentration

Figure 13 shows the measured oxygen concentration at the doors for center fires. Note that for easy comparison, the time in model scales has been scaled up. Clearly it shows that the medium scale perfectly matches the full scale. The oxygen concentration in small scale is slightly higher and there appears to be a delay. The reasons could be that the response time to correct the fuel flow has been scaled up, and the smoke layer height could be slightly higher in small scale.

Figure 13 O_2 concentration at the doors for center fires.

Figure 14 shows the measured oxygen concentration at the doors for corner fires. Note that for easy comparison, the time in model scales has been scaled up. Before 20 min, the same phenomenon can be observed as for center fires. After 20 min, the oxygen concentration in model scales are lower. There could be three reasons for the underestimation for large fires. Firstly, the smoke layer at the door in model scales could be slightly thinner according to the vertical temperature distributions at the doors,

especially in small scale. Secondly, the binders in the mineral wools may release some heat. Thirdly, slightly more fuel could be burnt in the room in model scales compared to the full scale due to the scaled-up experimental error.

Figure 14 O_2 concentration at the doors for corner fires.

Figure 15 shows the measured CO_2 concentration at the doors for center fires. Note that for easy comparison, the time in model scales has been scaled up. Clearly it shows that the medium scale perfectly matches the full scale. The data of small scale are not presented due to the failure of the measurement.

Figure 15 CO_2 concentration at door for center fires.

Figure 16 shows the measured CO_2 concentration at the doors for corner fires. The same phenomenon as observed in O_2 concentration can be found here. The same reasons can explain this. Note that the CO_2 concentration increases very slowly with time at the early stage. The reason is that the smoke layers for the corner fires are slightly thinner compared to the center fires and the smoke layer height decreases gradually as the ceiling gas temperature increases with time due to the decreasing heat loss.

Figure 16 CO_2 concentration at door for corner fires.

In summary, the gas concentrations are scaled well for both center fires and corner fires. However, for large fires, the oxygen concentrations are slightly overestimated and the CO_2 concentrations are overestimated. The reason could be mainly due to the smoke layer height at the door in model scales are slightly overestimated for large fires.

6.3 Scaling of incident heat flux

6.3.1 Center fires

Figure 17 shows the measured incident heat fluxes in different scales of center fires. Note that for comparison, both the time and the incident heat fluxes in model scales has been scaled up. It shows that there is very good agreement in different scales.

Before 10 min, the radiation is slightly higher in model scales. The main reason is that during this period the main radiation comes from the flame. Note that the flame normally corresponds to a large emissivity, which results in slightly higher radiation in model scales after the data are scaled up.

(a) Full scale

(b) Model scale

(c) Small scale

Figure 17 Incident heat fluxes in full scale room (center fires, data scaled up).

6.3.2 Corner fire

Figure 18 shows the measured incident heat fluxes in different scales of corner fires. Note that for comparison, both the time and the incident heat fluxes in model scales has been scaled up. It shows that the medium scale matches the full scale very well at each stage. The heat fluxes are scaled relatively well in small scale before 20 min, except at PT2 where the heat flux are underestimated after 10 min. After 20 min, the heat fluxes in model scale are generally lower compared to full scale. Part of the reason could be that the convection heat fluxes are overestimated in model scales, especially in small scale with rough walls. Therefore, the overall heat flux imposed on the wall surfaces should be scaled better in small scale than what is shown here.

(a) Full scale

(b) Medium scale

(c) Small scale

Figure 18 Incident heat fluxes in full scale room (corner fire, data scaled up).

Figure 19 and Figure 20 show the comparisons between incident heat fluxes measured in full scale and model scales for center fires and corner fires respectively. Note that all the data have been scaled up to the full scale. Clearly, it can be seen that the incident heat fluxes scales well, especially in medium scales. However, it can also be seen that for small fires with a full-scale heat release rate of 100 kW, the incident heat fluxes are slightly overestimated in model scales, and for large fires with a full-scale heat release rate of 1.2 MW, the incident heat fluxes in small scale are slightly underestimated.

Figure 19 Comparison of incident heat fluxes for center fires (data scaled up).

Figure 20 Comparison of incident heat fluxes for corner fires (data scaled up).

In summary, the incident heat fluxes are scaled very well, especially in medium scale. The incident heat fluxes are only slightly overestimated in model scales for a full-scale heat release rate of 100 kW, and are slightly underestimated in small scale for larger fires.

6.4 Scaling of internal wall temperature

6.4.1 Center fire

The measured internal wall temperatures at TS1 in different scales for center fires are shown in Figure 21, Figure 22 and Figure 23. Note that for comparison, the time in model scales has been scaled up. 10 % indicates the thermocouple was placed at 10 % of the thickness below the wall surface, and et cetera. Clearly, a perfect correlation can be found

between the different scales, except that the data measured at 20 % of the thickness below the surface are obviously higher in full scale than model scales, which should be due to that this thermocouple was wrongly positioned.

Figure 21 Internal wall temperatures at TS1 in full scale (center fire).

Figure 22 Internal wall temperatures at TS1 in medium scale (center fire).

Figure 23 Internal wall temperatures at TS1 in small scale (center fire).

For clearer comparison, in the following we make a comparison of the internal wall temperatures measured at full-scale time 5 min, 10 min, 15 min, 20 min, 23.5 min and 27 min for center fires. In other words, the time has also been scaled up while picking up the data in model scale tests.

Figure 24 shows the comparison of the internal wall temperatures in full scale and medium scales of the center fires. Clearly, it shows that there is very good correlation between the full scale and the medium scale. Most data lie closely beside the equal line.

Further, it can be seen that the data scatter slightly. This could be partly due to the error induced by the placement of the thermocouples inside the walls in the tests. Note that the thermocouples are embedded inside the insulating materials and fixed to the exterior frame. Further, the interval distance between the thermocouples are generally very small, that is, the minimum interval is 6 mm in full scale, 8 mm in medium scale and 11 mm in small scale. Therefore, it is difficult to place them perfectly into the right positions, especially in full scale. This also indicates that larger errors could be induced in full scales than in model scales.

Figure 24 Internal wall temperatures in full scale vs. medium scale (center fires).

Figure 25 shows the comparison of the internal wall temperature in full scale and small scales of the center fires. Clearly, it shows that although the internal wall temperatures in the small scale is slightly lower at some positions, very good correlation can be found between the full scale and the small scale.

