New models for calculating maximum gas temperatures in large tunnel fires

Ying Zhen Li
Haukur Ingason

Fire Research
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Abstract

The work presented in this report focuses on estimating maximum gas temperatures at ceiling level during large tunnel fires. Gas temperature is an important parameter to consider when designing the fire resistance of a tunnel structure. Earlier work by the authors has established correlations between excess ceiling gas temperature and effective tunnel height, ventilation rate, and heat release rate. The maximum possible excess gas temperature was set as 1350°C, independent of the tunnel structure and local combustion conditions. As a result of this research, two models have been developed to better estimate possible excess maximum gas temperatures for large tunnel fires in tunnels with differing lining materials and structure types (e.g. rock, concrete). These have been validated using both model- and full-scale tests. Comparisons of predicted and measured temperatures show that both models correlate well with the test data. However, Model I is better and more optimal, due to the fact that it is more conservative and easier to use. The fire duration and flame volume are found to be related to gas temperature development. In reality, the models could also be used to estimate temperatures in a fully developed compartment fire.

Key words: Gas temperature, tunnel structure, maximum ceiling temperature, velocity, heat release rate, ceiling height, tunnel cross-section.

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Preface

This research was funded by the STA (Swedish Transport Administration; Trafikverket) and is presented in two reports; one on theoretical aspects (this document), and one on more practical applications of the results (STA report ‘TRV 2016/69/69492’, available in Swedish). The project was a co-operative effort between Brandskyddslaget AB, a consultancy firm that works with tunnel infrastructure projects in Sweden, and RISE Fire Research, which has a great deal of experience in undertaking research in the field of underground fires. We acknowledge the co-authors of the STA report ‘TRV 2016/69/69492’ – Niclas Åhmberg and Johan Häggström of Brandskyddslaget AB – for their co-operation, as well as Olle Olofsson and Maria D. Nilsson at the STA for their support and encouragement during the progression of this project. We also acknowledge Professor Ulf Wickström at Luleå University of Technology, for fruitful discussions and valuable suggestions.
Summary

This report presents new models for predicting the maximum excess ceiling gas temperature inside a tunnel. It constitutes further development of the work presented by the authors for small [1] and large [2] tunnel fires. In the previous MT (maximum temperature) model, a maximum possible excess gas temperature of 1350 °C, which was the highest measured tunnel ceiling excess gas temperature obtained, was set. However, this gas temperature in reality is a function of the HRR (heat release rate), fuel geometry, effective tunnel height, and ventilation conditions in the vicinity of the fire. In this work, two new models have been developed. In the new MT Models I and II, the type of tunnel structure and lining, as well as cross-sectional area, have been introduced as parameters in order to better simulate the thermal response of smoke flows in tunnels in the vicinity of a fire. This has enabled improvements to be made to the model, and facilitated more accurate predictions of the maximum possible excess gas temperature beneath tunnel ceilings during large fires.

Through comparisons of calculation models with experimental data from both full- and small-scale tests, the predictions made using both models correlate well with the test data. However, Model I is more optimal due to the fact that it is more conservative and easier to use.

The new MT models factor in the effects of scaling through the energy balance of the flame volume in the vicinity of the fire. This enables final gas temperature to be more accurately predicted for a given duration. In the previous model for large fires [2], the maximum excess gas temperature on both large and small scales was difficult to predict due to the fact that no consideration was given to the thermal response of the surrounding structure, as well as the fact that scaling was not factored in. The previous model resulted in very high excess gas temperatures for large fires on the model scale, while the new ones predict scaled experimental data in a more realistic fashion. The importance of the duration of the fire is also factored into the model in the form of both thermal response and the heating of lining surfaces near to the fire. The most important parameters with regard to determining the final excess gas temperature are the exchange of flame heat flux and the heating of nearby surfaces. The models can factor in different types of lining material, road surface material, and burning vehicle envelopes.

The models could also be used in the real world to estimate temperatures in a fully developed compartment fire, with the exception of different definitions for two key parameter.

The model has been validated against both large- and model-scale test data. The correspondence is encouraging, and further development and validation of the model by testing has been proposed. It is felt that focus should be placed on parametric study of the effects of lining, ventilation, and geometry in relation to both fuel geometry and tunnel cross-section.
### Nomenclature

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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tr>
<td>$a$</td>
<td>thermal diffusivity ($\text{m}^2/\text{s}$)</td>
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<tr>
<td>$a_1$</td>
<td>coefficient in Eq. (10)</td>
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<td>$a_2$</td>
<td>coefficient in Eq. (10)</td>
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<tr>
<td>$A$</td>
<td>tunnel cross-sectional area ($\text{m}^2$)</td>
</tr>
<tr>
<td>$A_F$</td>
<td>area of flame profile ($\text{m}^2$)</td>
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<tr>
<td>$A_{ob}$</td>
<td>contact area between flame and object ($\text{m}^2$)</td>
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<tr>
<td>$A_w$</td>
<td>contact area between flame and tunnel ($\text{m}^2$)</td>
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<tr>
<td>$b_{fo}$</td>
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**Greek**

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1 Introduction

The heat that a tunnel structure is exposed to in the event of fire is described using standardised time-temperature curves. These vary depending on the type of vehicle using the tunnel, e.g. road vehicles or rolling stock. The time-temperature curves represent thermal loads, informing designers regarding the load-bearing capacity of structures. The primary problem is selecting the correct time-temperature curve with regard to a specific, designed fire in terms of its energy output in MW, meaning that the time-temperature curves of each tunnel project are often determined independent of the fire in question.

Heat exposure is represented by a standard time-temperature curve such as ISO 834 [3], the hydrocarbon curve (HC) [4], or the RWS curve [5], which are widely used to test the fire performance of tunnel linings and are shown in Figure 1.

![Figure 1](image-url)  
*Figure 1  The most common standardised time-temperature curves used in tunnel applications.*

Other time-temperature curves that are used in specific applications include RABT/ZTV [6], HCM [7], and EBA [8]. Each of these is derived in a different way, and most are based on large- or small-scale tests or agreements within technical committees working nationally or internationally. When choosing between different curves, no single guideline document suggests how to choose a curve in relation to engineering parameters such as HRR, longitudinal ventilation velocity, or ceiling height.

Therefore, Li et al. [1] and Li and Ingason [2] developed a method for calculating excess ceiling gas temperature as a function of these parameters, which can produce time-temperature curves using realistic HRRs (design fires). Alternatively, through comparison with the obtained time-temperature curve, a standardised time-temperature curve, e.g. the HC curve, can be selected for use in a specific project. The temperature value obtained can be used to calculate the heat flux that the structure is exposed to, and the temperature inside the structure can be calculated as a function of the distance from its exterior surface. The internal temperature of the reinforced bars of a concrete tunnel structure can be calculated; when it reaches a certain, critical value, the time to failure can be determined.
Standardised time-temperature curves are usually obtained from prescriptive-type codes, with authorities selecting one based on a given design fire. As engineering methods become more widespread, however, requirements may focus more on performance-based design – an approach that is perhaps quite likely in future tunnel projects, given the predicted development of better engineering models.

The original work that attempted to correlate ceiling gas temperature in tunnels with HRR was carried out by Kurioka et al. [9] who proposed an empirical equation to predict the maximum gas temperature rise below the tunnel ceiling and its position relative to the centre of the fire. Hu et al. [10] compared Kurioka’s equation with their full-scale data and showed that there was a good agreement between the two. However, the fire sizes in the full-scale tests of Hu et al. [10] were quite small. In the equation given by Kurioka et al., the increase in maximum gas temperature below the ceiling approaches infinity when the ventilation velocity approaches zero, meaning that it is increasingly difficult to correctly predict increases in maximum gas temperature below the ceiling as the ventilation velocity approaches zero. Further, the maximum excess gas temperatures in the work of Kurioka et al. were set to approximately 770°C, which is much lower than the values measured in large-scale tunnel fire tests. Moreover, the proposed correlation was originally obtained through empirical correlation rather than theoretical analysis [2].

The equations proposed by Li et al. [1] for small tunnel fires were compared with those of Kurioka et al. The results showed that the former are better able to predict the maximum gas temperature, particularly for low ventilation velocities. However, the proposed equations may not be valid if the HRR is so large that the combustion zone extends to the tunnel ceiling. In such a scenario, the maximum gas temperature is expected to be a constant that is independent of HRR, ventilation velocity, and tunnel height, and so Li and Ingason [2] proposed a new correlation for large fires, which is the basis for the work presented in this report.

Li et al. [1] found that the experimental data can be divided into two regions according to the dimensionless ventilation velocity, which is defined as the ratio of the longitudinal ventilation velocity to the characteristic plume velocity. The main parameters taken into account in the theoretical analysis include HRR, ventilation velocity, effective tunnel height, and the dimensions of the fire source. For a small fire in a tunnel, the maximum excess gas temperature beneath the tunnel ceiling increases linearly with HRR, and decreases linearly with the longitudinal ventilation velocity when the dimensionless ventilation velocity exceeds 0.19 m/s. When the dimensionless ventilation velocity is \( \leq 0.19 \) m/s, the maximum gas excess temperature beneath the tunnel ceiling varies as a function of a two-thirds power law of the HRR, independent of the longitudinal ventilation velocity. In both regions, the maximum gas excess temperature varies as a function of a -5/3 power law of the effective tunnel height, i.e. the vertical distance between the ceiling and the bottom of the burning object.

For a large fire in a tunnel – one in which the flame impinges on and extends along the tunnel ceiling – it was found that the maximum excess temperature beneath the ceiling approaches a constant value, regardless of the ventilation velocity. However, it was noted that the thermal properties of the tunnel structure, water dripping and flowing from cracks in blasted rock tunnels, the duration of the high temperature period (due to e.g. high HRR and low ventilation velocity), and fuel type are key parameters that can influence the specific value of the maximum temperature in any given tunnel for any given scenario [2]. Thus, the maximum temperature in regions of near-constant temperature is not a universal constant, but dependent on the specific conditions of a given tunnel fire. This is of critical importance for the further development of the improved model presented herein.
Based on a theoretical model and analysis of the experimental data, a correlation for the maximum temperature beneath the tunnel ceiling was proposed by Li and Ingason [2]. This has been found to be valid up to a maximum excess gas temperature of 1350°C, which was the maximum temperature obtained during large-scale tunnel tests. It was, however, clear that maximum gas temperature levels depend mainly on HRR, effective tunnel height, and ventilation type and velocity. Other factors that are less instrumental but nonetheless important when determining the level of the maximum ceiling temperature, are the type of tunnel lining or structure (i.e., rock, concrete, cracked and leaking blasted rock) and fuel type [2].

The fuel type category can be divided into vehicles (i.e. the vehicle’s fire load) and solid fuels and liquid fuels. Solid or liquid fuel fires with no coverage have been found to yield the highest maximum ceiling gas temperatures, provided HRRs are sufficiently high. The various types of materials used to construct vehicles, such as steel, aluminium, and fibreglass, affect the amount of convective and radiant heat transported towards the ceiling, in turn affecting the final maximum ceiling gas temperature value for the same HRR. It was also observed that gas temperatures in model-scale tests do not exceed 1100°C. This is related to the specific conditions and thermal exchange between the fire and the material used in these model-scale tests. This observation, reported by Li and Ingason [2], has been investigated further in this work.

