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The impact of wall heating on longitudinal ventilation

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ABSTRACT

When a downgrade tunnel is ventilated longitudinally, the increase of air temperature due to a fire has two main negative consequences: a loss of efficiency of the jet-fans, and adverse buoyancy effects. To estimate air velocity, bulk air temperature has therefore to be computed precisely. In the present paper, a simplified model is first described, showing the importance of taking into account wall heating. The code Camatt, which incorporates a finite volume formulation of the heat equation in the walls of tunnels, is then used. It shows that such a computer programme should be used systematically when designing a ventilation system for a tunnel, at least as a verification, but preferably as early as the first definition calculations.

NOMENCLATURE

C_p	Thermal capacity	[J.kg ⁻¹ .K ⁻¹]
D_h	Hydraulic diameter of the tunnel	[m]
f	Friction factor from Moody's diagram	[-]
F	View factor between smoke and wall	[-]
h	Global heat transfer coefficient	[W.m ⁻² .K ⁻¹]
h_c	Convective heat transfer coefficient	[W.m ⁻² .K ⁻¹]
h_e	Effective heat transfer coefficient	[W.m ⁻² .K ⁻¹]
h_r	Radiative heat transfer coefficient	[W.m ⁻² .K ⁻¹]
Pr	Prandtl number of air	[-]
Q	Total heat release rate of the fire	[W]
S	Tunnel section	[m ²]
T_f	Bulk temperature in the fluid	[K]
T_r	Temperature at infinite depth in the wall	[K]
T_s	Surface temperature of the wall	[K]
U	Air velocity	[m.s ⁻¹]
x	Longitudinal coordinate along the tunnel	[m]
z	Coordinate along the direction perpendicular to the wall surface	[m]
α	Thermal diffusivity of the solid: $\alpha=\lambda/(\rho C_p)$	[m ² .s ⁻¹]
ε	Emissivity of the wall	[-]
$\phi(x,t)$	Heat flux given by the air to the wall per square meter of tunnel wall	[W.m ⁻²]
λ	Thermal conductivity of the wall	[W.m ⁻¹ .K ⁻¹]
ρ	Air density	[kg.m ⁻³]
σ_0	Stefan Boltzmann constant (= 5.68e-8)	[W.m ⁻² .K ⁻⁴]

INTRODUCTION

In road tunnels, large fires are relatively rare, but are a major threat for the safety of road users. The French regulations (Ministère de l'équipement 2000) demand the effect of heat to be taken into account for assessing the efficiency of jet-fans for longitudinally ventilated tunnels. It is therefore needed to calculate the evolution of air temperature in tunnels in case of a fire. Because the principle of longitudinal ventilation is to push all the smoke in one direction, creating a lot of mixing, the air temperature at a given section can be approximately considered as homogeneous, except in a thin boundary layer very close to the walls. The problem of calculating the air temperature is therefore to determine the correct thermal boundary condition at the wall.

The simplifying assumption made in the seventies was to consider that the walls were adiabatic, thus overestimating a lot the air bulk temperature, and underestimating too much the efficiency of tunnel ventilation. It appeared therefore very soon that a better model was needed. Constant temperature walls were then introduced in models, with a convective and radiative heat transfer from the fluid to the wall. However, for high air bulk temperature, this leads to very high heat fluxes, and therefore to a too fast cooling.

In order to have the results of these models match the values found in experiments, the values of the heat transfer coefficient are consequently set empirically to much smaller values than their theoretical ones.

With the increase of computer power, it is now possible to perform coupled simulations, where the evolution of air temperature in the tunnel is computed after having solved the heat equation in the walls. The target of the present paper is to analyse the effects of taking into account this wall heating during a fire.

This paper is organised as follows: a simplified problem in which the air velocity was maintained constant during the complete fire is analysed first. This allowed us to design a simplified model for wall heating and to analyse the impact on sizing of jet fans for longitudinal ventilation. As a consequence of this study, it was decided to implement a finite volume quasi-1D heat equation for the walls in the new software Camatt (Cetu 2003a), which was being designed at that time. Results from Camatt for a realistic case are shown, emphasising the very large influence of taking into account wall heating.