Figure 25 Internal wall temperatures in full scale vs. small scale (center fires).

6.4.2 Corner fires

6.4.2.1 Small fires

In this section, the fires before 20 min at full scale are focused on at first. The corresponding heat release rates are lower than 300 kW at full scale and these fires called as small fires.

Figure 26 shows the comparison of the internal wall temperature in full scale and medium scale of the corner fires. Note that only tests data before 20 min are used for comparison. Clearly, it shows that there is very good correlation between the full scale and the medium scale although some data scatters between 100 °C and 300 °C.

Figure 26 Internal wall temperatures in full scale vs. medium scale (corner fires, within 20min).

Figure 27 shows the comparison of the internal wall temperature in full scale and small scale of the corner fires within 20min. Clearly, it shows that there is good correlation between the full scale and the medium scale although the internal wall temperatures measured in small scale are slightly lower after 600 °C.

Figure 27 Internal wall temperatures in full scale vs. small scale (small corner fires).

6.4.2.2 Large fires

The fires with a full-scale heat release rate of 1.2 MW are considered here, which are called large fires in the report.

Note that in full scale corner fire tests with a heat release rate of 1.2 MW, the mineral wool panels in the vicinity of the fire source, started to melt on the surface at around 23 min and some portion of the materials fell down, while they were exposed to gases with high temperatures, see Figure 28. The thicknesses of the insulating materials decreased which resulting in the change of the positions of the thermocouples relative to the wall surfaces. Therefore, the temperature increases significantly after 23 min, which can be seen from Figure D8. The two location significantly affected by this in full scale are TS5 and TS3. These two positions corresponded to high flame temperatures above approximately 1100 °C. Further, in the medium scale test, the panel above the fire source was slightly detached after the test and the measurements at these two positions, i.e. TS5 and TS3, were also affected. Due to these two reasons, the measured data at these two positions for large fires are not used for comparison in the following.

Figure 28 A photo after the full scale test 4.

Figure 29 shows the comparison of the full scale internal wall temperatures to the medium scale values. Figure 30 shows the comparison of the full scale internal wall

temperatures to the small scale values. Clearly, it shows in Figure 29 that there is good agreement between the full scale and medium scale for internal wall temperatures lower than 800 °C. Further, it shows in Figure 30 that there is good agreement between the full scale and the small scale for internal wall temperatures lower than 600 °C.

However, it can also be noticed that the internal wall temperatures are underestimated for temperatures over 800 °C in medium scale and over 600 °C in small scale. There are four reasons for this.

Firstly, as discussed previously, the wall materials in the vicinity of the fire source in full scale were melted on the surface and some portion of the materials fell down while exposed to such high temperatures. This also happened in other scales. From the observation of the wall panels after the tests the thickness of the wall panels changed most significantly in full scale, less in medium scale, and very slightly in small scale. Therefore, the internal wall temperatures measured by the thermocouples in full scale are much greater than the values measured in model scales where no such significant change in the thickness occurred.

Secondly, the upper layer gas temperatures in full scale are slightly higher than in medium scale, and the upper layer gas temperatures in small scale are the lowest.

Thirdly, note that the interval distance between the thermocouples are generally very small, that is, the minimum interval is 6 mm in full scale, 8 mm in medium scale and 11 mm in small scale. This indicates that the internal wall temperatures are most sensitive to the changes in the thickness in full scale, less sensitive in medium scale and least in small scale. This could result in the highest internal wall temperatures measured in full scale, higher in medium scale and the lowest in small scale. Further, note that the changes of the positions of the thermocouples have the strongest influence on the thermocouple close to the surface, and much less for the thermocouples deep in the walls. Therefore, the discrepancy between different scales should be much smaller in a deep thermocouple compared to one close to the surface as explained, that is, a much better agreement can be found for low temperatures. This can explain why still good agreement can be found between the full scale and model scales for large fires with gas temperatures below 800 °C in medium scale and below 600 °C in small scale.

Fourthly, during the large fires, the heat had penetrated the physical thickness of the mineral wools, reached the exterior structure consisting of 13 mm gypsum boards and the wooden frames. Note that the structures were not scaled. This indicates too much heat could be lost in model scales through the structure after the heat penetrated the insulating materials, which in turn results in a lower internal wall temperatures in model scales.

Despite all these factors which tend to reduce the internal wall temperatures measured in model scales, the scaling of internal wall temperatures in large fires works relatively well.

Figure 29 *Internal wall temperatures in full scale vs. medium scale (corner fire, including large fires).*

Figure 30 *Internal wall temperatures in full scale vs. small scale (corner fire, including large fires).*

In summary, the scaling of internal wall temperatures for both small center fires and corner fires works very well. For large corner fires, the scaling of internal wall temperatures works relatively well, and slightly better in medium scale compared to small scale.

6.5 Calculation of ceiling gas temperatures

In post-flashover enclosure fires, the mass flow rate can be easily obtained due to the insensitivity of the mass flow through the openings, and in turn the gas temperature could be estimated based on the energy equation, assuming the heat released inside the compartment is known.

In pre-flashover enclosure fires, the mass flow rate through the openings cannot be simply obtained, instead, a numerical solution is required for a quasi-steady fire. This in turn can be used to estimate the gas temperature based on the two-zone model assumption, which should be appropriate for normal enclosure fires. However, simple explicit empirical equation is much easier and more widely used in engineering applications, such as the well-known McCaffrey, Quintiere and Harkleroad (MQH) equation [33], which is expressed as:

$$\Delta T_g = 6.85 \left(\frac{Q^2}{A_o \sqrt{H_o} h_k A_T} \right)^{1/3} \quad (14)$$

Before thermal penetration, the conductive heat coefficient:

$$h_k = \left(\frac{k_s \rho_s c_s}{t} \right)^{1/2} \quad (15)$$

and after thermal penetration, the conductive heat coefficient:

$$h_k = \frac{k_s}{\delta_s} \quad (16)$$

The penetration time can be esitamted as:

$$t_p = \frac{\delta_s^2}{4a} \quad (17)$$

This gas temperature equation is obtained from dimensional analysis and keeps a form similar to the theoretical implicit equation for the gas temperature. In any case, this equation is empirical.