The model presented herein is based on a theoretical approach, and has been partially validated using available experimental data. There is, however, a need to validate all of the new parameters introduced into the model in a systematic way. The best method of accomplishing this is to carry out fire tests using a scale model. In the following, the theoretical development – on the part of both Li et al [1, 2] and the authors of this report – is presented.
2 Theory

2.1 The previous maximum gas temperature model

Li et al. developed a model of maximum ceiling gas temperature in tunnel fires under variable ventilation conditions [1, 2]. Much effort has been made to validate this, on both full and model scales.

This model was the basis for the development of the new models described in this work, and is referred to throughout as the “previous model”. Figure 2 is a diagram that shows a turbulent flame impinging on a tunnel ceiling during a large tunnel fire.

![Diagram of a turbulent flame impinging on a tunnel ceiling during a large tunnel fire.](image)

Figure 2 A turbulent flame impinging on a tunnel ceiling during a large tunnel fire.

The maximum excess gas temperature beneath the ceiling during a tunnel fire is dependent on the HRR but independent of ventilation velocity if the ventilation velocity across the fire source is very low as compared to the HRR. However, it approaches a constant if the area of the flame volume containing the combustion zone is at the level of the tunnel ceiling. In other words, if \( V' \leq 0.19 \) (‘Region I’), the maximum excess gas temperature can be expressed as:

\[
\Delta T = \begin{cases} 
DTR I, & \text{if } DTR I < 1350 \\
1350, & \text{if } DTR I \geq 1350
\end{cases}
\] (1)

where the Delta T in Region I, DTRI, is defined as:

\[
DTR I = 17.5 \frac{Q^{2/3}}{H_{ef}^{5/3}}
\]

and dimensionless velocity is defined as:

\[
V' = u_o \left( \frac{\rho c_p T_b b_0}{gQ} \right)^{1/3}
\]

If the ventilation velocity across the fire source increases, the maximum excess temperature beneath the ceiling depends on both the HRR and ventilation velocity. However, maximum excess temperature also approaches a constant if the continuous combustion zone is at the level of the tunnel ceiling. In other words, if \( V' > 0.19 \) (‘Region II’), the maximum excess gas temperature can be expressed as:

\[
\Delta T = \begin{cases} 
DTR II, & \text{if } DTR II < 1350 \\
1350, & \text{if } DTR II \geq 1350
\end{cases}
\] (2)

where the Delta T in Region II, DTRII, is defined as:
The maximum flame temperature was considered to be a constant, e.g. 800 °C for excess flame temperature according to McCaffrey’s fire plume theory [11]. However, gas temperatures exceeding 1000°C and up to 1365-1370°C were measured beneath tunnel ceilings during full-scale testing such as the tests carried in the Runehamar [12, 13] and Memorial tunnels [14].

There are three primary reasons for the high gas temperatures beneath the tunnel ceiling. Firstly, the size (HRR) of such fires is generally large; secondly, forced ventilation enhances the combustion; thirdly, the heat feedback from the tunnel structure increases the gas temperature. In an open fire, by comparison, the energy of the flame and hot gases is dissipated in the surrounding area as radiation with almost no heat feedback from the surroundings. Furthermore, in an enclosure fire, the maximum HRR is restricted to a great extent by the openings (e.g. doors, windows).

Figure 3 and Figure 4 show the maximum ceiling excess gas temperatures in tunnel fires in Region I ($V' \leq 0.19$) and Region II ($V' > 0.19$) respectively. The test data correlated well with the above equations.

**Figure 3**  Maximum excess gas temperature beneath the tunnel ceiling (Region I) [2].

**Figure 4**  Maximum excess gas temperature beneath the tunnel ceiling (Region II) [2].
Figures 3 and 4 show that most of the data pertaining to large fires (in the regions of near-constant temperatures) lies in a narrow region, and that the use of 1350°C is relatively conservative. However, the temperatures obtained during model-scale testing are universally lower than those obtained during large-scale testing. This is likely due to the fact that the maximum excess gas temperature is also dependent on other parameters, such as the thermal properties of the tunnel wall. This will be examined in the following analysis.

Note that a parameter termed ‘radius of the fire source’, \( b_{ps} \), is used in the above equations. The radius of a circular fire source is easy to determine but, for that of a rectangular fire source, a circular radius equivalent to the area of the fire source should be used, i.e. \( \sqrt{A/\pi} \) (where \( A \) is the horizontal or projected area of the fuel source). The same method has been applied to wood crib fires. In such cases, the projection area (or bottom area) of the fire source can be regarded as being equivalent to the fire source.

It should also be noted that the height values used herein do not relate to tunnel height, but to effective tunnel height, \( H_{ef} \), i.e. the vertical distance between the bottom of the fire source and the tunnel ceiling. In arched tunnels, this is the height of the tunnel ceiling directly above the location of the fire. This parameter is very important in determining the maximum excess gas temperature beneath the ceiling.

The correlations can be used to calculate excess gas temperature as a function of HRR and ventilation velocity for any given tunnel and fire scenario. These equations are extremely useful for converting a standardised time-temperature curve for a given tunnel into a corresponding HRR curve, and vice versa.

This model is termed the ‘previous MT model’.

### 2.2 Adiabatic flame temperature of diffusion flames

It is that the gas temperatures of diffusion flames are lower than those of pre-mixed flames – generally below 1350°C. Quintere [15] proposed the following correlation for flame temperature based on data in the literature:

\[
\Delta T_F = C_{T,f} \left(1 - X_r\right) \frac{Y_{O2} \Delta H}{c_p r}
\]  

(3a)

where \( X_r \) is radiation fraction, \( C_{T,f} \) is a correlation factor, \( Y_{O2} \) is oxygen concentration, \( \Delta H \) is the heat of combustion of the fuel, \( c_p \) is heat capacity, and \( r \) is the stoichiometric mass ratio of air to fuel.

By setting \( X_r \) to 0, the adiabatic temperature for diffusion flames, \( T_{ad} \), can be obtained:

\[
\Delta T_{ad} = \Delta T_{F,max} = C_{T,f} \frac{Y_{O2} \Delta H}{c_p r}
\]

(3b)

The constant \( C_{T,f} \) has been found to lie in the range of 0.48-0.59. An average value of 0.523 is assumed here for the purpose of estimating the adiabatic flame temperature of diffusion flames. In reality, \( C_{T,f} \) can be interpreted as the percentage of oxygen that is
consumed during the process of combustion. Note that, in the above equations, a value of 1 kJ/kg K [15] is used for heat capacity (c_p). Calculations made using Eq. (3b) show that the adiabatic flame temperature is 1581°C; comparing this to the adiabatic temperature of a pre-mixed flame (or a theoretical adiabatic flame temperature) of 2169°C [16] indicates that around 73% of oxygen is consumed in the combustion of a diffusion flame \( C_{TF} = 0.73 \). For carriage fires, it was found that approximately 61% of the oxygen in a carriage is consumed [17], giving an estimated temperature of roughly 1541°C when the heat capacity is estimated to be 1.2 kJ/(kg · K) (based on \( C_{TF} = 0.61 \)). These values correlate reasonably well with each other, but this does not mean that oxygen cannot be entirely depleted during a large tunnel fire. If the fuel is well mixed due to additional turbulence, or several fire sources are burning simultaneously, the oxygen level could be close to zero. This may, however, be interpreted as meaning that the oxygen is consumed step by step, accompanying with the heat loss to the surroundings. Therefore, the value of 1581°C will be regarded as a reasonable value for the adiabatic flame temperature of diffusion flames.

Note that the adiabatic flame temperature of diffusion flames is proportional to the oxygen mass fraction (see Eq. (3b)). In well-ventilated conditions, i.e. when there is a relatively high (2-3 m/s) longitudinal ventilation flow within the tunnel, the oxygen concentration is generally just below 21%. When the velocity falls below this level, the smoke (particles and combustion gases) begins to flow against the direction of the air flow (backlayering), and part of it is drawn back into the fire zone, causing the oxygen concentration to drop below 21%. When the longitudinal ventilation is as low as 0.5 m/s or less, vitiation, i.e. inertisation of oxygen gas with combustion gases, may become an issue due to a significant amount of reverse flow occurring. This highly vitiated air is drawn back into the fire zone, and the oxygen fraction at the fire site therefore falls below ambient levels. This in turn causes the adiabatic flame temperature for diffusion flames to fall, reducing the possibility of very high gas temperatures occurring.

### 2.3 New MT model I

To distinguish the model presented in this section from the previous MT model, it is termed ‘New MT model I’.

#### 2.3.1 Correlation

Through analysis of test data, it is possible to observe the transient behaviour of excess gas temperature in large tunnel fires. Gas temperature does not increase as rapidly above 800-1000°C as below. In other words, it seems that it takes significantly longer time for gas temperatures to reach maximum values above this level. The main reason for this is the strong interaction between the flame volume and the surrounding structure or linings.

The HRR of large fires can be approximately expressed as follows:

\[
\dot{Q} = \dot{m}_g c_p \Delta T_{ad}
\]  

(4)

The energy equation for the flame volume can be expressed as follows:

\[
\dot{Q} = \dot{m}_g c_p (T - T_0) + \dot{Q}_k + \dot{Q}_r
\]  

(5)
where $\dot{Q}_k$ is heat loss to the tunnel structure, and $\dot{Q}_r$ is radiative heat loss to the tunnel sections outside of the flame zone.

The heat resistance of the heat transfer from flames to inner walls is dominated by the conductive heat transfer, as a result of the large incident radiation heat flux from the flames. Therefore, the heat loss to tunnel walls in the flame zone can be expressed as follows:

$$
\dot{Q}_k = h_k A_w (T - T_o)
$$

(6)

where $A_w$ is the surface area of the wall that is within the flame zone, and $h_k$ is the heat transfer coefficient.

Radiation loss to the tunnel sections outside of the flame zone can be expressed as follows:

$$
\dot{Q}_r = \varphi \varepsilon \sigma A (T^4 - T_o^4)
$$

(7)

where $\sigma$ is the Stefan-Boltzmann constant ($5.67 \times 10^{-11}$ kW/(m$^2$·K$^4$)), $A$ is tunnel cross-sectional area (m$^2$), and $\varphi$ is a correction factor. Under high ventilation conditions, the area upstream of the fire is free of smoke and the tunnel area can be used as the radiation area, with the radiation loss at the flame tip downstream as a secondary effect. As a result, the correction factor, $\varphi$, could be set to 1. Under natural ventilation conditions, the flames and hot smoke in the vicinity of the fire will appear mainly in the upper layer, e.g. the upper half of the tunnel height. On each side of the fire site, half of the tunnel cross-sectional area can be used to estimate the radiation loss; here, the re-radiation from the lower part of the tunnel section outside of the flame zone may be assumed to be negligible. Thus, the correction factor, $\varphi$, in Eq. (7) may still be close to 1. As a first approximation, the correction factor of unity, i.e. $\varphi = 1$, is applied.

Emissivity, $\varepsilon$, can be considered to be unity for full-scale tunnel fires. On the model scale, an effective emissivity obtained via testing can be used.