SIMPLIFIED MODEL

Wall heating

The usual geometry of road tunnel cross sections (ratio of the concrete thickness to the tunnel hydraulic radius) makes it possible to approximate the theoretically axis-symmetric heat transfer problem to a one-dimensional one. Moreover, the thermal diffusivity of most concretes is small. This means that the temperature elevation in the wall is slow and not very deep. The wall can therefore be considered as a semi-infinite solid for the whole fire duration, leading to the one dimensional form of the heat equation:

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial z^2} \quad \text{Eq. 1}$$

where z is the coordinate along the axis perpendicular the wall surface ($z = 0$ at the wall surface). The temperature at infinite depth is T_r . For constant surface temperature $T_s = T_r$, the analytical solution of Eq. 1 is (Incropera and DeWitt 1996):

$$\frac{T(z, t) - T_s}{T_r - T_s} = \text{erf}\left(\frac{z}{2\sqrt{\alpha t}}\right) \quad \text{Eq. 2}$$

where erf is the error function defined by: $\text{erf}(x) = \frac{2}{\sqrt{\pi}} \int_0^x e^{-u^2} du$

Using Eq. 2, one finds that the conductive heat flux at the surface is:

$$\Phi_{cd}(t) = -\lambda \left. \frac{\partial T}{\partial z} \right|_{z=0} = \frac{\lambda(T_s - T_r)}{\sqrt{\pi \alpha t}} \quad \text{Eq. 3}$$

In reality, the wall surface is not maintained at a fixed temperature, but air is flowing along it, with a known bulk temperature. This makes it necessary to use an exchange coefficient description of the heat exchanges taking place in the very thin thermal boundary layer in the fluid. The convective heat flux given by the fluid to the solid per unit area of wall is:

$$\Phi_c = h_c(T_f - T_s) \quad \text{Eq. 4}$$

where h_c is the convective heat transfer coefficient. The correlation proposed by Pethukov (Incropera and DeWitt 1996) for turbulent flow in circular tubes is widely used and is of the form:

$$h_c = \frac{\left(\frac{f}{8}\right) C_p \rho U}{1.07 + 12.7 \left(\text{Pr}^{2/3} - 1\right) \sqrt{\frac{f}{8}}} \quad \text{Eq. 5}$$

where f is the friction factor given in the Moody diagram and defined by :

$$f = \frac{-2 \left(\frac{dP}{dx}\right) D_h}{\rho U^2}.$$

Supposing that the smoke behaves like a black body, the net heat flux given by radiation by the smoke to the wall per unit area is:

$$\Phi_r = \sigma_o \varepsilon F (T_f^4 - T_s^4) \quad \text{Eq. 6}$$

where the view factor F is the ratio of energy emitted by the smoke that reaches the wall and ε the wall emissivity. In a first approximation it can be fixed at a value of 1. Eq. 6 can be written using a radiative exchange coefficient h_r which is a function of the temperature surface T_s :

$$\Phi_r = h_r (T_f - T_s) \quad \text{where} \quad h_r = \sigma_o \varepsilon F (T_f + T_s) (T_f^2 + T_s^2) \quad \text{Eq. 7}$$

The conservation of the heat flux at the wall surface imposes that Φ is also equal the heat flux penetrating the solid at the wall surface:

$$-\lambda \frac{\partial T}{\partial z} \Big|_{z=0} = (h_c + h_r) (T_f - T_s) \quad \text{Eq. 8}$$