Delichatsios et al. [34] also proposed a correlation for gas temperatures in enclosure fires based on the adiabatic gas temperature, which is mainly dependent on the heat release rate. This equation has a form similar to the MQH equation, but the comparison made by Delichatsios et al. [34] showed that the MQH overestimates the gas temperatures. However, they assumed the smoke layer height as 50 % of the smoke layer height and the calculated adiabatic gas temperature could be as high as 2000 K. Further, the proposed equation is implicit and needs numerical solution and therefore not discussed further here.

Mowrer and Williamsson [35] investigated the gas temperature in fires that were flush against walls or were positioned in a corner. They found the MQH equation still works by by using a correction factor. they recommended that the results from the MQH equation are multiplied by 1.3 for fires flush against walls and by 1.7 for fires in corners.

In the following, the upper layer gas temperature for small fires, i.e. pre-flashover fires, are investigated. The data corresponding to a full-scale heat release rate of 1.2 MW, which are related to post-flashover fires, are not taken into account.

6.5.1 Center fires

Note that the gas temperatures below the ceiling are dependent on the location relative to the fire source. The gas temperature above the fire is normally higher than at other

positions. At first, we focus on the maximum gas temperature measured below the ceiling, i.e. the measured maximum value within thermocouples T1, and T12 to T15. Figure 31 shows the measured maximum ceiling excess gas temperatures for center fires. The full-scale data of the maximum excess gas temperatures for center fires suggest the best fit line:

$$\Delta T_g = 6.8 \left(\frac{Q^2}{A_o \sqrt{H_o} h_k A_T} \right)^{1/3} \quad (18)$$

It is shown in Figure 31 that the data for maximum ceiling excess gas temperature for center fires comply well with the equation which is almost the same as the MQH equation. Note that the maximum excess temperatures were mostly measured right above the fire, i.e. at T1.

Figure 31 Measured maximum ceiling excess temperature (center fires)

Figure 32 shows the average ceiling excess gas temperature within T12 to T15 for center fires. The average values are representative of the excess temperature of the whole upper layer, and more meaningful in practice compared to the maximum ceiling excess gas temperature discussed above. The full-scale data of the average excess gas temperatures for center fires suggest the best fit line:

$$\Delta T_g = 5.7 \left(\frac{Q^2}{A_o \sqrt{H_o} h_k A_T} \right)^{1/3} \quad (19)$$

Comparing the equations for the maximum temperatures and for the average temperatures for center fires show that the average excess gas temperatures are approximately 84 % of the maximum values for center fires. Further, the MQH equation significantly overestimates the average ceiling excess gas temperatures for center fires.

Figure 32 Measured average ceiling excess temperature (center fires)

6.5.2 Corner fires

Figure 33 shows the maximum ceiling excess gas temperature for corner fires. The full-scale data of the maximum excess gas temperatures for corner fires suggest the best fit line:

$$\Delta T_g = 8.5 \left(\frac{Q^2}{A_o \sqrt{H_o h_k A_T}} \right)^{1/3} \quad (20)$$

Comparing the above equation with the MQH equation shows that the maximum ceiling excess gas temperature for corner fires are much higher than those estimated using the MQH equation. Note that the maximum excess temperatures were all measured in the vicinity of the fire, i.e. at T14.

Figure 33 Maximum ceiling excess gas temperature (corner fires)

Figure 34 shows the average ceiling excess gas temperature for center fires. The full-scale data of the average excess gas temperatures for corner fires suggest the best fit line:

$$\Delta T_g = 7.15 \left(\frac{Q^2}{A_o \sqrt{H_o} h_k A_T} \right)^{1/3} \quad (21)$$

Clearly, the average excess gas temperatures are also approximately 84 % of the maximum values for center fires.

Figure 34 Measured average ceiling excess gas temperature (corner fires)

These results show that both maximum ceiling excess gas temperatures and average ceiling excess gas temperatures for corner fires are higher than those for center fires, and the ratio between corner fires and center fires is 1.25, for both the maximum and the average ceiling excess gas temperatures. Compared to the coefficient of 1.7 for corner fires proposed by Mowrer and Williamsson [35], the correction factor of 1.25 obtained here is much lower. The correction factor for a fire source detached to the wall is supposed to be lower than this value. This may indicate that the placement of the fire source detached to the corner or the wall does not have strong influence on the gas temperatures. In these cases, the flame tilts towards the opposite side of the wall or corner and the total entrained flow rate by the flame and smoke plume is only slightly lower compared to center fires.

Further, the maximum temperature exists in the vicinity of the fire. The maximum excess gas temperatures are approximately 19 % higher than the average values for both center fires and corner fires.

For center fires, the MQH equation can be used to estimate the maximum ceiling gas temperatures, but it significantly overestimates the average gas temperatures. For corner fires, the MQH equation can be used to approximately estimate the average ceiling gas temperatures, but it significantly underestimates the maximum ceiling gas temperatures.

Based on the above analysis we may propose the following equation for calculation of upper layer excess gas temperatures in enclosure fires:

$$\Delta T_g = 5.7 C_1 C_2 \left(\frac{Q^2}{A_o \sqrt{H_o} h_k A_T} \right)^{1/3} \quad (22)$$

where

$$C_1 = \begin{cases} 1 & \text{for center fires} \\ 1.25 & \text{for corner fires} \end{cases}$$

and

$$C_2 = \begin{cases} 1 & \text{for average temperatures} \\ 1.20 & \text{for maximum temperatures} \end{cases}$$

Note that the center fires discussed here could be equivalent to any fire in other location in the room except the locations detached to a wall or a corner or placed right in front of one opening. The equation is expected to be valid for gas temperatures below 700 °C or slightly higher, but not over 1000 °C.

Now we can make an estimation of the transient temperatures in full scale. Figure 35 and Figure 36 show comparisons of measured average temperatures to the calculated values using the proposed equation, i.e. Eq. (22), for test 1 and test 4 respectively. Clearly, the proposed equation correlates well with the tests data. However, the gas temperatures are overestimated during the first 1 min and slightly underestimated during the first 10 min.

Figure 35 Measured average ceiling excess temperature (center fires test 1)

Figure 36 Measured average ceiling excess temperature (corner fires test 4)

It should be kept in mind that the mineral wools are used in the tests. Therefore, the proposed equation could only be used for estimation of gas temperatures in compartments covered by similar insulating materials.