Therefore, from the energy equation, the excess gas temperature can be expressed as follows:

$$
\Delta T(t) = \frac{\Delta T_{ad} - \varphi \varepsilon \sigma A (T(t)^4 - T_o^4) / [\dot{m}_g(t)c_p]}{1 + h_k(t_c) / K_{ef}}
$$

(8)

The parameter, $K_{ef}$, is:

$$
K_{ef} = \frac{\dot{m}_g c_p}{A_w} = \frac{155AH_{ef}^{1/2}}{(1 + C_u)PH}
$$

(9)

where

$$
C_u = \begin{cases} 
1 - 3.3u^* & u^* < 0.3 \\
0 & u^* \geq 0.3 
\end{cases} \text{ and } u^* = \frac{u_o}{\sqrt{gH}}
$$

The deduction of the parameter $K_{ef}$ is discussed in detail in Section 2.3.2.
The analytical solution for the correlation, in a form of \( a_1 T^4 + T - a_2 = 0 \), can be expressed as follows:

\[
T_{\text{avg}} = \frac{1}{2} \sqrt[3]{\frac{3.494 a_2}{a_1^{1/3} (9 + \omega)^{1/3}} - \frac{(9 + \omega)^{1/3} + 4.079}{a_1^{1/3} \psi} - \frac{4.079}{4.079}} \tag{10}
\]

where

\[
\omega = \sqrt[3]{27 + 256 a_1 a_2^3}, \quad \psi = \sqrt{\frac{(18 + 2 \omega)^{2/3} - 14.537 a_1^{1/3} a_2}{a_1^{2/3} (9 + \omega)^{1/3}}}
\]

\[
a_1 = \frac{\sigma A / \dot{m} \varepsilon c_p}{1 + h_k / K_{ef}}, \quad \text{and} \quad a_2 = \frac{\Delta T_{ad} + \sigma A T^4_{o} / \dot{m} \varepsilon c_p + T_o}{1 + h_k / K_{ef}}
\]

Alternatively, an explicit time-marching procedure can be used; i.e. the temperature at the previous stage \( t \) can be used to estimate the temperature at this stage \( t + dt \):

\[
\Delta T(t + dt) = \frac{\Delta T_{ad} - \varphi \varphi \varepsilon A[T(t)^4 - T^4_o]}{1 + h_k (t_c) / K_{ef}}
\]

In this process, the gas temperature at the previous stage, \( T(t) \), is used to estimate the radiation loss in order to obtain the excess temperature at the new stage, \( \Delta T(t + dt) \). Ordinarily, an interval of 10 seconds can be used without producing any large error. This time-marching procedure can be performed easily using an excel sheet.

For manual calculation, a gas temperature can be assumed for \( T \) on the right-hand side, and then a new gas temperature \( T \) can be obtained. After repeating this simple procedure twice, an approximate solution can be obtained.

Note that it is assumed in Eqs. (8) and (10) that the fire is so large that the potential temperature (the temperature obtained using the previous model) is greater than 1350°C. This, however, may not be true in some cases, e.g. if the fire becomes smaller for a period of time. Therefore, there is a need to ascertain whether the values obtained using Eqs. (8) and (10) are reasonable or not. In other words, an estimated value obtained using “new MT model I”, should be lower than that obtained using the “previous MT model”, i.e.:

\[
\Delta T = \min(\Delta T_{\text{previous}}, \Delta T_{\text{new}})
\tag{11}
\]

This means that if \( \Delta T_{\text{previous}} > \Delta T_{\text{new}} \), \( \Delta T_{\text{new}} \) is valid (the minimum value). As the temperature in the regions of near-constant temperature is directly obtained using the new MT model I, the value of 1350°C is no longer used to calculate \( \Delta T_{\text{previous}} \). Instead, \( \Delta T_{\text{previous}} \) can be obtained as follows:

\[
\Delta T_{\text{previous}} = \min(DTR I, DTR II) = \min(17.5 \frac{\dot{Q}^{2/3}}{H_{ef}^{SRT}}, \frac{\dot{Q}}{u_k b_{fo} H_{ef}^{1/3}})
\tag{12}
\]
In other words, the value for $\Delta T_{\text{new}}$ obtained using Eqs. (8) or (10) should be compared to $\Delta T_{\text{previous}}$ according to Eq. (12) before the final excess gas temperature at $t + dt$ is determined.

### 2.3.2 Deduction of the key parameter $K_{ef}$

The surface area of the wall that is within the flame zone, $A_w$, could be estimated using:

$$A_w = C_{\text{con}} PL_f$$

where $L_f$ is total flame length (the sum of the upstream and downstream flame lengths), $C_{\text{con}}$ is the flame length correction factor, and $P$ is the tunnel perimeter. $C_{\text{con}}$ is used to correct the flame length. It is known that the majority of heat is released in the continuous flame region, and experiments [18] have shown that the continuous flame length is mostly in the range of 75-85% of the total flame length. The value of 75% has been selected for the correction factor used here.

Note that the flame length in varying longitudinal flows can be expressed as [18]:

$$\frac{L_f}{H} = C_F(1 + C_u)Q^*$$

where

$$Q^* = \frac{\dot{Q}}{\rho_c c_p T_o S^{1/2}AH_{ef}^{1/2}}$$

$$C_u = \begin{cases} 1 - 3.3u^* & u^* < 0.3 \\ 0 & u^* \geq 0.3 \end{cases}$$

$$u^* = \frac{u}{\sqrt{gH}}$$

In the above equation, the parameter $C_F$ is equal to 6.0 [18].

Therefore:

$$K_{ef} = \frac{m_c c_p T_o S^{1/2}AH_{ef}^{1/2}}{C_{\text{con}} C_F(1 + C_u)QPH}$$

Note that the HRR is proportional to the mass flow rate by Eq. (4):

$$\dot{Q} = m_c c_p \Delta T_{ad}$$

Therefore, the parameter $K_{ef}$ can be expressed as:

$$K_{ef} = \frac{\rho_c c_p T_o S^{1/2}AH_{ef}^{1/2}}{C_{\text{con}} C_F \Delta T_{ad}} PH(1 + C_u)$$

The correlation can be simplified to:

$$K_{ef} = \frac{155AH_{ef}^{1/2}}{(1 + C_u)PH}$$

### 2.3.3 Heat transfer coefficient
The thermal resistance between the flame and the surface area of the wall that is within the flame zone, $A_w$, is considered to be negligible as compared to the thermal resistance of the wall or lining itself. Therefore, heat flux through a surface can be estimated using:

$$\dot{Q}_k = h_k A_w (T - T_0)$$  \hspace{1cm} (19)

The conductive heat transfer coefficient before thermal penetration, $\delta < \delta_p$, becomes:

$$h_k = \frac{k \rho c}{\pi t}$$  \hspace{1cm} (20)

and after thermal penetration, $\delta > \delta_p$, is:

$$h_k = \frac{k}{\delta}$$  \hspace{1cm} (21)

The penetration depth is calculated using:

$$\delta_p = 2\sqrt{at}$$  \hspace{1cm} (22)

where approximately 15% of the excess temperature has reached the rear boundary.

This method is flexible as it can factor in various quantities of wall or lining materials. For tunnel structures consisting of different materials, e.g. an asphalt floor and concrete walls and ceiling, the heat transfer coefficient can be expressed as:

$$h_k = \sum_i (h_{k,i} X_i)$$  \hspace{1cm} (23)

where $X_i$ is the percentage of area of the $i$th material in the total exposed area, i.e. $A_i/A_w$.

For a given tunnel area consisting of several layers, the parameter $h_k$ can be estimated using the following equation:

$$h_k = 1 / \sum_j (1 / h_{k,j})$$  \hspace{1cm} (24)

It should be noted that only the layers in which thermal penetration has taken place ($\delta > \delta_p$) should be considered. Consequently, the penetration depth, $\delta_p$, should be checked during the calculation process.

For example, prior to the penetration of the first layer of a two-layer wall, the heat transfer coefficient can be estimated using:

$$h_t = \frac{1}{\sum_{i} 1 / h_{k,i}} = \frac{1}{1 / h_{k,1}(t)}$$  \hspace{1cm} (25)

After the penetration of the first layer, it can be estimated using:
\[ h_i = \frac{1}{1/h_{i1}(\delta_i) + 1/h_{i2}(t - t_{p,i})} \]  \tag{26} 

where the penetration depth, \( \delta_i \), and the penetration time, \( t_{p,i} \), of the first layer can be estimated using:

\[ \delta_i = 2\sqrt{a_i t_{p,i}}, \quad t_{p,i} = \frac{\delta_i^2}{4a_i} \]  \tag{27} 

and the heat transfer coefficient for the second layer is:

\[ h_{k2}(t - t_{p,i}) = \sqrt{\frac{k \rho c}{\pi(t - t_{p,i})}} \]  \tag{28} 

Note that for the second layer, the effective time after the heat has been transferred into this layer is used to calculate the heat transfer coefficient.

If there are other objects in the tunnel that could potentially absorb a large amount of heat, e.g. a train body that is on fire, the heat loss to these objects should also be taken into account. The heat transfer coefficient can be estimated using:

\[ h_k = \sum_i (h_{k,i} X_i) + \frac{A_{\text{object}}}{A_w} h_{k,\text{object}} \]  \tag{29} 

The above equations for estimating heat transfer coefficients are based on the assumption that the gas temperature (i.e. surface temperature in this model) is a constant during the heat transfer process. In reality, it is more likely that the gas temperature increases rapidly with an increasing HRR in the early stages, and then increases slowly in the steady-burning period. There is a need to correct the time, as without time correction the heat transfer coefficient during this process is generally underestimated – a consideration that is further discussed in Section 2.3.5.

### 2.3.4 Estimation of mass flow rate

If the HRR is not known, the fire is assumed to be fuel-rich (or ventilation-controlled). Thus, the mass flow rate is the amount of fresh air introduced into the tunnel.

For forced ventilation, the mass flow rate is estimated using:

\[ \dot{m}_{g,fv} = \rho_v u_v A \]  \tag{30} 

For natural ventilation \((u_v = 0)\), the total amount of fresh air flowing into the tunnel from both portals can be estimated using the following equation, assuming negligible stratification at the tunnel portals:

\[ \dot{m}_{g,nv} = 2C_m A\sqrt{H} \]  \tag{31} 

where the factor, \( C_m \), is \([19]:\)
\[ C_m = \frac{2}{3} C_d \sqrt[3]{\frac{2g \rho_o (\rho_o - \rho)}{1 + (\rho_o / \rho)^{1/3}}} \]  

where \( \rho \) is the smoke density at the tunnel portals (or at smoke front).

![Graph showing the factor \( C_m \) as a function of gas temperature.](image)

**Figure 5**  The factor \( C_m \) as a function of gas temperature.

The factor \( C_m \) is a function of gas temperature (see Figure 5). The temperature at a tunnel portal strongly depends on the tunnel length and fire location, and is usually in the range of 50-200°C for tunnels that are 500-2000 m long. The corresponding \( C_m \) factors are 0.26 and 0.48 respectively. However, in order to err on the side of caution, a value of 0.5 has been used. If more detailed information is available, a better prediction is feasible, and likely preferable.

A simple correlation can be used to estimate the total amount of fresh air introduced into the tunnel by forced or natural ventilation, \( \dot{m}_g \):

\[ \dot{m}_g = \max(A\sqrt{H_r \rho_o u_o A}) \]  

Combining the above equation with Eq. (8) shows that the tunnel cross-sectional area can be eliminated in the numerator of the right-hand term of Eq. (8). Note that it is assumed that the fire is ventilation-controlled when obtaining the correlation for mass flow rate. For smaller fires, the mass flow rate is lower, but the radiation area could also be smaller. Therefore, it is reasonable to assume that the ratio of the radiation heat loss to the mass flow rate of a smaller fire can still be expressed in this way.