The simplified model proposed in this section consists in approximating the temperature profile in the wall by a linear profile of slope $-(T_s - T_r) / \sqrt{\pi \alpha t}$ for $z \leq \sqrt{\pi \alpha t}$, and a flat profile $T = T_r$ for $z > \sqrt{\pi \alpha t}$, as shown in Figure 1. This very simple model gives the exact value for the conductive heat flux in the solid (see Eq. 3). It underestimates the penetration length of the heat wave by a factor of about 2. Indeed, at a given time t , the penetration length given by the simplified model is $\sqrt{\pi \alpha t}$ (see Figure 1), whereas the penetration length in the exact solution is $4\sqrt{\alpha t}$. However, as far as heat exchanges are concerned, the penetration length has not to be calculated precisely. The main use of this penetration length is to check the hypothesis of infinite depth of the solid : as long as the solid depth is larger than $2\sqrt{\pi \alpha t}$, the simplified model gives correct values of the heat flux and surface temperature. Hence, if the actual thickness of the solid is L instead of being infinite, the simplified model for computing heat fluxes and surface temperature is valid at times smaller than $\tau = L^2 / (16\alpha)$.

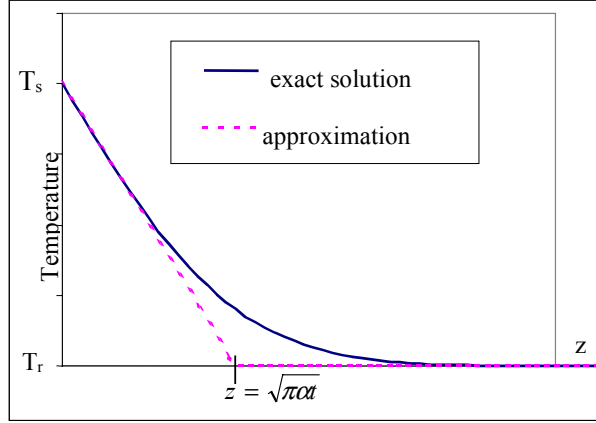


Figure 1: Exact and approximated temperature profiles in the solid for a given time t for a constant surface and infinite depth temperature boundary condition.

When the boundary condition is an actual convective and radiative condition, we still assume that the temperature profile can be mimicked by a linear profile, but introduce an empirical parameter β . The heat flux is supposed to be equal to $\lambda(T_s - T_r) / (\beta \sqrt{\pi \alpha t})$ in the solid, and therefore

$$\frac{\lambda(T_s - T_r)}{\beta \sqrt{\pi \alpha t}} = h_c (T_f - T_s) + \varepsilon \sigma_o F (T_f^4 - T_s^4) \quad \text{Eq. 9}$$

This simplified model is presented in depth in Carlotti & Voeltzel (2002), and the approximations made discussed and justified. It is in particular shown in Carlotti & Voeltzel (2002) that a good compromise value for the parameter β is $\beta = 0.75$.

Coupled model

In order to use the model presented above, it has to be coupled with a 1D model of the air-flow in the tunnel. In this section, a very simplified coupling is used, where it is assumed that the air velocity is known, and therefore there is only an equation for air temperature. The calculation of air temperature will then be used to compute the head losses and the actual thrust of the jet fans of the tunnel; if the thrust is larger than the head losses, the air velocity in the tunnel will be larger than the one postulated, and smaller in the other case.

Considering the air as a perfect gas and the mass conservation equation, writing the enthalpy balance for an infinitesimal tunnel fluid volume of length dx gives the following differential equation:

$$-\frac{1}{4} D_h \rho C_p \left(\frac{\partial T_f}{\partial t} + U \frac{\partial T_f}{\partial x} \right) = h_c (T_f - T_s) + \varepsilon \sigma_o F (T_f^4 - T_s^4) \quad \text{Eq. 10}$$

where x is the coordinate along the tunnel axis. Note that T_f and T_s are unknown. This equation is to be coupled with the heat equation in the solid. Therefore, in order to design a very simple coupled model, we need to simplify the equation in the fluid itself by getting rid of the fast terms and then to apply the ideas developed above. By neglecting the very beginning of a fire where the smoke front is propagating in the tunnel, and by assuming that the heat release of the fire is constant, we can write:

$$-\frac{1}{4}D_h\rho C_p\frac{\partial T_f}{\partial t}=h_c(T_f-T_s)+\varepsilon\sigma_oF(T_f^4-T_s^4) \quad \text{Eq. 11}$$

This equation is coupled with Eq. 8 to compute T_f as a function of x (see Carlotti & Voeltzel 2002).