6.6 Applications in lightweight constructions

The above analysis has proved that the internal temperatures of the insulating materials can be scaled well except for the fact that the insulating materials start to melt and fall down while exposed to high temperatures over 1000 °C. For realistic lightweight constructions there are always some panels placed above the insulating materials, which can prevent the collapse of the insulating materials even when it partly performs as thermoplastic materials.

The scaling method proposed in this project can be used to scale the scenarios of lightweight constructions even for large fires if the exterior surfaces of the insulating materials are appropriately covered by a panel.

7 Conclusions

The scaling of enclosure fires including internal wall temperatures and heat fluxes is thoroughly analyzed and a method for scaling of internal wall temperatures is proposed. Two series of room fire tests were carried out in three different scales, i.e. full scale (1:2), medium scale (1:2) and small scale (1:3.5), according to the theory. The fire sources were either placed at the center or in the corner of the enclosures. The main focus of these tests was scaling of internal wall temperatures inside the insulating materials. Based on the tests data, the scaling of internal wall temperatures, incident heat fluxes, gas temperatures, gas velocities and gas concentrations is analyzed. Further, the calculation of upper layer gas temperatures in the compartment fires is investigated.

The scaling of internal wall temperatures for both small center fires and corner fires works very well. For large corner fires, the internal wall temperatures are scaled relatively well, and slightly better in medium scale compared to small scale.

The incident heat fluxes are scaled very well, especially in medium scale. The incident heat fluxes are only slightly overestimated in model scales for a full-scale heat release rate of 100 kW, and are slightly underestimated in small scale for larger fires.

For center fires, the measured upper layer gas temperatures correlate well in different scales, especially in medium scale tests where a perfect match can be found. In small scales, the upper layer gas temperatures are slightly lower, especially at the 2nd stage. For corner fires, the ceiling gas temperatures are scaled very well before 20 min but are slightly underestimated in model scales after 20 min at a heat release rate of 1.2 MW at full scale. The vertical temperature distribution at the center of the room is scaled well for both corner fires and center fires, with the exception that some data points in center fires scattered due to the strong influence of the air flow through the openings on the flame. The vertical temperature distribution at the door is relatively scaled well before 20 min, however, after 20 min, the gas temperatures at the top and at the bottom in model scales are lower in model scales compared to full scale mainly due to relatively higher heat loss through the openings in model scales.

The gas velocities are scaled very well for both center fires and corner fires. The gas concentrations are scaled well for both center fires and corner fires. However, for large fires, the oxygen concentrations are slightly overestimated and the CO₂ concentrations are overestimated. The reason could be mainly due to the smoke layer height at the door in model scales are slightly overestimated for large fires.

Both maximum ceiling excess gas temperatures and average ceiling excess gas temperatures for corner fires are higher than those for center fires, and the ratio is 1.25 for both the maximum and the average ceiling excess gas temperatures. The maximum temperature normally exists right above the fire. The maximum excess gas temperatures are approximately 19 % higher than the average values for both center fires and corner fires. For center fires, the MQH equation can be used to estimate the maximum ceiling gas temperatures, but it significantly overestimates the average gas temperatures. For corner fires, the MQH equation can be used to approximately estimate the average ceiling gas temperatures, but it significantly underestimates the maximum ceiling gas temperatures. A correlation has been proposed to estimate the upper layer gas temperatures in compartments covered by similar insulating materials for both center fires and corner fires. The equation is expected to be valid for gas temperatures below 700 °C or slightly higher, but not over 1000 °C. Note that the proposed equation could only be used for estimation of upper layer gas temperatures in compartments covered by similar insulating materials.

This report should raise our concerns on the importance of wall materials in the scaling of enclosure fires. The scaling of the thermal properties of wall materials should be taken into account in model scale fire tests in the future.

8 References

1. Heskestad G (1975) Physical Modeling of Fire. *Journal of Fire & Flammability* 6:p. 253 - 273
2. Quintiere JG (1989) Scaling Applications in Fire Research. *Fire Safety Journal* 15:3-29
3. Perricone J, Wang M, Quintiere J (2007) Scale Modeling of the Transient Thermal Response of Insulated Structural Frames Exposed to Fire. *Fire Technology* 44 (2):113-136. doi:10.1007/s10694-007-0029-1
4. Croce PA, Xin Y (2005) Scale modeling of quasi-steady wood crib fires in enclosures. *Fire Safety Journal* Vol. 40:245-266
5. Heskestad G (2002) Scaling the interaction of water sprays and flames. *Fire Safety Journal* 37:535-548
6. Heskestad G (2003) Extinction of gas and liquid pool fires with water spray. *Fire Safety Journal* 38:301-317
7. Quintiere J.G., Su G.Y., N. S (2007) Physical scaling for water mist fire suppression – a design application. *International Journal on Engineering Performance-Based Fire Codes* 9 (2):87-108
8. Yu H.Z., Zhou X.Y., Ditch B.D. Experimental validation of Froude-modeling-based physical scaling of water mist cooling of enclosure fires. In: 9th International Symposium on Fire Safety Science (Poster), Karlsruhe, Germany, 21-26 September 2008. IAFSS, pp 553-564
9. Jayaweera T.M., Yu H.Z. (2008) Scaling of fire cooling by water mist under low drop Reynolds number conditions. *Fire Safety Journal* 43:7
10. Yu H.Z. Physical scaling of water mist suppression of pool fires in enclosures. In, College Park, MD, 2011. 10th International Symposium on Fire Safety Science.
11. Ingason H, Li YZ (2010) Model scale tunnel fire tests with longitudinal ventilation. *Fire Safety Journal* 45:371-384
12. Ingason H, Li YZ (2011) Model scale tunnel fire tests with point extraction ventilation. *Journal of Fire Protection Engineering* 21 (1):5-36
13. Ingason H (2007) Model Scale Railcar Fire Tests. *Fire Safety Journal* 42 (4):271-282
14. Li YZ, Lei B, Ingason H (2010) Study of critical velocity and backlayering length in longitudinally ventilated tunnel fires. *Fire Safety Journal* 45:361 - 370
15. Li YZ, Lei B, Ingason H (2011) The maximum temperature of buoyancy-driven smoke flow beneath the ceiling in tunnel fires. *Fire Safety Journal* 46 (4):204-210
16. Li YZ, Ingason H (2012) The maximum ceiling gas temperature in a large tunnel fire. *Fire Safety Journal* 48:38-48
17. Li YZ, Ingason H (2012) Position of Maximum Ceiling Temperature in a Tunnel Fire. *Fire Technology*. doi:10.1007/s10694-012-0309-2
18. Li YZ, Ingason H (2010) Maximum Temperature beneath Ceiling in a Tunnel Fire. SP Report 2010:51, SP Technical Research Institute of Sweden,
19. Li YZ, Lei B, Ingason H (2011) Theoretical and Experimental Study of Critical Velocity for Smoke Control in a Tunnel Cross-Passage. *Fire Technology*. doi:10.1007/s10694-010-0170-0
20. Li YZ, Lei B, Ingason H (2012) Scale modeling and numerical simulation of smoke control for rescue stations in long railway tunnels. *Journal of Fire Protection Engineering* 22 (2):101-131. doi:10.1177/1042391512445409
21. Li YZ, Ingason H The Fire Growth Rate in a Ventilated Tunnel Fire. In: 10th International Symposium on Fire Safety Science (IAFSS), , Maryland, USA, 2011. pp 347-358. doi:10.3801/iafss.fss.10-347
22. Lönnermark A, Lindström J, Li YZ (2011) Model-scale metro car fire tests. SP Technical research Institute of Sweden, Borås, Sweden
23. Lönnermark A, Lindström J, Li YZ, Claesson A, Kumm M, Ingason H (2012) Full-scale fire tests with a commuter train in a tunnel. SP Technical Research Institute of Sweden, Borås, Sweden