### 2.3.5 Time correction

Note that the above equation is valid when a continuous flame impinges on the ceiling of the tunnel and the gas temperature is very high (\( > 800^\circ\text{C} \)). However, fires generally have a longer growth period. In order to calculate the full expression of the gas temperature as a function of time, it is often necessary to compensate for the time in the growth period.
Before the continuous flame impinges on the ceiling, it may be assumed that the heat loss to the boundary, along with the limit for adiabatic flame temperature, do not significantly influence the maximum ceiling gas temperature, and thus the previous models apply (see Eqs. (1) and (2)). The point of distinction is estimated to be above the excess gas temperature of 800°C by comparison between test data and calculated values. Before this value is obtained, calculated gas temperature values do not deviate significantly from the previous MT model. This also fits well with the representative flame temperature value chosen for continuous flames in open fires. The scientific explanation for this is that, before this temperature is obtained, the influence of surrounding structures on changes in gas temperature is considered to be negligible, i.e. the time delay is relatively small. When the continuous flames impinges on the ceiling, the new MT model begins to apply. For large fires in a large scale a higher value can be used, but doing so does not affect the final results to any significant degree.

If a design fire is known, according to the previous MT model, the time corresponding to a temperature of 800 °C, \( t_{01} \), can be obtained, as the corresponding HRR can be estimated by the following equation:

\[
\hat{Q}(t) = \max(H_{ef}^{3/2} \left( \frac{\Delta T}{17.5} \right)^{3/2}, u_f b_{ef}^{1/3} H_{ef}^{5/3})
\]  

(34)

where \( \Delta T_c \) is the excess gas temperature at the point of distinction, and time, \( t_{01} \), is the point in time at which 800°C is obtained.

According to the new MT model, time, \( t_{02} \), indicates how long it takes for the gas temperature (surface temperature) to reach 800°C if the structure is surrounded by large flames from the beginning. The corrected time used in the new MT model, i.e. Eq. (8), is therefore \( (t + t_{02} - t_{01}) \). The heat transfer coefficient is then obtained by:

\[
h_k = \frac{k \rho c}{\pi t_c} = \sqrt{\frac{k \rho c}{\pi(t + t_{02} - t_{01})}}
\]  

(35)

To estimate the time, \( t_{02} \), a modified flame temperature is defined as follows:

\[
\Delta T'_{ad} = \Delta T_{ad} - \Phi \varepsilon \sigma A(T^4 - T^4_o) / \dot{m}_g c_p
\]  

(36)

The time \( t_{02} \) is dependent on the modified flame temperature and thus cannot be directly calculated. However, it has been noted that the modified flame temperature varies very little with time. Therefore, by considering it to be a constant, a rough estimate of \( t_{02} \) is feasible. For a tunnel structure that is covered by a single material, the time, \( t_{02} \), can be estimated using Eqs. (8), (20), and (36):

\[
t_{02} = \frac{k \rho c}{\pi K_{ef}^2} \left( \frac{\Delta T}{\Delta T'_{ad} - \Delta T} \right)^2
\]  

(37)

For a tunnel structure that is covered by \( N \) different materials, the time, \( t_{02} \), can be simply estimated using:

\[
t_{02} = \left[ \sum_{i}^N \left( X_i \sqrt{\frac{(k \rho c)_i}{\pi}} \right) \right]^2 \frac{1}{K_{ef}^2} \left( \frac{\Delta T}{\Delta T'_{ad} - \Delta T} \right)^2
\]  

(38)
If there are objects within the tunnel, the time can be estimated using:

\[ t_{o2} = \sum_{i} \left( X_i \sqrt{\frac{(k_i \rho c)}{\pi}} \right) + \frac{A_{\text{object}}}{A_w} \left( \frac{(k_i \rho c)}{\pi} \right)^2 \frac{1}{K_{ef} \left( \Delta T - \Delta T_{ad} \right)^2} \]  \quad (39)

Here \( \Delta T \) was set to 800°C in order to estimate the point of distinction. Based on a series of calculations that were conducted in order to estimate the time, \( t_{o2} \), it was found that a value of 1500°C can be used for the modified flame temperature, \( \Delta T_{ad} \). However, a constant modified flame temperature is not recommended when performing general calculations of gas temperatures, as in reality the value generally decreases with time.

Numerical analysis of typical heat transfer processes in different tunnel structures has been conducted, and the results show that the estimated heat transfer coefficients based on the corrected time are close to the numerical values; it was also found that, without time correction, lower values are predicted, as was expected. It should be noted that the time-correction method proposed here has been developed for fast-growing fires, and may not be suitable for a fire that grows very slowly.

The heat transfer model assumes that the tunnel structure always absorbs heat. However, during the decay period when the gas temperature is lower than that of the tunnel structure, it may instead release heat. This process has not been considered here. However, as the decay process is here considered to be of less importance, the primary concern of this report is the heating process. Thus, the uncertainty in the decay process has been deemed to be insignificant.

### 2.4 New MT model II

To distinguish the model presented in this section from the previous MT model, it is termed ‘MT Model II’.

#### 2.4.1 Correlation

Note that the energy equation for the control volume (Eq. (5)) can also be expressed as:

\[ \dot{m}_g c_p (T'_{ad} - T) = \dot{q}_w^* A_w \]  \quad (40)

where the modified flame temperature is defined according to Eq. (36).

For a structure that is directly exposed to a high heat flux in a large tunnel fire – as much as 400 kW/m² – it may be assumed that the thermal resistance between the flame and the tunnel structure is infinitely small (as is discussed above). In other words, the gas temperature can be assumed to be the same as the structure’s surface temperature. Therefore, the above equation can be rewritten as follows:

\[ \dot{q}_w^* = -k \frac{dT}{dz} \bigg|_{z=0} = K_{ef} (T'_{ad} - T) \]  \quad (41)

By analogy, the above equation can be regarded as the first boundary condition for the thermal diffusion equation. Similar analogy methods were used by Wickström [20] to
obtain solid surface temperatures in room fires and also by Ingason et al [21] to obtain the solution of solid temperatures with ‘the fourth boundary condition’. The thermal diffusion equation can be expressed as follows:

\[ \rho c \frac{\partial T}{\partial t} = \frac{\partial}{\partial z} (k \frac{\partial T}{\partial z}) \] (42)

For a fixed modified adiabatic flame temperature, \( \Delta T'_{ad} \), and constant thermal properties for an infinitely thick material, the analytical solution for the gas temperature (or surface temperature), \( T \), is expressed as follows:

\[ T = T_o + \Delta T'_{ad} [1 - \exp(\beta^2) \text{erfc}(\beta)] \] (43)

where

\[ \beta = K_{ef} \sqrt{\frac{t}{k \rho c}} \]

The analytical solution is for a constant temperature, \( T'_{ad} \), (assumed to be equivalent to gas temperature). However, in order to better predict the temperature, the use of time-varying \( T'_{ad} \) is recommended.

In contrast to the new MT model I, Model II does not require that the heat transfer coefficient be calculated. Instead, the parameter \( \beta \) is used to factor in the properties of the tunnel structure.

It should be kept in mind, however, that this model does not work for tunnel structures consisting of several layers of different materials – more precisely, it does not work when the heat penetrates the second layer of a multi-layer structure. In such a scenario, MT model I is preferable.

2.4.2 Complex boundaries

It should be noted that the equations presented in Section 2.4.1 only apply to tunnel structures consisting of one material. Their applications could, however, extend to a tunnel consisting of \( N \) surfaces of different materials if the function is linearised. The heat transfer processes in the \( N \) different materials may be assumed to be independent of one another. For tunnel structures with surfaces covered by \( N \) different materials, such as various road surfaces and lining materials, the equation for the lumped surface heat flux (with regard to total surface area, \( A \)) is:

\[ q_{s}^* = \sum_{i}^{N} (X_i q_{w.i}) = -\sum_{i}^{N} (X_i k \frac{dT_{i,i}}{dz} \bigg|_{z=0}) = K_{ef} (T'_{ad} - T) \] (44)

where \( X_i \) is the percentage of area of the \( i \)th material that lies within the area that is exposed to fire.

The heat fluxes on the \( N \) surfaces are linearly correlated. The equation for a structure that is fully covered by the \( i \)th material is as follows:
\[ \dot{Q}_{w,d}^* = -k_i \frac{dT_{ad}'}{dz} \bigg|_{z=0} = K_{ef} (T_{ad}' - T_i) \] (45)

The \( i \)th temperature is therefore:

\[ T_i = T_o + \Delta T_{ad}' [1 - \exp(\beta_i^2) \text{erfc}(\beta_i)] \] (46)

and

\[ \beta_i = K_{ef} \sqrt{\frac{t}{(k \rho c)}} \] (47)

Assuming that the modified adiabatic flame temperature is similar for all of the materials, we have:

\[ \dot{Q}_{w}^* = \sum_{i}^{N} [X_i K_{ef} (T_{ad}' - T_i)] = K_{ef} [T_{ad}' - \sum_{i}^{N} (X_i T_i)] \] (48)

Note that:

\[ \dot{Q}_{w}^* = K_{ef} (T_{ad}' - T) \] (49)

Therefore, the lumped temperature is:

\[ T = \sum_{i}^{N} (X_i T_i) \] (50)

The temperatures are thus linearly correlated. Note that, in order to obtain the final solution, the temperature for each material is obtained by first assuming that the tunnel is fully lined with a single material. Thereafter, the temperatures are lumped according to the surface area.

This shows that it is possible to use MT Model II for a tunnel that is lined with several different materials, e.g. a tunnel with concrete walls, an insulated ceiling, and an asphalt floor. However, it is not possible to use MT Model II for a structure that consists of varying materials below the same surface. This is, however, possible using MT Model I.

### 2.4.3 Time correction

The parameter \( \beta \) is defined based on the corrected time. As with MT Model I, corrected time is expressed as:

\[ t_c = t + t_{o2} - t_{o1} \]

To perform the time correction, a parameter is first defined:
\[ \phi = 1 - \exp(\beta^2) \text{erfc}(\beta) \]  
(51)

The above function is plotted in Figure 6.

For a tunnel surface structure that is lined with a single material, the temperature is:

\[ T_g = T_o + \Delta T'_{ad} \phi \]  
(52)

Using Eqs. (8), (20), and (51), the time, \( t_{02} \), could be simply estimated using:

\[ t_{02} = \frac{k \rho c \beta^2}{K_{ef}^2} \]  
(53)

The parameter \( \beta \) can be obtained using the parameter \( \phi \), according to the correction curve in Figure 6.

For a tunnel structure that consists of surfaces covered by several different materials, it is difficult to directly calculate \( t_{02} \). Instead, a direct calculation from time=0 can be made, and the time \( t_{02} \) can be known for a given \( T_g \), e.g. \( T_g = 800 \, ^\circ C \).

![Figure 6](image_url)

**Figure 6** The function \( \phi = 1 - \exp(\beta^2) \text{erfc}(\beta) \).

Thus, it is more difficult to perform the time correction for Model II as compared to Model I.

