A SIMPLE ENGINEERING MODEL FOR SPRINKLER SPRAY INTERACTION WITH FIRE PRODUCTS

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ABSTRACT

Analytical model is developed to predict smoke layer temperatures after activation of sprinkler or water mist system. The governing equations are solved to determine water droplet trajectory, temperature history and evaporation rate. As a result, a rate of heat absorption by water spray and temperature history of the hot smoke layer can be predicted. A good agreement is demonstrated between the analytical model and the results of full CFD simulations.

1. INTRODUCTION

The problem of predicting smoke layer temperature is typically arising in many fire safety applications. A part of this general problem is a prediction of smoke layer behaviour upon activation of fire suppression systems. Typical examples of such systems are water sprinklers and water mist nozzles. Heat absorption by water spray generally leads to the drop of the overall temperature inside the compartment. Quantitative prediction of this effect is quite important for estimation of attainable conditions inside buildings during fire events.

Primary differences between water sprinkler and water mist systems are in water discharge rates and droplet size distribution. Water spray for conventional sprinkler is relatively "coarse", with significant amount of droplets larger than 1 mm. In contrast, mist spray is defined by NFPA as the spray containing 99% of its volume in the droplets smaller than 1 mm. Recent interest in water mist application is mostly driven by phasing out of halons, and the need to find a replacement which can act as volumetric suppression agent. Due to very fine dispersion, mist is capable of flooding the room, although it does not have as good flooding properties as gaseous agents. Due to high surface area of the droplet phase, water mist evaporates very effectively, and therefore water discharge quantities may be kept much smaller, compared to conventional sprinklers. Water mist spray does not generally penetrate to the surface of burning material (due to low momentum of individual droplets), and suppress flame in gaseous phase. In contrast, sprinkler sprays are most effective for unshielded fuel surfaces, as droplets easily penetrate to burning material, cool the surface, and suppress pyrolysis process.

Computational Fluid Dynamics (CFD) modelling of water sprinkler sprays is well-established [1-3]. However, this technique is relatively expensive. Typically, tens of thousands of trajectories are tracked to represent water spray statistically.

For real fire engineering applications, this may not be necessary in all situations. Basic understanding of water spray behaviour can be achieved using simple estimations of droplet dynamics, similar to considerations applied in the pioneering work of Chow and Tang [4].

The model described in the present paper provides a more accurate estimation of droplet interaction with the hot layer of combustion products. In contrast to the paper [4], heat transfer problem for the moving droplet is solved in addition to the dynamic problem. The major advancement of the model is the analytical solution for the droplet motion, obtained for the case of real drag coefficient correlations, applicable for spherical particles moving in a turbulent gas stream.

The equations and the method to obtain analytical solutions are discussed first. From the solutions, the droplet trajectories and heat absorption rates for individual droplets can be predicted. Such predictions are illustrated for various droplet sizes and different temperatures of the hot layer. Finally, the model is utilized to make predictions of temperature histories in compartments after activation of water sprinklers. The results from the present analytical model are compared to the CFD simulations of the similar problem.

2. RESULTS AND DISCUSSION

2.1 Sprinkler Spray Dynamics within the Smoke Layer

Consider an interaction of a sprinkler spray with the hot layer of certain thickness Δ and temperature T (Fig. 1). It is assumed that absorption of heat by the spray within the layer is uniform. This assumption is reasonably accurate if the width of the spray is comparable with the length scale of the fire ceiling jet, as has been demonstrated in [7].

The droplets injected into the hot smoke layer undergo two major stages:

(1) heat-up stage and
(2) evaporation stage.

The overall heat absorption by a single droplet depends on its residence time within the layer. If the residence time is large, then the droplet may be completely evaporated.

Droplet dynamics and heat transfer equations must be solved simultaneously to obtain droplet trajectory and temperature history.

In order to develop a simplified analytical solution, let us consider the stages (1) and (2) separately.

2.1.1 Heat-up stage

Droplet dynamics: During the heat-up stage, the droplet is considered as having constant mass, i.e. there is no evaporation. This assumption can be shown to be quite accurate [5].

The general equation of the droplet motion through a gas layer (neglecting forces other than drag and gravity) may be written as the following set of two differential equations for the velocity components [6]:

$$m\frac{dU}{dt} = -\frac{\pi}{8} d^2 \rho C_D \sqrt{U^2+V^2}\, U$$

$$m\frac{dV}{dt} = -\frac{\pi}{8} d^2 \rho C_D \sqrt{U^2+V^2}\, V + mg \qquad (1)$$

For a typical conventional sprinkler the Volumetric Mean Diameter is of the order of 1 to 2 mm. Taking the characteristic droplet size as 1 mm, Reynolds particle number for the droplets exiting orifice at velocities of 10 to 20 ms^{-1} may be estimated to be Re ~ 700 to 1500. In this range the drag coefficient for spherical particles is a weak function of the particle Reynolds number [6] and varies between 0.47 and 0.6. Therefore, it may be represented by a constant average value with a reasonable accuracy.