24. Ingason H (2008) Model scale tunnel tests with water spray. *Fire Safety Journal* 43 (7):pp 512-528
25. Li YZ, Ingason H (2011) Model scale tunnel fire tests – Automatic sprinklers SP Report 2011:31
26. Hertzberg T, Arvidson M, Axelsson J Large-scale Fire Tests in a Passenger Cabin. In: *Fire and Materials 2009 - Proceedings of the 11th International Conference and Exhibition, Fisherman's Wharf, San Francisco, USA, 26-28 January 2009*. Interscience Communications, pp 527-533
27. Hertzberg T (2009) LASS, Lightweight Construction Applications at Sea. SP Report 2009:13
28. Li YZ, Ingason H, Lönnermark A (2013) Correlations in different scales of metro carriage fire tests. SP Report 2013:13
29. Heskestad G (1983) Luminous Heights of Turbulent Diffusion Flames. *Fire Safety Journal* 5:103-108
30. Ingason H, Wickström U (2007) Measuring incident radiant heat flux using the plate thermometer *Fire Safety Journal* Vol. 42 (2):161-166
31. Häggkvist A (2009) The Plate Thermometer as a Mean of Calculating Incident Heat Radiation - A practical and theoretical study. Luleå University of Technology,
32. McCaffrey BJ, Heskestad G (1976) Brief Communications: A Robust Bidirectional Low-Velocity Probe for Flame and Fire Application. *Combustion and Flame* 26:125-127
33. McCaffrey B.J., Quintiere J.G., Harkleroad M.F. (1981) Estimating room fire temperatures and the likelihood of flashover using fire test data correlations. *Fire Technology* 17 (2):98-119
34. Delichatsios M, Lee YP, Tofilo P (1981) A new correlation for gas temperature inside a burning enclosure. *Fire Safety Journal* 44 (8):1003-1009
35. Mowrer F, Williamson R (1987) Estimating room temperatures from fires along walls and in corners. *Fire Technology* 23 (2):133-145. doi:10.1007/bf01040428
36. Quintiere JG Fire behaviour in building compartments. In: *Proceedings of the Combustion Institutes*, 2002. pp 181-193

Appendix A - Scaling of internal wall temperatures

The method of scaling used in the tests is the most widely used Froude scaling. The model is geometrically scaled. According to the Froude scaling, some important parameters should be scaled as shown in Table 1. The scaling of the heat fluxes and internal wall temperatures is analyzed in the following.

A1. Scaling of heat conduction in the walls

A1.1 Thermally thick materials

It is well known that the heat conduction normal to the wall surface dominates the heat conduction in the wall. Therefore, only the one dimensional heat conduction equation for the material temperature is analyzed here.

Here we classify the wall materials into two categories: thermally thick materials and thermally thin materials. For a thermally thick material, there is always a temperature gradient inside the materials, even after the thermal penetration, which is considered as being important for us. For a thermally thin material, the temperatures inside are homogeneous. It can be easily known that the thermally thin materials are only special examples of the thermally thick materials. There is no clear distinction between these two types of “materials”, but depends on the specific case that is investigated. Generally, metal objects and very thin materials can be considered as thermally thin materials and others are thermally thick materials.

For thermally thick materials, the controlling equation for heat conduction can be written in:

$$\rho_s c_s \frac{\partial T_s}{\partial t} = k_s \frac{\partial^2 T_s}{\partial z^2} \quad (A1)$$

and the boundary conditions at $z=0$ and $z=\delta_s$, can be expressed as:

$$k_s \frac{\partial T_s}{\partial z} = \dot{q}_c'' + \dot{q}_r'' \quad (A2)$$

The above equations can be made dimensionless by introducing the following normalizing parameters: reference time t_o , reference temperature T_o , and reference thickness δ_s . The dimensionless variables are denoted by ($\hat{}$).

The controlling equation for heat conduction is normalized to:

$$\frac{\partial T_s}{\partial \hat{t}} = \frac{k_s t_o}{\rho_s c_s \delta_s^2} \frac{\partial^2 T_s}{\partial \hat{z}^2} \quad (A3)$$

with the boundary condition:

$$\frac{\partial T_s}{\partial \hat{z}} = \frac{(\dot{q}_c'' + \dot{q}_r'') \delta_s}{k_s T_o} \quad (A4)$$

Then we get π groups:

$$\pi_1 = \frac{k_s t_o}{\rho_s c_s \delta_s^2} \quad (A5)$$

and

$$\pi_2 = \frac{(\dot{q}_c'' + \dot{q}_r'') \delta_s}{k_s T_o} \quad (A6)$$

Here we may assume that the convection and radiation heat fluxes are well scaled, that is, $\dot{q}_c'' \propto l^{1/2}$ and $\dot{q}_r'' \propto l^{1/2}$. Thus Eq. (A5) and Eq. (A6) indicate:

$$\frac{\rho_s c_s \delta_s^2}{k_s} \propto l^{1/2} \quad (A7)$$

and

$$\frac{\delta_s}{k_s} \propto l^{-1/2} \quad (A8)$$

Introducing (A8) into (A7) suggests:

$$k_s \rho_s c_s \propto l^{3/2} \quad (A9)$$

$$\frac{k_s}{\delta_s} \propto l^{1/2} \quad (A10)$$

The wall materials and the noncombustible surface materials should be chosen according to the above two equations. Note that based on the scaling, the wall temperature at depth δ in full scale generally corresponds to that at a different position in model scale.