Consequences for tunnel ventilation

A longitudinal ventilation system is well sized if it is able to maintain a longitudinal air velocity in the tunnel greater than the critical velocity, defined as the velocity needed to avoid backlayering in case of a fire. For this, the thrust procured by the jet fans (total available pressure gain) must be greater than the airflow resisting forces for every possible fire location in the tunnel.

For each fire location in the tunnel and for a given time, the total available pressure gain can be computed from the longitudinal air temperature profile, taking into account the jet fan efficiency loss due to the air temperature elevation, and the possible destruction of some jet fans exposed to high temperatures. The total pressure drop in the tunnel is due to the friction of air on walls and the flow resistance of vehicles. It depends on the temperature profile as the friction losses are proportional to the air velocity, which depends on air temperature.

The ventilation is sufficient if the total available pressure gain is always larger than the total pressure drop, for times lower than the time needed to evacuate the users.

The simplified model presented above was used to check the sizing of a typical road tunnel. This tunnel is 1200 m long, with a downgrade slope of 2 %, and a rectangular section (11.5 m wide and 6 m high). The friction coefficient is $\lambda_c=0.02$. As the transport of dangerous goods is allowed in this tunnel, the design fire heat release rate is 200 MW (of which 2/3 is supposed to be convective). The ventilation system was sized to maintain a longitudinal air velocity larger than 4 m.s⁻¹ upstream of the fire, according to French regulation (Ministère de l'équipement 2000). The sizing computation resulted in 8 groups of 3 jet fans placed at 100, 200, 300, 400, 800, 900, 1000 and 1100 m from the tunnel entry. All the groups of jet fans have the same characteristics, given in Table 1. The ventilation sizing was made using the hypotheses of a constant surface wall temperature during all the fire, and a total heat exchange coefficient depending on air and surface temperature as given in Eq. 7. A pressure difference of 50 Pa is taken into account to represent the meteorological pressure difference between the tunnel portals. Moreover, the blocking effect due to the volumic expansion of the fire zone is empirically evaluated to a value of 30 Pa (see Cetu 2003b). The sizing also takes into account the chimney effect, and the traffic (piston effect and resistance to flow induced by the vehicles present in the tunnel). The computations made with the simplified model used the same hypotheses as for the sizing, except for the computation of the heat transfer between air and wall, which takes into account the wall warming up.

Table 1: Characteristics of each of the jet fan groups

Thrust at 20°C	Ejection velocity	Fan effectiveness	Destruction criterion
4500 N	30 m.s ⁻¹	0.85 (-)	T _f > 400 °C

Figure 2 presents the curves of total pressure- drop and gain as a function of the fire location in the tunnel obtained with the simplified model for different times and different boundary conditions for T_s . It shows that in case of a constant surface

temperature boundary condition, the available total pressure gain is greater than the pressure drop for all fire locations in the tunnel. When taking into the wall heating in the simulation, as the time increases, the total pressure drop grows whereas the total available pressure gain decreases. At time $t = 15$ min, the available total pressure gain is lower than the total pressure drop for fire located in a short zone near the entry portal. Two hours after the fire beginning, the ventilation system is no more able to maintain the air velocity above 4 m.s^{-1} if the fire is located in the first 300 m from the entry portal.