It is obvious that even with the constant drag coefficient, the equations (1) are still coupled. However, since droplets hit the sprinkler deflector, their vertical velocity is usually small compared to the tangential velocity. Since the drag coefficient is assumed to be constant, then in the case V << U, equations (1) become decoupled and may be solved separately.

The system is simplified as follows:

$$m\frac{dU}{dt} = -\frac{\pi}{8} d^2 \rho C_D U^2$$

$$m\frac{dV}{dt} = -\frac{\pi}{8} d^2 \rho C_D UV + mg \qquad (2)$$

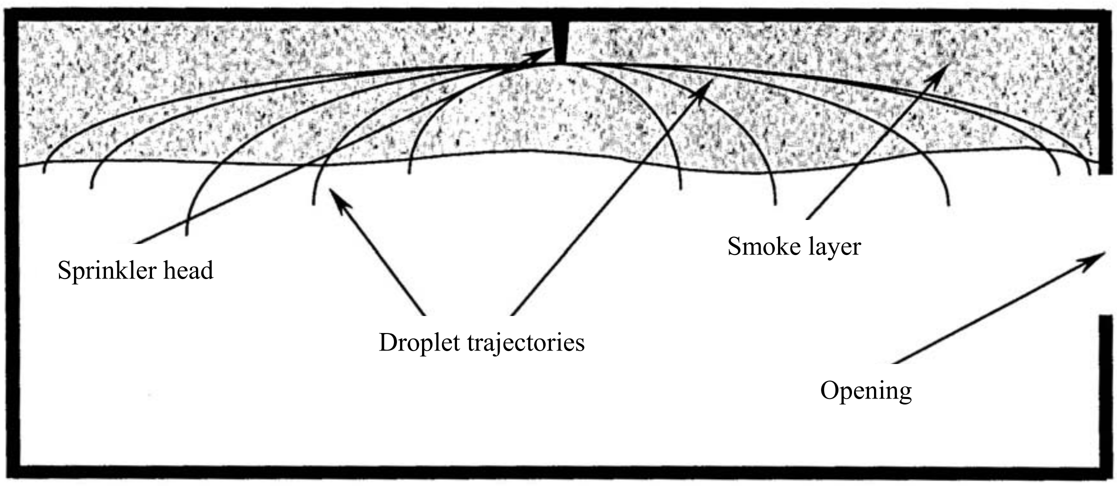

Fig. 1: Schematic view of smoke layer and droplet trajectories in a compartment

The first equation contains only tangential component, and can be easily solved by separating variables. The solution has the following form:

$$U = \left(\frac{1}{U_0} + \frac{\pi}{8m} d^2 \rho C_D t \right)^{-1} \qquad (3)$$

Substitution of this expression to the second (V – component) equation yields:

$$m \frac{dV}{dt} = -\frac{\pi}{8} d\rho C_D \left(\frac{1}{U_0} + \frac{\pi}{8m} d^2 \rho C_D t \right)^{-1} V + mg \qquad (4)$$

To find solution of (4) one needs to solve homogeneous equation first (which is done by separating variables). Then a method of variation of parameters may be applied to find solution of the non-homogeneous equation.

The final solution of equation (4) can be written as:

$$V = \frac{\kappa \left(V_0 - \frac{g}{2} \kappa \right)}{t + \kappa} + \frac{g}{2} (t + \kappa) \qquad (5)$$

where

$$\kappa = \frac{4}{3C_D} \frac{\rho_p}{\rho} \frac{d}{U_0} \qquad (6)$$

If the droplet leaves the smoke layer during its heat-up stage, then the following equation may be easily derived to estimate the droplet exit time t^* from the layer:

$$\kappa \left(V_0 - \frac{g}{2} \kappa \right) \ln \left(1 + \frac{t^*}{\kappa} \right) + \frac{g}{4} t^* \left(t^* + 2\kappa \right) = \Delta \qquad (7)$$

Heat absorption: The heat transfer equation for the droplet heat-up stage may be written as:

$$\frac{4}{3} \pi (d/2)^3 \rho_p C_p \frac{dT_d}{dt} = \frac{kNu}{d} \pi d^2 (T_{sm} - T_d) \qquad (8)$$

The correlation for the Nusselt number in the case of spherical droplets moving through the air is taken in the form [6]:

$$Nu = 2.0 + 0.6 \cdot Re^{1/2} Pr^{1/3} \qquad (9)$$

It is seen from this expression that the Nusselt number is a rather weak function of the particle Reynolds number ($Nu \sim Re^{1/2}$) and may be represented with sufficient accuracy by its constant average value.

Droplet temperature at any moment during the heat-up stage is obtained by straightforward integration of equation (8):

$$T_d(t) - T_0 = \left[1 - \exp \left[-\frac{6kNu}{\rho_p C_p d^2} t \right] \right] \cdot (T_{sm} - T_0) \qquad (10)$$

Expression (10) allows an estimation of the degree of a single droplet heat-up to be made. The total heat absorption by the spray can be calculated provided the droplet size distribution in the spray is known.