We may simply check the obtained scaling laws. At the beginning of an enclosure fire, that is, when the heat has not yet penetrated the walls, the conductive heat fluxes through wall surfaces can be written in:

$$\dot{q}_k'' = k_s \frac{dT_s}{dz} \propto \frac{k_s (T_w - T_i)}{\sqrt{k_s t} / \rho_s c_s} \quad (A11)$$

This indicates

$$\dot{q}_k'' \propto (k_s \rho_s c_s)^{1/2} l^{-1/4} \quad (A12)$$

Note that $k_s \rho_s c_s \propto l^{3/2}$, the above equation indicates

$$\dot{q}_k'' \propto l^{1/2} \quad (A13)$$

After a period when the thermal penetration occurs, the conductive heat flux through wall surfaces can be expressed as:

$$\dot{q}_k'' = \frac{k_s}{\delta_s} (T_w - T_{bw}) \quad (A14)$$

Note that $k_s / \delta_s \propto l^{1/2}$, the above equation indicates the same results.

Therefore it is clear that the heat conduction in thermally thick materials can be scaled well if Eq. (16) and Eq. (17) are fulfilled in model scale and the wall surface temperature scales well.

A1.2 Thermally thin materials

For thermally thin materials, e.g. thin wall materials or metal objects, the properties inside the materials can be assumed to be homogeneous, and the controlling equation can be simply expressed as:

$$\rho_s V_s c_s \frac{\partial T_s}{\partial t} = \dot{q}_{net}'' A_s \quad (A15)$$

The characteristic parameters are introduced to normalize the above equation:

$$\frac{\partial T_s}{\partial \hat{t}} = \frac{\dot{q}_{net}'' A_s t_o}{\rho_s V_s c_s T_o} \quad (A16)$$

Here we also assume that the convection and radiation heat fluxes follow are scaled well, that is, $\dot{q}_{net}'' \propto l^{1/2}$. Thus the above equation indicate:

$$\rho_s V_s \propto l^3 \quad (A17)$$

This suggests that in order to scale the heat conduction in thermally thin materials, the same materials can be used, if the materials are geometrically scaled. In reality, the scaling of the thermally thick materials also fulfills Eq. (A17) assuming the temperatures inside are homogeneous. As discussed previously, a thermally thin material is only one special example of a thermally thick material.

In summary, to scale the heat conduction inside the thermally thick materials, Eqs. (A9) and (A10) need to be preserved while choosing the materials and the wall thickness. For thermally thin materials, the same materials can be used if the materials are geometrically scaled.

A2. Scaling of convection heat transfer

In an enclosure fire, the key pattern of the convection heat transfer is the forced heat transfer due to the movement of the ceiling jets in the upper layer and also the forced heat transfer in the lower layer due to the movement of fresh air through the openings. The main mechanism should be forced convection heat transfer. For turbulent flow, the convection heat transfer coefficient h_c could be correlated with the Nusselt Number and the Prandtl number:

$$\text{Nu} = \frac{h_c l}{k} = 0.037 \text{Re}^{4/5} \text{Pr}^{1/3} \quad (A18)$$

where

$$\text{Re} = \frac{ul}{\nu}, \quad \text{Pr} = \frac{\nu}{a}$$

The above equations indicate:

$$\dot{q}_c'' \propto h_c \propto l^{1/5} \quad (A19)$$

For laminar flow, the convection heat transfer coefficient h_c can also be correlated with Nusselt Number and Prandtl number, which can be expressed as:

$$\text{Nu} = \frac{h_c l}{k} = 0.66 \text{Re}^{1/2} \text{Pr}^{1/3} \quad (A20)$$

This means:

$$\dot{q}_c'' \propto h_c \propto l^{-1/4} \quad (A21)$$

For a heated wall at lower layer, the heat transfer could be natural convection, and the Nusselt number could be expressed as:

$$\text{Nu}_L = 0.678 \text{Ra}_l^{1/4} \left(\frac{\text{Pr}}{\text{Pr} + 0.952} \right) \quad (A22)$$

where the Rayleigh number is defined as:

$$\text{Ra}_l = \frac{g \beta \Delta T l^3}{\nu^2} \text{Pr} \quad (A23)$$

Thus we also obtain Eq. (A21).

Comparing Eqs. (A19) and (A21) with Eq. (8) shows that the convection heat transfer is overestimated in model scales, especially for laminar flow.

A3. Scaling of Radiation heat transfer

The radiation heat transfer coefficient on wall surfaces could be written in a similar form similar to convection heat transfer:

$$h_r = \varepsilon_g \sigma (T_g^2 + T_o^2) (T_g + T_o) \quad (A24)$$

where

$$\varepsilon_g = 1 - e^{-\kappa_m L_m} \quad (A25)$$

It is clear that the emissivity of the gas in the model scale gets smaller than in large scale, however, the ratio is difficult to estimate. Here we just make a simple analysis of the optically thick and optically thin cases.

In the optically thick cases, the emissivity is close to unit. Therefore the scaling of radiation heat flux is:

$$\dot{q}_r'' \propto l^0 \quad (A26)$$

In the optically thin case, κL_e approaches zero. Therefore the scaling of radiation heat flux is:

$$\dot{q}_r'' \propto \kappa_m L_m \propto l \quad (A27)$$

Note that the radiation heat flux is the inverse of the radiation resistance R_r . At the beginning of an enclosure fire, the scenario is optically thin. However, after a few minutes, the scenario could turn to be optically thick. Therefore, the radiation heat flux could be scaled as l^a ($0 < a < 1$). Note that the scaling of the heat flux should scale as $l^{1/2}$. It could be expected that in model scale, the radiation heat flux may be underestimated at the beginning of the fire and overestimated a certain time after ignition. Based on tests data the coefficient a in the optically thick case could be ascertained for some given locations.