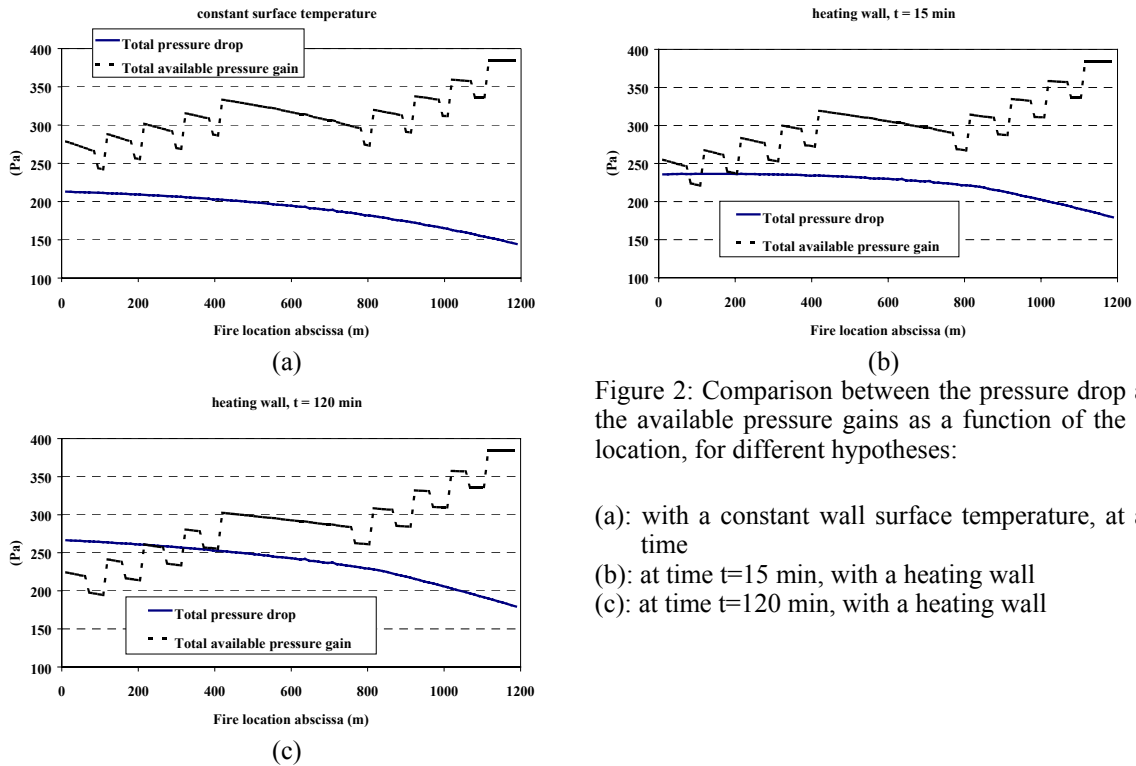


Figure 2: Comparison between the pressure drop and the available pressure gains as a function of the fire location, for different hypotheses:

- (a): with a constant wall surface temperature, at any time
- (b): at time $t=15$ min, with a heating wall
- (c): at time $t=120$ min, with a heating wall

USE OF A 1D MODEL TAKING INTO ACCOUNT WALL HEATING

In this section, a calculation using the code Camatt (see Cetu 2003a) is shown. This code was funded by Cetu and based on specifications from Cetu. After a call for bids, SETEC-TPI was commissioned to develop the code, which was finally delivered to Cetu in September 2003, after a series of validations performed by Cetu.

The code Camatt incorporates a full computation of heat exchanges through the wall, using a finite volume discretisation of the heat equation in the walls (Eq. 1), and computes the air flow using a 1D momentum equation along the tunnel. It is designed for non steady calculations, allowing us to see the evolution of air flow and temperature as function of time for a given thrust of the jet fans. The code also incorporates modules for transverse ventilation, and for the transient effect of traffic, but they are not used in the present calculation.

The heat transfer model used in Camatt was validated using comparisons of the results of Camatt with those of the simplified model shown above. For these comparisons, the module in charge of computing air velocity from the momentum equation was switched

off, and fresh air velocity was imposed as a volume flux. The results (not shown here) proved a very good agreement with the simplified model.