## 2.5 Limit for \( K_{ef} \)

The energy equation can be written in the following form:

\[ q_w = -k \frac{dT'}{dz} \bigg|_{z=0} = K_{ef} (T'_{ad} - T) \]  
(54)
where

\[ K_{ef} = \frac{\dot{m}_g c_p}{A_w} \]

Note that the above equation is very similar to the following equation:

\[ q'_w = -k \frac{dT_s}{dz} \bigg|_{z=0} = h(T'_s - T) \]  

(55)

where \( h \) is a lumped heat transfer coefficient.

In other words, the coefficient \( K_{ef} \) is comparable to the coefficient of the total heat transfer from the modified adiabatic flame for diffusion flames to the surface. In reality, the measured maximum gas temperature in tunnel fires are generally around 1350°C, and this value can be used in place of the adiabatic flame temperature in order to estimate the largest heat transfer coefficient. Therefore, we may estimate the coefficient as follows:

\[ h = h_c + \varepsilon \sigma (T'_{ad} + T)(T'_{ad}^2 + T^2) < 4\sigma T'_{ad} + h_c \approx 980 \]  

(56)

Therefore:

\[ K_{ef} < 980 \]

In typical tunnel fire scenarios, the parameter is mostly not greater than 200, which is below the limit.

### 2.6 Considerations regarding travelling fires

A travelling fire may occur if enough fuels are available along the length of the tunnel, and can involve a fire spreading from one part of a vehicle or object to another. Consequently, the location of the core fire 'travels', and the fire has thus become a travelling fire. The thermal impact of a travelling fire on the structure of a tunnel is very interesting; the locations where the structure experiences most severe effects may not be in the area directly above the ignition site of the core fire but downstream of it, as the cumulative effects on the structure may vary.

Gas temperatures at a location far downstream of the core fire may, however, remain at a relatively low level for a relatively long time prior to the arrival of the flame front, meaning that some of the assumptions made for the models presented in this work are not valid; thus, for this scenario, the use of the new models is not recommended.

For reinforced concrete, the main concern is that the temperature of the steel bars within the concrete does not exceed a certain value, in order to avoid a reduction in strength.

The highest gas temperatures are generally obtained close to the upstream side of the fuels, as the oxygen level is generally high in such areas. If the fire continues to move downstream, and is thus a travelling fire, the location at which the maximum temperature occurs can also move. The highest temperature during the entire incident can occur when the fire moves to the fuel that is furthest from the initial fire site.
The temperature of the steel bars may not be greatly affected by the maximum temperature, but generally is by the duration of the fire. For a travelling fire, a holistic analysis that examines both smoke flow and heat conduction inside the concrete needs to be conducted in order to calculate the maximum temperature of the steel bars. In other words, fire flows that is induced by a travelling fire together with heat conduction should be simulated.

2.7 Applications of the models to compartment fire scenarios

The two new models presented above can also be used to estimate gas temperatures in a fully developed room fire. The uncertainty of the models in such an application is lower than when used for a tunnel fire scenario due to the need for fewer assumptions to be made in order to obtain correlations. For the purposes of validation, this process will be described in brief.

For a fully developed compartment fire, the mass flow rate through any openings can be estimated using the following equation:

\[ \dot{m}_g = 0.5A_w \sqrt{H_o} \]  

(57)

Comparing this with Eq. (33) indicates that, for a compartment with an opening of the same size as the cross-section of a tunnel, the mass flow rate is lower.

The surface area of the wall that is within the flame zone, \( A_w \), is equal to the total internal area minus the areas of the openings.

The parameter \( K_{ef} \) is therefore:

\[ K_{ef} = \frac{\dot{m}_g c_p}{A_w} = \frac{500A_w \sqrt{H_o}}{A_w} \]  

(58)

Thus, MT Models I (Eqs. (8) or (10)) and II (Eq. (43)) can still be used for a fully developed compartment fire. The only differences in terms of the scenarios are the different expressions of the parameter \( K_{ef} \) and mass flow rate, \( \dot{m}_g \).

The models are used below to provide a point of comparison with data obtained during compartment fire testing. However, the models are only applicable to compartment fires with high temperatures, i.e. over 800°C. For lower temperatures, several assumptions relating to e.g. flow rate need to be revised. This issue is not discussed further in this report.

2.8 Scaling maximum ceiling gas temperature

It has been observed that maximum ceiling gas temperatures are generally lower in model-scale tests. In the following, plausible reasons for this phenomenon are given.

The radiation heat flux from flames can be estimated using the following equation:
\[ \dot{Q}_r = A_r \dot{q}^* = A_r \varepsilon \sigma T_\#^4 \]  
(59)

According to Froude scaling,
\[ \dot{Q}_r = X \dot{Q} \propto I^{5/2} \]  
(60)

It is well-established that the radiation fraction of both enclosure and open fires is approximately 30%. This value may vary somewhat, but heat loss by radiation is limited. Therefore, the above correlation applies to a great extent.

Furthermore, it should be noted that the flame shapes are generally similar, based on comparisons of the ceiling flame lengths in tests on different scales. In other words, \( A \propto I^2 \), according to scaling theory. Therefore:
\[ \varepsilon \sigma T_\#^4 \propto I^{1/2} \]  
(61)

According to Li and Hertzberg [22], the incident heat flux from flames and hot gases in enclosures scaled very well on three different scales (1:1, 1:2, and 1:3.5), indicating that the above correlation applies.

For large fires in tunnels and enclosures, soot radiation is a central element of total radiation. Effective soot emissivity can be estimated using:
\[ \varepsilon = 1 - \exp(-\kappa_m L_{mn}) \]  
(62)

where the mean absorption coefficient, \( \kappa_m \), is [23]:
\[ \kappa_m = 3.72 \frac{C_o}{C_2} X C,T \]  
(63)

and the mean beam length, \( L_{mn} \), is:
\[ L_{mn} = 3.6 \frac{V_b}{A_b} \]  
(64)

The equation for the mean absorption coefficient can also be expressed as:
\[ \kappa_m = 3.72 \frac{C_o}{C_2 \rho_s} \rho T = 3.72 \frac{C_o}{C_2 \rho_s} \frac{P_o}{R} \]  
(65)

where \( C_o \) is a constant that varies between 2 and 6 dependent on the refractive index (a value of 4 is applied in PRS), \( C_2 \) is Planck’s second constant \( (1.4388 \times 10^2 \text{m} \cdot \text{K}) \) and \( R \) is gas constant.

Thus, the mean absorption coefficient depends primarily on the mass fraction of soot in the smoke. If the soot yield is the same across all of the scales and the temperatures are scaled reasonably well, the mean absorption coefficient should be reasonably similar.

Effective soot emissivity thus depends largely on mean beam length. To scale temperature well, the effective soot emissivity should be scaled as \( \frac{1}{2} \) the power of the
length scale. The apparent discrepancy indicates the difficulty inherent in scaling the radiation intensity of flames and hot gases. Another problem in scaling radiation comes from the scaling of wall emissivity which, according to scaling theory, should also be scales as \( \frac{1}{2} \) the power of the length scale [21]. This, however, is difficult to execute, and is thus generally ignored in model scales; the discrepancy indicates the difficulty inherent in scaling the radiation of walls.

Given that there appears to be a difference in emissivity between scales, the following correlation for temperature on the model scale can be proposed:

\[
T_M = T_F \left( \frac{\varepsilon_M}{\varepsilon_F} \right)^{-1/4} \left( \frac{l_M}{l_F} \right)^{1/8}
\]

This equation correlates gas temperatures between different scales. If the emissivity on both scales is unity, the temperature on the model scale will be lower.

The above analysis is based on the assumption that the radiation fraction has the same value on different scales. If this is not the case, the equation needs to be modified to:

\[
T_M = T_F \left( \frac{\varepsilon_M}{\varepsilon_F} \right)^{-1/4} \left( \frac{X_{r,M}}{X_{r,F}} \right)^{1/4} \left( \frac{l_M}{l_F} \right)^{1/8}
\]

### 2.9 Location of maximum flame temperature

There are two parameters that influence the possible maximum flame temperature: local oxygen concentration, and heat loss to the tunnel structure.

It should be noted that a higher oxygen concentration leads to more intense combustion, resulting in a higher gas temperature. If diluted gases are involved in combustion, a lower gas temperature will occur due to the heat loss to the structure during the transport of the diluted gases to a new location. Therefore, it can be expected that the maximum ceiling gas temperature occurs in the vicinity of the fire source, at the point at which fresh air flow can be directly entrained, rather than downstream of the fire where any fresh air has been diluted and the gases in the upper layer consist of a significant quantity of combustion products. This has been confirmed by many fire tests in tunnels [24].

A simple comparison is made here, which assumes no heat loss to boundaries (tunnel surfaces). For flames in the vicinity of a fire, where fresh air is available, the temperature can be calculated using:

\[
\Delta T_g = C_{r,f} \frac{\Delta H}{c_p r} Y_{O_2}
\]

In a tunnel with longitudinal ventilation (see Figure 7), the oxygen in the flame at a certain distance downstream is considered to be vitiated or diluted by the combustion products upstream. Assuming that this percentage is \( Y_t \), the gas temperature in this flame volume is:

\[
\Delta T_g' = Y_t \Delta T_{g,i} + (1 - Y_t) C_{r,f} Y_{O_2} \frac{\Delta H}{r c_p}
\]
Note that the combustion products come from the previous combustion zone (upstream), and thus in reality the gas temperature decreases when it reaches this location due to heat loss and entrainment, i.e. $\Delta T_{g,1} < \Delta T_g$. Thus:

$$\Delta T'_g < \Delta T_g$$

(70)

For a travelling fire in a tunnel with longitudinal ventilation (see Figure 7), it is likely that the highest gas temperature will be registered at the point marked ‘LN’. This will, however, occur only when the fuel, $N$, is exposed to ventilation flows that are not vitiated, i.e. the fuel that lies upstream of $N$ has been consumed. Furthermore, sufficient fuel needs to be available at this moment.

Figure 7  The location of the maximum ceiling gas temperature in a tunnel with high ventilation conditions.
3 Verification of the models

The previous MT model was validated using a large quantity of model- and full-scale data [1, 2]. As discussed above, however, the maximum excess gas temperature was set to 1350°C for large fires for the sake of simplicity, and the approach does not factor in the effects of fire size, ventilation conditions, and tunnel structure for large fires.

Comparisons between the previous and new MT models are provided in Figure 8 and Figure 9, together with test data from the Runehamar tunnel fire tests 1 and 4 [12]. The primary differences are for temperatures above 800°C. Above this value, the temperature grows more slowly, which is well predicted by the new MT models, but not the previous MT model. Furthermore, the previous MT model is relatively conservative, giving predictions that can be regarded as upper limits for large tunnel fires.

**Figure 8**  A comparison of the previous and new models for Runehamar test 1.

**Figure 9**  A comparison of the previous and new MT models for Runehamar test 4.
3.1 Full-scale tunnel fires

3.1.1 The Runehamar tunnel fire tests, 2003

The maximum ceiling gas temperatures were calculated based on the HRR measured during Runehamar tunnel Fire Tests 1-4 [12, 13].

The fire tests were carried out using HGV (Heavy Goods Vehicle) mock-ups in the Runehamar tunnel, Norway, in 2003. In the vicinity of the fire sources, the tunnel walls and ceiling were covered with Promatec T boards with a thermal inertia of approximately $1.9 \times 10^5 \text{W}^2 \text{sm}^{-4} \text{K}^{-2}$; the floor is made of asphalt.