The absorption coefficient is mainly depends on the soot yields and smoke volume flow rate. In other words, it depends on the fuel type, ventilation condition and the specific geometries of the room and the burner. In model scales, if the fuels are the same as in full scale, it can be expected that the absorption coefficient is also the same as in full scale, given that the ventilation conditions and specific geometries are well scaled. Therefore, the scaling of radiation heat flux is mainly related to the mean beam length. In engineering applications, the global radiation heat flux could be calculated using a mean beam length, L_e , which could be estimated by:

$$L_m = 3.6 \frac{V_b}{A_b} \quad (A28)$$

where V_b is the volume of the hot gases and A_b is its bounding area. The mean beam length in an enclosure fire is mainly related to the smoke depth. Therefore, it varies with the fire sizes. For large flames and sooty smoke, the emissivity always approach to unit and the mean beam length plays an insignificant role in the radiation. However, for small fires, both the absorption coefficient and the smoke depth are small values, and thus it could play a key role in the total emissivity.

We may chose different fuels in model scales to explicitly scale the radiation heat flux in enclosure fires. However, it should be kept in mind that the absorption coefficient is not only related to the fuel type and heat release rate, but also the length scale, the entrainment of the smoke flows, and the combustion conditions, e.g. an under-ventilated fire produces much more soot. In realistic fires, the absorption coefficient is a time-dependent variable during a fire. In any case, the scaling of radiation is difficult based on the above analysis. However, in reality, the fire scales the radiation by itself. Note that limited heat is available to be lost by radiation. Therefore in a real fire, the temperature will decrease if the emissivity is too high in model scales. The radiation fraction in the total heat release rate in open fires and enclosure fires have been observed to be around 20 % to 40 %. This mainly results from the self-adjustment of the heat radiation. Although the gas temperature could decrease in model scales, the different in the gas temperature between scales could still be insignificant since heat radiation is proportional to the fourth power of gas temperature. In short, the scaling of radiation heat transfer is still expected to be acceptable in model scales.

A4. Scaling of heat balance in an enclosure

The total heat released in a lightweight material constructed room could be balanced by the heat loss by smoke flow out through openings, i.e. doors and windows, Q_{conv} , and the heat loss by conduction into the walls, Q_k , and the the radiation through the openings, Q_r .

$$Q = Q_c + Q_k + Q_r \quad (A29)$$

A1.1 Heat loss by convection through vents

The heat loss by convection through the vents can be expressed as:

$$Q_c = \dot{m}_g c_p (T_g - T_o) \quad (A30)$$

In an enclosure fire, the mass flow rate of smoke flow through an opening normally can be written in such a way:

$$\dot{m}_g \propto C_d A_o \sqrt{\Delta P} \quad (A31)$$

Assuming that the smoke layer can be scaled well and noting that the pressure difference are scaled as the length scale, the smoke mass flow rate through an opening should approximately be scaled as:

$$\dot{m}_g \propto l^{5/2} \quad (A32)$$

For a flashover fire, which is probably ventilation controlled, the mass flow rate of the fresh air flowing into the enclosure should equal the smoke mass flow rate out of the opening, both of which can be approximately expressed:

$$\dot{m}_{air} = \dot{m}_g = 0.5 A_o \sqrt{H_o} \propto l^{5/2} \quad (A33)$$

The heat loss by smoke flow out through the openings can be expressed as:

$$Q_{conv} \propto l^{5/2} \quad (A34)$$

According to this, it is clear that the heat loss by smoke flow out through openings can scale well even for a flashover fire.

A1.2 Heat loss by conduction into the walls

According to the analyses in Section 2.2.1 and 2.2.2, the convection heat flux and the radiation heat flux cannot be scaled precisely as $l^{1/2}$. This definitely will affect the heat conduction inside the wall, since all the heat into the walls comes from the wall surfaces. To analyze the influence of convection and radiation heat transfer on the heat conduction in the walls, the circuit analog of the heat loss to the walls in an enclosure fire is given, as shown in Fig. 1.

Figure A1. Circuit analog of heat loss to the walls in an enclosure fire

According to Fig. 1, the total heat flux to can be simply expressed as:

$$\dot{q}'' = \frac{(T_g - T_o)}{R_t} = \frac{(T_g - T_o)}{\bar{R} + R_k} \propto l^{1/2} \quad (\text{A35})$$

where

$$R_k = \frac{1}{h_k}, R_c = \frac{1}{h_c}, R_r = \frac{1}{h_r}, \bar{R} = \frac{R_r R_c}{R_r + R_c}$$

The heat loss by conduction to the walls can also be expressed as:

$$Q_k = h_t A_w (T_g - T_o) \quad (\text{A36})$$

where the total heat transfer coefficient h_t is defined as:

$$\frac{1}{h_t} = \frac{1}{h_c + h_r} + \frac{1}{h_k} \quad (\text{A37})$$

The heat flux in model scale should scale as:

$$\dot{q}'' \propto \frac{Q}{A} \propto l^{1/2} \quad (\text{A38})$$

According to the previous analysis, it is clear that the conductive heat transfer is scaled as 1/2 power of length scale, assume that Eq. (16) and Eq. (17) are fulfilled in model scale and the other parameters are scaled well. However, in practice the heat flux may not scale as 1/2 power of the length scale due to the influence of the scaling of the convection heat flux and the radiation heat flux. This will be discussed in the following.

Note that it is assumed that the radiation heat transfer is directly related to the flame or gas temperature and is proportional to the temperature difference, i.e. has the same form as the convection heat transfer. However, the radiation heat transfer coefficient is not a constant except for a constant radiation temperature. The formation of the equation is only used for a clear expression of the interaction between different modes of heat transfer.