Once the heat transfer model validated with the momentum equation switched off, it can be used with the full dynamical model. The case demonstrated now is the same tunnel as above, with a length of 1200m, a downgrade slope of 2%, a rectangular section (11.5m wide and 6m high), and a friction coefficient λ_c equal to 0.02. The tunnel is ventilated longitudinally, with 8 groups of 3 jet fans placed at 100, 200, 300, 400, 800, 900, 1000, and 1100 from the entrance, each jet fan group having the characteristics of Table 1.

For a 200MW fire located 180m from the entrance, three simulations were performed. Simulation A assumes that the tunnel lining is plain concrete. Simulation B assumes that the tunnel ceiling is covered with insulating material, while the reminder is made of concrete. Simulation C assumes that the lining is at a constant temperature $T_w = 15^\circ\text{C}$. The characteristics of the materials are given in Table 2. In the simulations, it was assumed that the group of jet fans situated 20m downstream of the fire was destroyed at the beginning of the fire. The heat resistance of the other jet-fan groups is 400°C . The temperature of the air entering the tunnel is 15°C .

Table 2: Characteristics of lining materials

	density (kg/m^3)	Thermal conductivity (W/m/K)	Specific heat (J/kg/K)
plain concrete	2400	1.60	920
insulating material	900	0.20	1000

Since the target of the present study is to analyse the influence of wall heating, all other factors are made as simple as possible. In particular, the jet-fans located at 100, 300, 400, 800, 900, 1000 and 1100 m from the entrance portal are already running before the beginning of the fire and the flow without fire has reached a steady state. Similarly, the maximal heat release rate is assumed to be reached very quickly (1 min), which is not pertinent for a real fire, but simplifies the analysis.

Figure 3 shows results of simulations A, B, and C. It shows that, according to simulation C, the jet-fans are oversized, since they give a fresh air velocity of 5 m/s while 4 m/s is enough in such a case according to French regulations. On the other hand, in simulation A, the fresh air velocity falls below 4 m/s after 25 min, and in simulation B after about 15 minutes. In both simulations A and B, the fresh air velocity falls to even smaller values, in particular when the temperature becomes too high for a group of jet-fans, which occurs at 1h 35 min in simulation A, and at 55 minutes and 1h 40 min in simulation B.