The HRR curves and average flow velocities were used to produce time-temperature curves, which are shown in Figure 10, Figure 11, Figure 12 and Figure 13. It should be noted that, in Test 1, all of the measurements worked well during the entire test, but in Tests 2 and 3, the thermocouple located 10 m downstream of the fire worked well during the early stages of the test, but failed after around 10 min. Therefore, the maximum gas temperatures are not known after that point in time.

The calculated gas temperature curves correlate well with the measured temperature curves and the maximum gas temperatures are well predicted. It can also be observed that the measured temperature rose much more rapidly during Test 3, which was likely the result of a measurement error at an early stage of the test. It should be noted that the fire grew extremely rapidly in this test with polyurethane foams, as compared to the others. Furthermore, the measuring of the HRR was performed 463 m downstream of the fire site. Therefore, during the rapidly growing period, the transportation of the combustion products to the measurement station could cause a relatively larger error as compared to the other tests. In addition, the fire radius is considered to be a constant in the calculation, i.e. the largest fire radius is used, but varies with fire size. A smaller fire radius results in higher gas temperatures, according to the previous model.

![Figure 10](image-url)  

*Figure 10 A comparison of the calculated transient maximum temperature beneath the ceiling and the data obtained during Runehamar Test 1.*
Figure 11  A comparison of the calculated transient maximum temperature beneath the ceiling and the data obtained during Runehamar Test 2.

Figure 12  A comparison of the calculated transient maximum temperature beneath the ceiling and the data obtained during Runehamar Test 3.
Figure 13  A comparison of the calculated transient maximum temperature beneath the ceiling and the data obtained during Runehamar Test 4.

3.1.2 Brunsberg tunnel fire tests, 2011

Three full-scale carriage fire tests were performed in the old Brunsberg tunnel, located between Kil and Arvika in western Sweden [17, 25]. To protect the concrete lining of the tunnel (approximately 10 cm of reinforced shotcrete), the tunnel ceiling and around 60% of the tunnel walls between a point 15 m upstream and 35 m downstream of the fire were covered with 50 mm thick insulation material. The insulation consisted of U Protect Wired Mat 2.0 with a density of 55 kg/m$^3$ and a thermal inertia, $k \rho c_v$ of $5 \times 10^3$ W$^2$sm$^{-4}$K$^{-2}$. The train body was made of thin steel.

The maximum ceiling gas temperatures were calculated using MT Model I, based on HRR values measured during Test 3. MT Model II was not used, as it is not suitable for scenarios involving a train body in a tunnel, as is discussed above. The HRR was obtained using the oxygen consumption method, and the ventilation velocity was around 2 m/s. A comparison with test data is given in Figure 14. The entire testing period evinced a strong agreement with calculations. When gas temperatures are high during testing, the flow is quite unsteady and thus gas temperatures show a scattering trend.
3.2 Model-scale tunnel fires

In this section, model-scale test data is compared to the values obtained using the calculation model.

3.2.1 The 1:10 model-scale tests

The model tunnels used in the model-scale tests were 12.5 m long (14.5 m if the fan section is included), 0.6 m high, and 1 m (Tunnel A) and 0.6 m (Tunnel B) wide [18].

The model tunnels were constructed using 4 cm thick non-combustible Calcium silicate boards (Promatec L), with the exception of the lower part (50%) of one side of the tunnel, which was covered with a fire-resistant window panel, mounted on steel frames. The Promatec L has a conductivity of 0.083 W/m · K, a density of 450 kg/m³, a heat capacity of 1130 J/kg · K, and a thermal inertia, $k \rho c$, of approximately $4.2 \times 10^4$ W·m²·K⁻². The material was chosen according to the scaling theory proposed by Li and Hertzberg [22] in order to simulate the concrete and rock that are used in tunnels (or a mixture of dense and medium-dense concrete).

Test 705 in Tunnel B was selected for comparison against the values obtained using the calculation models, the results of which are shown in Figure 15. During Test 705, a longitudinal flow occurred along the tunnel, and the HRR increased incrementally, from 16 to 32, 63, 158, 237, 300, and 395 kW in time increments of 5, 5, 5, 3, 3, 3, and 3 min, respectively. The velocity of the flow was initially 0.75 m/s, but had fallen to around 0.5 m/s by the end of the test. It should be noted that changes made to the fire size at 24 min caused a slight fluctuation in both the fire size and gas temperature for a short period.

Very strong agreement was found with the measured temperatures shown in Figure 15. MT Model II produces a slightly lower temperature after 20 min. MT Model I appears to yield more realistic results.

Figure 14  A comparison of the calculated transient maximum temperature beneath the ceiling and the data obtained during Brunsberg Test 3.
3.2.2 The 1:23 model-scale tests

The tunnels used in the 1:23 scale tests were 10 m long, 0.4 m wide and had two heights: 0.2 m and 0.3 m [26]. The model was constructed using non-combustible Calcium silicate boards (Promatect H in these tests) with a thickness of 15 mm [26]. The manufacturer of the boards provides the following technical data: their density is 870 kg/m$^3$, heat capacity is 1130 J/kg K, heat conduction is 0.175 W/m K, and thermal inertia, $k \rho c$, is approximately $1.7 \times 10^5$ Wsm$^{-2}$K$^{-2}$. Tests 4 and 5 were selected for comparison against the values obtained using the calculation models.

The comparison of the values obtained using the calculation model to the data recorded during Test 4 is shown in Figure 16. In Test 4, with a tunnel height of 0.3 m, the maximum HRR was roughly 280 kW and the velocity of the longitudinal flow was approximately 0.7 m/s.
Figure 16  Comparison of the calculated transient maximum temperature beneath the ceiling and the data obtained during the 1:23 model-scale Test 4.

The comparison of the values obtained using the calculation model to the data recorded during Test 5 is shown in Figure 17. In Test 5, with a tunnel height of 0.3 m, the maximum HRR was roughly 140 kW and the velocity of the longitudinal flow was approximately 0.52 m/s.

Figure 17  Comparison of the calculated transient maximum temperature beneath the ceiling and the data obtained during the 1:23 model-scale Test 5.
3.3 Compartment fires

As is discussed in Section 2.7, the new models can also be used for a fully developed compartment fire, aside from the different expressions of the parameter $K_{ef}$ and the mass flow rate, $\dot{m}_g$. In the following, data from two series of compartment fire tests carried out at SP Fire Research by Li and Hertzberg [22] and Sjöström et al. [27] are compared against results obtained using MT Models I and II.

3.3.1 Li and Hertzberg’s compartment fire tests

A total of 7 tests were carried out using full- (1:1), medium- (1:2) and small-scale (1:3.5) rooms, with a primary focus on scaling the internal temperatures of the wall materials. Data from the full-scale Test 4 and medium-scale Test 5 is used in this report for comparison. The full-scale compartment was 3.6 m long, 2.4 m wide, and 2.4 m high, with a 0.8 m wide and 2 m high door. Mineral wools were used as structural materials. In both Tests 4 and 5, the fire source was placed in the corner of the compartment. The fire burned at 100 kW for 10 min, followed by 300 kW for another 10 min and 1.2 MW for 7 min. For model-scale Test 5, the compartment was geometrically scaled; the process of selecting the materials in a model scale was chosen according to the scaling method proposed by Li and Hertzberg [22].

The comparison of the values obtained using the calculation models to the test data for the full-scale Test 4 is shown in Figure 18, where T12-15 are thermocouples that were located below the ceiling. The predictions of the two MT models were very close to the highest measured gas temperatures.

![Figure 18](image)

Figure 18 Comparison of the calculated transient maximum temperature for the full-scale compartment fire Test 4 of Li and Hertzberg’s compartment fire test series.

The comparison of the values obtained using the calculation models to the test data for the medium-scale Test 5 is shown in Figure 19, where T12-15 are thermocouples that were located below the ceiling. The predictions of the two MT models were slightly higher than the measured maximum ceiling gas temperatures.
3.3.2 Sjöström et al.’s compartment fire tests

A total of 12 tests were carried out by Sjöström et al. [27] in a 2.7 m long, 1.8 m wide, and 1.8 m high compartment constructed of various materials. The primary purpose of the tests was to provide data to validate compartment fire models.

In this report, data from Test A1 [27] is used for comparison. A fire of 1 MW was placed in the centre of the room. The entire structure was made of lightweight concrete. Before the test, the measured density was 760 kg/m$^3$, moisture content was 39%, heat capacity was 851 J/(kg · K), and the thermal conductivity was 0.33 W/(m · K). After the test, the heat capacity was 835 J/(kg · K) and the thermal conductivity was 0.166 W/(m · K), while the density and moisture were not reported. Assuming that all of the moisture content evaporated over the course of the test, the density at the end of the test can be estimated as being 547 kg/m$^3$.

A comparison of the calculated temperatures to the test data collected by the five ceiling-mounted plate thermometers in Test A1 is shown in Figure 20. It should be noted that the initial physical and thermal properties relating to the presence of moisture inside the concrete were used in the calculations. Both models accurately predicted the highest measured values, particularly MT Model I, the values of which correlate very well during the initial 25 min, after which there is a slight tendency towards underestimation. This is likely due to the changes that occurred in the structure in terms of physical and thermal properties over the course of the test, given that the properties of the lightweight concrete that were recorded before the test (with e.g. a high moisture content) were used in calculations.
3.4 Summary

The new MT model I and II produced very similar gas temperature values. In general, the gas temperatures predicted by MT Model II are slightly lower than those of MT Model I.

The comparisons of the calculated gas temperatures and the measured values in full- and model-scale tunnel fire tests performed in this chapter indicate that the models presented in this report are suitable for estimating maximum gas temperatures in tunnel fires.

A strong correlation between the measured data and the predictions of temperature values in compartment fires also indicates that the models presented here are realistic, and that the models can also be used to estimate maximum gas temperatures in compartment fires.

In comparison to MT Model II, MT Model I has a greater degree of flexibility with regard to structural materials. Furthermore, it is much easier to perform the time correction when using Model I, and the model produces relatively conservative results. Therefore, it is recommended that MT Model I be used, as is done in the following analysis.
4 Parametric analysis

4.1 Thermal effect

Tunnel walls are typically made of concrete and rock, or a combination of the two (e.g. shotcrete). However, the surfaces (walls and ceilings) of some tunnels are covered in non-combustible insulation boards (Calcium silicate, magnesium oxide, cement-bonded fibreglass-reinforced lightweight concrete, etc.) of different thicknesses for fire protection. In order to conduct a parametric analysis of the various input parameters, an analysis was carried out using MT Model I. In the following, a simple case study is presented in order to compare the effect of different tunnel surfaces (boundary conditions) on maximum excess gas temperatures in large tunnel fires. A large quantity of liquid fuel is assumed to have leaked from a fuel tanker, forming a large pool in a 6 m high and 6 m wide tunnel. The fire develops immediately following an ignition, and becomes ventilation-controlled. The tunnel structure in terms of its thermal properties is thick concrete or rock.

Non-combustible insulations boards are placed on the concrete surfaces of some tunnels to protect the structure. The thicknesses of these vary, but a reasonable range is 15-60 mm. In the case study presented here, 25 mm wide and 40 mm thick Calcium silicate boards with a thermal inertia, $kpc$, of $1.9 \times 10^7 W^2 sm^{-4}K^{-2}$ are used.