The above equation indicates that the all the heat resistances should be scaled as:

$$R \propto l^{-1/2} \quad (\text{A39})$$

Note that the smaller one between R_r and R_c dominates the heat transfer to the wall surfaces. Quintiere [36] gave typical ranges for the heat transfer coefficients where $h_k \approx$

10 - 30 W/m²·K, $h_r \approx 5 - 100$ W/m²·K, and $h_c \approx 5 - 60$ W/m²·K. For commonly used gypsum board, the conductive heat transfer coefficient is about 28 W/m²·K in half an hour after ignition and 14 W/m²·K in one hour. We can also calculate the radiation heat transfer coefficient using Eq. (36). Assume that the emissivity equals 0.8 and the wall surfaces are bounded by hot gases, the radiation heat transfer coefficient is about 34 W/m²·K for a gas temperature of 500 °C and 125 W/m²·K for 1000 °C. It is clearly shown that after the gas temperature increases to about 500 °C the radiation dominates the heat transfer to the wall surface. It can be expected that for a large enclosure fire, the conductive heat transfer dominates the total heat transfer from the hot gases to the surrounding walls during a long period. Especially in this project, the walls are covered by insulating materials, i.e. mineral wools. This means that if the heat conduction scales well, the total or overall heat transfer should also scale well in such cases. In our cases, the wall temperatures between the wall surface (T_g) and the backside (T_o) are the focus.

Note that the total heat transfer corresponds to the heat transfer from the flame and hot gases (T_g) to the penetration boundary inside the wall (T_o). However, the penetration boundary moves deeper into the wall as the time goes by. Therefore, the effective thermal resistance of the wall is not constant but increases with time. For the wall temperatures at a position close to the wall surface (far away from the penetration boundary), the effective thermal resistance of the wall at the beginning of the fire could not dominate the overall heat transfer from the hot gases to the wall, and thus the internal wall temperature could not scale very well. However, as the penetration depth increases with time and the thermal resistance soon becomes the dominant term in the overall heat transfer from hot gases to the internal walls. In such cases, the internal wall temperatures should scale well.

A1.3 Heat loss by radiation through the vents

The heat loss by radiation through the vents can be written in:

$$Q_r = \phi \varepsilon \sigma A_o (T_g^4 - T_o^4) \quad (A40)$$

where ϕ is the ratio of the effective radiation area to the area of the opening, which could be approximately estimated based on the smoke layer depth at the opening.

For enclosures with small openings, the heat loss by radiation through the vents can be ignored, compared to the others. However, for enclosures with large openings, the heat loss by radiation through the vents should be taken into account.

The heat loss by radiation through the vents scales the same as the radiation heat flux. Its effect on the entire heat balance mainly depends on the vents area. In our cases, the heat loss by radiation through the vents is not so important.

A5. Short summary

Based on the above analysis, the scaling of the heat fluxes is given, as shown in Table A1. Note that at a certain time after ignition, especially for a sooty fire, the main thermal resistance of the heat transfer from the hot gases to the backside wall comes from the wall materials. Note that the materials used in the tests are related to thermally thick materials. It can be expected that the heat conduction and the temperature inside these wall materials can be scaled well if Eq. (A9) and Eq. (A10) are fulfilled in model scale.

In summary, it is known that a large amount of heat released in an enclosure fire is taken away by smoke flow out through openings, which is normally called the convective heat release rate. The fraction of the convective heat release rate in the total heat release rate is normally in a range of 60 % to 80 %. This is the main reason why the simplified Froude

scaling can scale the fire scenario well even in model scales even when all the heat fluxes are explicitly scaled.

Table A1 Extra list of scaling correlations for heat fluxes.

Type of unit	Scaling	Equation number
	$(k_s \rho_s c_s)_M / (k_s \rho_s c_s)_F \propto (l_M / l_F)^{3/2}$	(A41)
Thermally thick materials	$(\frac{k_s}{\delta_s})_M / (\frac{k_s}{\delta_s})_F \propto (l_M / l_F)^{1/2}$	(A42)
Thermally thin materials	$\rho_s V_s \propto l^3$	(A43)
Conductive heat flux (kW)	$\dot{q}_{cond,M}'' / \dot{q}_{cond,F}'' = (l_M / l_F)^{1/2}$	(A44)
Radiation heat flux	$\dot{q}_{r,M}'' / \dot{q}_{r,F}'' = (l_M / l_F)^a (0 \leq a \leq 1)$	(A45)
Convection heat flux	$\dot{q}_{c,M}'' / \dot{q}_{c,F}'' = (l_M / l_F)^a (-1/4 \leq a \leq 1/5)$	(A46)

Appendix B - Layout of instrumentation in model scales

Figure B-1. Layout of instrumentation in 1:2 model tests.

Figure B-2. Layout of instrumentation in 1:3.5 model tests.

Appendix C - Photos of the tests

(a) First stage

(b) Second stage

Figure C-1. Photos in full scale test 1.

(a) First stage

(b) Second stage

Figure C-2. Photos in medium scale test 2.

(a) First stage

(b) Second stage

Figure C-3. Photos in medium scale test 3.

(a) First stage

(b) Second stage

(c) Third stage

Figure C-4. Photos in full scale test 4.

(a) First stage

(b) Second stage

(c) Third stage

Figure C-5. Photos in medium scale test 5.

(a) First stage

(b) Second stage

(c) Third stage

Figure C-6. Photos in small scale test 6.

(a) First stage

(b) Second stage

(c) Third stage

Figure C-7. Photos in medium scale test 7.

Appendix D - Test Results

Figure D1 Gas temperatures, velocity, heat flux and concentration in Test 1.

Figure D2 Measured temperatures inside the wall at positions TS1 to TS5 in Test 1.

Figure D3 Gas temperatures, velocity, heat flux and concentration in Test 2.

Figure D4 Measured temperatures inside the wall at positions TS1 to TS5 in Test 2.

Figure D5 Gas temperatures, velocity, heat flux and concentration in Test 3.

Figure D6 Measured temperatures inside the wall at positions TS1 to TS5 in Test 3.

Figure D7 Gas temperatures, velocity, heat flux and concentration in Test 4.

Figure D8 Measured temperatures inside the wall at positions TS1 to TS5 in Test 4.

Figure D9 Gas temperatures, velocity, heat flux and concentration in Test 5.

Figure D10 Measured temperatures inside the wall at positions TS1 to TS5 in Test 5.

Figure D11 Gas temperatures, velocity, heat flux and concentration in Test 6.

Figure D12 Measured temperatures inside the wall at positions TS1 to TS5 in Test 6.

Figure D13 Gas temperatures, velocity, heat flux and concentration in Test 7.

Figure D14 Measured temperatures inside the wall at positions TS1 to TS5 in Test 7.

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