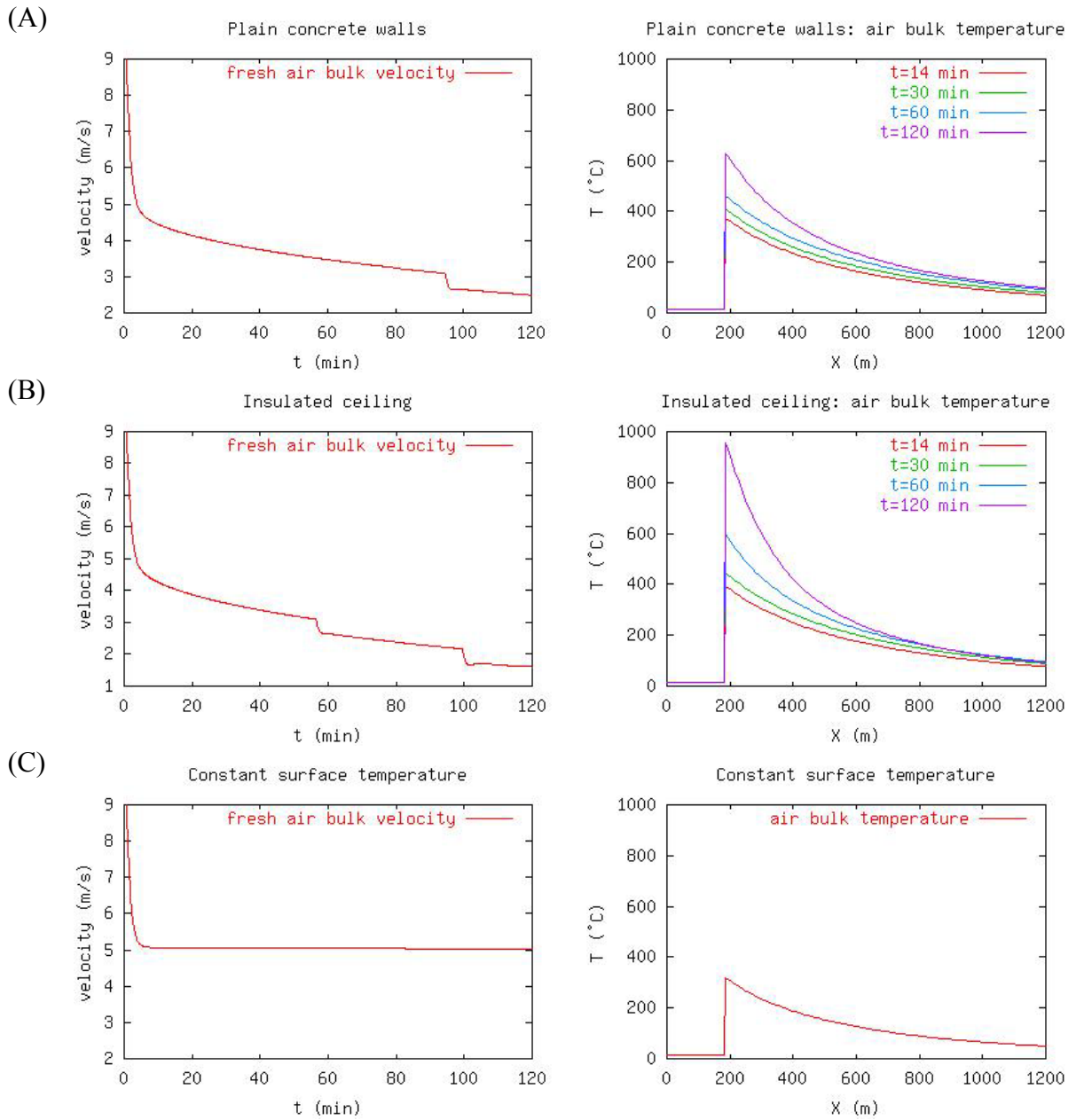


Fig 3: fresh air velocity as a function of time and temperatures as functions of distance to the entrance portal for several times; simulations (A), (B) and (C). Note that for simulation C, all the temperature curves are superposed.

CONCLUSION

A vehicle fire in a tunnel can reach a high Heat Release Rate, up to 200 MW for lorries transporting dangerous goods. Therefore very high temperatures can be reached in the bulk of the air flow. The present paper presented investigations on how heat exchanges with the walls of the tunnel should be taken into account in such a case. A simplified

model was first exposed. It showed that the effect of wall heating has to be taken into account when simulating a high heat release rate fire, since it progressively blocks the heat flux towards the wall, thus leading to an ever increased bulk temperature of the mixture of air and smoke.

As far as longitudinal ventilation is concerned, this increased temperature can lead to a severe drop of jet fan performance, or even to their destruction, and to a very strong adverse buoyancy effect in downgrade tunnels, making it impossible to maintain a longitudinal flow above the air velocity which pushes all the smoke on a single side of the fire (the so-called critical velocity). This could lead to the failure of longitudinal ventilation to ensure the safety of tunnel users in such a case.

As a consequence, it was decided that the new 1D code Camatt (Cetu 2003a), developed from Cetu specifications, should incorporate wall heating in its abilities. Camatt, as demonstrated in the last section of the present paper, allows its user to optimise the sizing of jet-fans, by taking into account the adverse (or sometimes favourable) effect of wall heating on the bulk temperature of the mixture of air and smoke.

The calculations using Camatt confirmed the results found with the simplified model, and showed in particular that, when designing a tunnel with insulating material, the thermal characteristic of the lining material have a large influence on air motion and temperature in the tunnel. Such a computer programme should be used systematically when designing a ventilation system for a tunnel, at least as a verification, but preferably as early as the first definition calculations.

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