Gas temperatures were calculated for four $6 \times 6$ m tunnels of different structures (see Figure 21). The resultant values were higher for more insulating materials, i.e. tunnels with a lower thermal inertia. For the example given here, thermal data for a Calcium silicate board has been used. The difference in gas temperatures between a tunnel equipped with 25 mm thick boards and one with 40 mm thick boards is relatively small. In the initial 15 min, the temperatures are identical as the thermal penetration of the insulation boards does not occur, but a difference, though small, becomes apparent after a further 15 min. The temperature difference between concrete, rock, and insulation boards can be explained by the thermal inertia of the tunnel surface. This phenomena is well known from furnace and enclosure fire testing, and has been observed in previous studies by the authors [13].

![Figure 21](image_url) **Figure 21** The effect of tunnel structure on gas temperature.

For a concrete tunnel with reinforced steel bars, the temperature at a given depth can be of interest. The temperatures at 50 mm ($z_o = 50$ mm) and 100 mm ($z_o = 100$ mm) below the concrete surface are shown in Figure 22, along with the results when a 25 mm thick...
Calcium silicate cover is added above the concrete (denoted in Figure 21 by ‘+ 25 mm’). In the following, ‘z₀’ refers to the original location of the concrete surface.

Numerical modelling of heat conduction in the tunnel structure was conducted using the gas temperatures in Figure 21 as the inputs. Figure 22 shows that the addition of 25 mm thick Calcium silicate boards to the concrete surface decreases the temperature and increases the time required for the heat to penetrate to a given depth. This demonstrates the benefits conferred by using insulation boards, which is as expected. The difference in the temperature for the steel bars at these two depths is in the order of 100-200°C, 120 min in.

![Figure 22: The effect of Calcium silicate boards on steel bar temperatures at various depths from the concrete surface/interface.](image)

However, for the sake of a fair comparison of different solutions, it should be borne in mind that, following the addition of the Calcium silicate boards to the concrete surface, the thickness of the material that covers the tunnel structure increases (i.e. there is a greater distance between the hot surface and the steel bar). Thus, the corresponding steel bar temperatures when a concrete slab of the same thickness is added to the surface are of interest. The comparison of the steel bar temperatures at 75 mm (z₀ = 50 mm) and 125 mm (z₀ = 100 mm) below the uppermost surface of the additional boards and concrete is shown in Figure 23, where it can be seen that the boards initially delay the heat transfer process as compared to concrete. For the 25 mm thick Calcium silicate boards, it takes approximately 6 min longer for a steel bar to reach e.g. 100°C as compared to the same thickness of concrete. In general, the steel bar temperature increases slightly as a result of the use of a concrete cover as compared to Calcium silicate boards, although the difference is small. At 75 mm below the surface (z₀ = 50 mm), the difference is less than 35°C; at 125 mm below the surface (z₀ = 100 mm), less than 15°C.

There are two reasons for the fact that this difference is relatively small. Firstly, the gas temperatures in the scenario in which a Calcium silicate cover is used are generally higher due to the influence of the tunnel surface on the overall gas temperature. Secondly, the steel bar temperature values shown in Figure 23 were obtained for the same distance between the surface and the location of the steel bar.
Figure 23  A comparison of the temperatures inside the concrete; with a 25 mm thick concrete cover, and with a 25 mm thick Calcium silicate board.

Using thicker Calcium silicate boards would likely result in a slightly larger difference as compared to a concrete slab of the same thickness. In the following, the analysis of the 40 mm thick Calcium silicate boards is described. The actual depths below the surface of the structure were 90 mm ($z_0 = 50$ mm) and 140 mm ($z_0 = 100$ mm), and the analysis showed that the difference resulting from the type of cover used in the temperature of a steel bar at a given depth is slightly larger for a 40 mm thick board than a 25 mm thick one. The 40 mm thick Calcium silicate cover took approximately 10 min longer to reach a temperature, e.g. 100°C, than a concrete board of the same thickness. At 90 mm below the surface ($z_0 = 50$ mm) the difference is less than 50°C, and at 140 mm below the surface ($z_0 = 100$ mm) the difference is less than 25°C (see Figure 23).

Figure 24  A comparison of the temperatures inside the concrete; with a 40 mm thick concrete cover, and with a 40 mm thick Calcium silicate cover.

25-40 mm thick Calcium silicate boards do not appear to have a strong influence on the temperature of steel bars as compared to concrete layers (or sprayed concrete) of the same thickness. One of the main reasons for this is the influence of these materials on the final gas temperatures obtained inside the tunnel for the same HRR, which is usually not taken into consideration when determining which time-temperature curve should be used in the design of the tunnel. However, it should be borne in mind that these boards are useful in
some scenarios, e.g. in a tunnel in which spalling of concrete or rock is very likely to occur, or when infrastructure is located in critical areas where a possible spalling or collapse of parts of the structure may be detrimental to the fire protection measures that are in place.

It should be noted that the constant thermal properties of both concrete and Calcium silicate boards are used in the above calculations. These can, however, vary with temperature to some extent. The previous calculations of temperatures using constant properties for Calcium silicate boards [12] have showed that simulated temperatures are generally slightly higher than measured ones. A similar trend has also been found for concrete.

In the above analyses, all of the tunnel surfaces are assumed to be covered with the same material. In reality, however, the tunnel floor is likely mostly made of asphalt or concrete, and so the difference between the calculated temperatures can be expected to be even smaller. Some uncertainties relating to the estimation of gas temperatures for different structures need, of course, to be considered.

### 4.2 Scale effect

In the following, test data from model-scale tests is discussed, and the measured and calculated maximum ceiling gas temperatures for tunnel fires are presented in Table 1. Gas temperatures on the model scale are estimated using Eq. (66), based on full-scale temperature values and emissivity values measured during model-scale testing. As a rough estimation, the maximum gas temperature obtained during the Runehamar tunnel fire tests – 1362°C – was used as the full-scale gas temperature. The emissivity values were obtained through comparison of measured gas temperatures and incident heat fluxes. The calculations show that the gas temperatures on the model scale are lower. Comparing the measured and calculated gas temperatures indicates a reasonably good agreement.

<table>
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<th>Test series</th>
<th>Scale ratio</th>
<th>$H$ (m)</th>
<th>$W$ (m)</th>
<th>$\varepsilon$</th>
<th>Measured temperature ($^\circ$C)</th>
<th>Calculated Temperature ($^\circ$C)</th>
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<td>0.4</td>
<td>0.62</td>
<td>982-1037</td>
<td>972</td>
</tr>
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<td>Point extraction [28]</td>
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<td>0.4</td>
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<td>944</td>
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<td>Ceiling jet [18]</td>
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<td>0.6</td>
<td>0.85</td>
<td>1153</td>
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<td>5</td>
<td>1.00</td>
<td>1362</td>
<td>1362^*</td>
</tr>
</tbody>
</table>

Used as a reference value for Eq. (66) for calculations of temperature on other scales.

A comparison of the measured and calculated maximum ceiling gas temperatures in compartment fires is presented in Table 2. Three different scales of test, carried out by Li and Hertzberg [22], were used for comparison. The predictions made using Eq. (66) and MT Model I correlate reasonably well with the test data, particularly the values predicted by Eq. (66). Furthermore, the temperatures at smaller scales are lower.
Table 2  A comparison of the measured and calculated maximum gas temperatures in fully developed enclosure fires.

<table>
<thead>
<tr>
<th>Scale ratio</th>
<th>$H$ (m)</th>
<th>$W$ (m)</th>
<th>$L$ (m)</th>
<th>$\varepsilon$</th>
<th>Measured temperature (°C)</th>
<th>Calculated Temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Eq. (66)</td>
<td>Model I</td>
</tr>
<tr>
<td>1:1 [22]</td>
<td>2.4</td>
<td>2.4</td>
<td>3.6</td>
<td>1.0</td>
<td>1181</td>
<td>1181*</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1167</td>
</tr>
<tr>
<td>1:2 [22]</td>
<td>1.2</td>
<td>1.2</td>
<td>1.8</td>
<td>1.0</td>
<td>1049</td>
<td>1060</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1104</td>
</tr>
<tr>
<td>1:3.5 [22]</td>
<td>0.69</td>
<td>0.69</td>
<td>1.03</td>
<td>1.0</td>
<td>978</td>
<td>970</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1053</td>
</tr>
</tbody>
</table>

*Used as a reference value for Eq. (66) for calculating temperature in other scales.

It should be noted that the fires discussed above were either large fires in tunnels or fully developed fires in enclosures—two scenarios that involve high gas temperatures of over 1000°C when in full scale. The emissivity of such fires at different scales is generally close to unity, and so emissivity is not scaled appropriately on the model scale, causing lower gas temperatures.

It can thus be concluded that scaling radiation generally results in lower gas temperatures at smaller scales.
5 Calculation examples

In the following, calculation examples are presented in order to demonstrate the process of estimating the gas temperature of a large tunnel fire using the new MT models.

5.1 Example 1: Concrete tunnel

The example described here is a 6 m wide and 6 m high concrete tunnel, with a road surface made of asphalt. Both the concrete walls and the asphalt floor are 100 mm thick, and beneath them is rock. A HGV fire follows the ultra-fast fire curve ($Q(t) = 0.19t^2$) up to a peak value of 200 MW at around 17.5 min. It continues to burn at this intensity for 20 min, then falls linearly over the course of 10 min. The ventilation velocity is assumed to be 3 m/s throughout the fire, and the emissivity is assumed to be unity. The ambient tunnel temperature is 10°C. The thermal inertia, $k\rho c$, of concrete is assumed to be $2.5 \times 10^6 W^2m^{-4}K^{-2}$, and of asphalt $1.5 \times 10^6 W^2m^{-4}K^{-2}$. The thermal diffusivity of concrete is $5.7 \times 10^{-7} m^2/s$, and of asphalt $3.8 \times 10^{-7} m^2/s$. The fire source is 10 m in length and 2.4 m in width. The maximum gas temperatures are calculated at 10 and 30 min using MT Model I (Eq. (9)).

Solution:

(1) Calculation of $K_{ef}$

\[
u^* = \frac{\nu_0}{\sqrt{gh}} = 0.39 > 0.33
\]

Therefore $C_u = 0$, according to Eq. (9). Note that the fire source could be considered to be 1 m above floor, i.e. $H_{ef} = 5$ m. Using $A = 36 m^2$ and $P = 24 m$, the parameter $K_{ef}$ is obtained from Eq. (9):

\[
K_{ef} = \frac{155AH_{ef}^{1/2}}{(1 + C_u)PH} = 87
\]

(2) Calculation of heat transfer coefficient

A time correction should be carried out. To estimate the time, $t_{01}$, the corresponding HRR can be calculated:

\[
\hat{Q}(t_{01}) = \max(H_{ef}^{5/2} \left(\frac{\Delta T}{17.5}\right)^{1/2} u_b^{1/3} H_{ef}^{5/3} \Delta T) = 49 MW.
\]

Here, the right term in the parenthesis yields a higher HRR (as $V' = 0.38 > 0.19$), and is therefore used. The parameters are as follows: $\Delta T = 800^\circ C$, $b_{f0} = 2.8$ (\(b_{f0} = \sqrt{\frac{\Delta T}{\pi}} = \sqrt{3\times10^3/\pi}\), and $u_0 = 3 m/s$. The value is obtained in kW, but presented here as MW. The corresponding time, $t_{01}$, is around 510 seconds according to the HRR curve ($\sqrt{490000/0.19}$).

The time, $t_{02}$, may be approximately estimated using the simplified new MT model I:
\[ t_{o2} = \left( \sum_{i} \left( \sqrt{\frac{(k \rho c)}{\pi} X_i} \right)^2 \frac{1}{K_{ef}^{\prime}} \frac{\Delta T}{\Delta T_{ad}^\prime - \Delta T} \right)^2 = 123 \text{ s} \]

The fraction \( X_1 \) and \( X_2 \) is 0.75 (18 m/24 m) for concrete and 0.25 (6 m/24 m) for asphalt. Further, \( k \rho c \) is \( 2.5 \times 10^6 \text{W/m}^2\text{K} \) for concrete and \( 1.5 \times 10^6 \text{W/m}^2\text{K} \) for asphalt, and \( K_{ef} \) is equal to 87. The modified adiabatic flame temperature, \( \Delta T_{ad}^\prime \), is 1500°C, and \( \Delta T \) is 800°C.

Therefore the correction time, \( t_c \), is:

\[ t_c = t + t_{o2} - t_{ol} = 213 \text{ s for 10 min (600 + 123 \text{ s) and 1413 s for 30 min (1800 + 123 \text{ s).}} \]

The thermal penetration depth can then be estimated: For concrete, it is around 0.022 m at 10 min and 0.057 m at 30 min. For asphalt, it is around 0.018 m at 10 min and 0.046 m at 30 min. All of these values are less than 0.1 m, indicating that thermal penetration has not occurred. Therefore, the heat transfer coefficient can be estimated using Eq. (23):

\[ h_k(t_c) = \sum_i (X_i h_{k,i}(t_c)) \]

where

\[ h_{k,i}(t_c) = \sqrt{\frac{(k \rho c)}{\pi t_c}} \]

The lumped heat transfer coefficient was calculated as being 58 W/m²K at 10 min, and 22 W/m²K at 30 min.

(3) Calculation of mass flow rate

The mass flow rate is:

\[ m_g = \max(A \sqrt{H}, \rho_o u_o A) = 130 \text{ kg/s} \]

where \( A = 36 \text{ m}^2, H = 6 \text{ m}, \rho_o = 1.2 \text{ kg/m}^3, \) and \( u_o = 3 \text{ m/s). The right-hand term in the parentheses yields a larger value.} \]

(4) Calculation of gas temperature

The initial focus is on the temperature at 10 min. Note that \( \varepsilon = 1 \) and \( \xi = 1. \) The temperature equation (Eq. (8)) is:

\[ T = T_o + \frac{\Delta T_{ad} - \sigma A(T^4 - T_o^4)}{1 + h_k(t_c)/K_{ef}} \]

Initially, a gas temperature of 1000°C at 10 min may be assumed (note that any temperature between 800 and 1580°C can be assumed). By using the value on the right-hand side of the above equation, the calculated excess gas temperature is 939°C, which is less than the assumed 1000°C. Therefore, the excess gas temperature should be between 939°C and 1000°C. A new value is assumed, with a very good guess being the obtained
value of 939°C. Repeating the process 2 times gives the final solution of 943°C at \( t = 10 \) min can be obtained. Similarly, the solution is 1216°C for the gas temperature at \( t = 30 \) min. Further, check the possible maximum temperature using the previous model, i.e. Eq. (12),

\[
\Delta T_{\text{previous}} = \min(17.5 \frac{\dot{Q}^{2/3}}{H_{\text{ef}}^{2/3}}, \frac{\dot{Q}}{u_0 b_0^{1/3} H_{\text{ef}}^{5/3}})
\]

giving 1100°C at \( t = 10 \) min and 1350°C at \( t = 30 \) min. Note that \( \dot{Q} \) is 68 MW and 200 MW, respectively. Therefore, the gas temperatures are 943°C at \( t = 10 \) min and 1216°C at \( t = 30 \) min according to Eq. (11).

If Model II is used, the results are 834°C at \( t = 10 \) min and 1172°C at \( t = 30 \) min. Similar results can be obtained using the two MT models, while results obtained using MT Model II are slightly lower, particularly at the beginning.

5.2 Example 2: Railway tunnel

The scenario is the same as in Example 1, except that the fire source is a train in a tunnel and so the heat loss to the train body needs to be factored in. The train body is assumed to consist of a thin steel plate of 1.5 mm thickness, below which is an insulated layer consisting of polyester foam of 30 mm thickness. The train is 4 m high, 2.9 m wide, and 200 m long. The thermal inertia, \( k \rho c \), of the foam is assumed to be \( 6 \times 10^4 \text{ W}^2\text{sm}^{-4}\text{K}^{-2} \).

The new MT model I is used here. Note that in the fire site, the train body is continuous along the tunnel. It is relatively obvious that thin steel has relatively little influence on the degree of heat loss to the train body, as its thermal resistance is very small compared to that of the insulating layer. For simplicity, it is ignored in the calculation.

*Solution:*

(1) Heat transfer coefficient

A time correction is required. To estimate the time, \( t_{01} \), the corresponding HRR should be calculated:

\[
\dot{Q}(t_{01}) = \max(H_{\text{ef}}^{5/2} \left(\frac{\Delta T}{17.5}\right)^{2/3}, u_0 b_0^{1/3} H_{\text{ef}}^{5/3} \Delta T) = 59 \text{ MW}.
\]

Here, the right-hand term in the parentheses yields a higher HRR (as \( V' = 0.43 > 0.19 \)), and is therefore used. The parameters are as follows: \( \Delta T = 800°C, b_0 = 4.8, \) and \( u_0 = 3 \) m/s. The value is obtained in kW, but presented here as MW. The corresponding time, \( t_{01} \), is around 557 seconds according to the HRR curve (\( 90000/0.19 \)).

The surface area fraction \( X_1 \) and \( X_2 \) is 0.75 (18 m/24 m) for concrete and 0.25 (6 m/24 m) for asphalt. The ratio of the surface area of the train body to that of the structure, \( A_{\text{object}}/A_{\text{w}} \), is \( (2 \times (3.8 + 2.9)) / (6 \times 6)) = 0.39 \). The time, \( t_{02} \), is:
\[
t_{o2} = \left[ \sum_{i}^{N} \left( X_{i} \sqrt{\frac{(k \rho c)}{\pi}} \right) + \frac{A_{\text{object}}}{A_{v}} \sqrt{\frac{(k \rho c)_{i}}{\pi}} \right]^{2} \frac{1}{K_{ef}^{2}} \left( \frac{\Delta T}{\Delta T_{ad} - \Delta T} \right)^{2} = 139 \text{ s.}
\]

Therefore the correction time, \( t_{c} \), is:

\[
t_{c} = t + t_{o2} - t_{o1} = 182 \text{ s for 10 min (600s +139s -557s) and 1382 s for 30 min (1800+139-557 s).}
\]

Calculations of the thermal penetration depths at 10 and 30 min also indicate that thermal penetration has not occurred. The heat transfer coefficient can therefore be calculated using:

\[
h_{k}(t_{c}) = \frac{A_{\text{object}}}{A} h_{k,\text{object}}(t_{c}) = 67 \text{ W/m}^{2}\text{K at 10 min and 24 W/m}^{2}\text{K at 30 min.}
\]

(2) Calculation of gas temperature

The initial focus is on the temperature at 10 min. Note that \( \varepsilon = 1 \) and \( \varphi = 1 \). The temperature equation is:

\[
T = T_{o} + \frac{\Delta T_{ad} - \sigma A(T^{4} - T_{o}^{4}) / (\dot{m}_{g}(t)c_{p})}{1 + h_{k}(t_{c}) / K_{ef}}
\]

In this scenario, the gas temperatures are 892°C at \( t = 10 \text{ min} \) and 1200°C at \( t = 30 \text{ min} \). The maximum temperature value can be verified using the previous model, i.e. Eq. (12), giving 1100°C at \( t = 10 \text{ min} \) and 1350°C at \( t = 30 \text{ min} \). Therefore, in this scenario the gas temperatures are 929°C at \( t = 10 \text{ min} \) and 1196°C at \( t = 30 \text{ min} \).

Although the two scenarios described above involve different fire sources, comparison of the results obtained is valuable. The temperature values that were calculated while factoring in heat loss in Example 2 are slightly higher than in Example 1, but the difference is fairly small. Therefore, for insulated trains, the heat loss to the train body generally has a negligible impact on gas temperatures.

In reality, a railway tunnel generally has gravel on the floor rather than asphalt. If gravel is assumed to have the same thermal inertia as rock, which is around \( 6 \times 10^{6} \text{ W/m}^{2}\text{K} \), the calculated gas temperatures are 879°C at \( t = 10 \text{ min} \) and 1162°C at \( t = 30 \text{ min} \).

If we further assume that the train body has the same thermal inertia as concrete, the calculated gas temperatures are 867°C at \( t = 10 \text{ min} \) and 1108°C at \( t = 30 \text{ min} \), although it should be noted that the temperatures are significantly lower than those with the insulated train body. Therefore, for trains that are not well-insulated, the train body can have a significant influence on gas temperatures.
6 Conclusions

This report presents new models for predicting the maximum excess ceiling gas temperature inside a tunnel. It constitutes further development of the work presented by the authors for small [1] and large [2] tunnel fires. In the previous MT (maximum temperature) model, a maximum possible excess gas temperature of 1350 °C, which was the highest measured tunnel ceiling excess gas temperature obtained, was set. However, this gas temperature in reality is a function of the HRR (heat release rate), fuel geometry, effective tunnel height, and ventilation conditions in the vicinity of the fire. In this work, two new models have been developed. In the new MT Models I and II, the type of tunnel structure and lining, as well as cross-sectional area, have been introduced as parameters in order to better simulate the thermal response of smoke flows in tunnels in the vicinity of a fire. This has enabled improvements to be made to the model, and facilitated more accurate predictions of the maximum possible excess gas temperature beneath tunnel ceilings during large fires.

Through comparisons of calculation models with experimental data from both full- and small-scale tests, the predictions made using both models correlate well with the test data. However, Model I is more optimal due to the fact that it is more conservative and easier to use.

The new MT models factor in the effects of scaling through the energy balance of the flame volume in the vicinity of the fire. This enables final gas temperature to be more accurately predicted for a given duration. In the previous model for large fires [2], the maximum excess gas temperature on both large and small scales was difficult to predict due to the fact that no consideration was given to the thermal response of the surrounding structure, as well as the fact that scaling was not factored in. The previous model resulted in very high excess gas temperatures for large fires on the model scale, while the new ones predict scaled experimental data in a more realistic fashion. The importance of the duration of the fire is also factored into the model in the form of both thermal response and the heating of lining surfaces near to the fire. The most important parameters with regard to determining the final excess gas temperature are the exchange of flame heat flux and the heating of nearby surfaces. The models can factor in different types of lining material, road surface material, and burning vehicle envelopes.

The models could also be used in the real world to estimate temperatures in a fully developed compartment fire, with the exception of different definitions for two key parameters.

The model has been validated against both large- and model-scale test data. The correspondence is encouraging, and further development and validation of the model by testing has been proposed. It is felt that focus should be placed on parametric study of the effects of lining, ventilation, and geometry in relation to both fuel geometry and tunnel cross-section.
References

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