Taking into account the temperature change along the droplet trajectory (10), the heat loss rate to the spray from the layer, which has the temperature T_{sm} is given by:

$$\dot{Q}_{spr} = \psi c_p \int_0^\infty f(s) \left[1 - \exp \left[-\frac{6kNu}{\rho_p C_p d^2} t^*(s) \right] \right] ds \cdot (T_{sm} - T_0) \qquad (11)$$

where ψ is the water discharge rate, f(s) is the probability density function for the droplet size distribution. The time $t^*(s)$ corresponds to the end of the heat-up stage for a droplet of a given size. At this time, either droplet evaporation begins, or the droplet exits from the smoke layer.

2.1.2 Evaporation stage

Droplet dynamics: Generally, water droplets evaporate as they move through the hot gas. Once the droplet is injected into the hot layer, it will experience a period of unsteady evaporation, followed by a steady-state evaporation regime, characterised by approximately constant surface droplet temperature ("wet bulb temperature") and constant surface regression rate [5].

At high environment temperatures, wet bulb temperature approaches boiling temperature, so it may be reasonably assumed that the steady-state evaporation occurs at the surface boiling temperature. Equations of motion (2) apply for both heat-up and evaporation stages. They can be re-written in the equivalent form:

$$\frac{dU}{dt} = -\frac{3\rho C_D}{4\rho_p} \frac{1}{d} U^2$$
$$\frac{dV}{dt} = -\frac{3\rho C_D}{4\rho_p} \frac{1}{d} U \cdot V + g \qquad (12)$$

During the heat-up phase, the droplet diameter does not change, and the solution is the same as for the set of equations (2), that is given by the formulae (3),(5) and (6).

During the steady-state evaporation, however, the equations of motion (12) must be solved with the droplet diameter varying as a function of time. This function is taken as the theoretically and experimentally observed "d^2 - law", which implies

$$d^2(t) = d_0^2(t) - k_{ev} \cdot t \qquad (13)$$

The parameter k_{ev} is the so-called evaporation constant, and may be taken in the form [5]:

$$k_{ev} = \frac{8\lambda_g}{\rho_p C_g} \ln\left(1 + \frac{C_g(T_{sm} - T_w)}{H_{fg}}\right) \qquad (14)$$

With the diameter regression rate (14), the equations (12) can still be integrated analytically.

The solution can be written in the following form:

$$U = \left[\frac{3\rho C_D d_0}{2\rho_p k_{ev}}\left[1 - \sqrt{1 - \frac{k_{ev}}{d_0^2} t}\right] + \frac{1}{U_1}\right]^{-1}$$

$$V = \frac{-\frac{2d_0^2}{k_{ev}} g\left(1 - \frac{k_{ev}}{d_0^2} t\right)\left[\frac{\alpha}{2} - \frac{1}{3}\sqrt{1 - \frac{k_{ev}}{d_0^2} t}\right] + K}{\alpha - \sqrt{1 - \frac{k_{ev}}{d_0^2} t}} \qquad (15)$$

where the parameters K and α are given by:

$$K = 2\frac{d_0^2}{k_{ev}} g\left(\frac{\alpha}{2} - \frac{1}{3}\right) + (\alpha - 1) \cdot V_1$$

$$\alpha = 1 + \frac{2\rho_p k_{ev}}{3\rho C_D d_0} \cdot \frac{1}{U_1} \qquad (16)$$

(U_1 and V_1 are the droplet velocity components at the end of the heat-up stage, and d_0 is the initial droplet diameter).

Heat absorption: For calculation of the total heat loss to the spray we note that the heat loss due to evaporation of a single droplet is:

$$Q_{d_0} = m_{d_0} \cdot C_p(T_w - T_0) + H_{fg} \cdot \Delta m_{d_0} \qquad (17)$$

where the droplet mass loss at a given time is calculated as:

$$\Delta m_{d_0} = \frac{1}{6}\pi\rho_p\left[d_0^3 - \left(d_0^2 - k_{ev} \cdot t\right)^{3/2}\right] \qquad (18)$$

The total heat loss rate to the spray may then be obtained as:

$$\dot{Q}_{spr} = \dot{\psi}\int_0^{\infty}\left[C_p(T_w - T_0) + H_{fg} \cdot \left[1 - \left(1 - \frac{k_{ev}}{s^2} t^{**}(s)\right)^{3/2}\right]\right] \cdot f(s)ds \qquad (19)$$

2.1.3 Test calculations

For the purpose of calculations, the droplet size distribution function of the spray is discretized, and the droplets, therefore, are divided between finite number of groups. Each group contains droplets of the same diameter. Calculation procedure is designed in such a way that the equations for both heat-up and evaporation stages are solved consecutively along droplet trajectory for one representative droplet from each of the groups.

In general, each droplet undergoes both (heat-up and evaporation) stages, and the total heat absorption by the droplet depends on its residence time in the smoke layer.

The residence time for any particular droplet is determined using the following equation:

$$\int_0^{t^*} V^1(s)ds + \int_0^{t^{**}} V^2(s)ds = \Delta \qquad (20)$$

where $V^1(s)$ and $V^2(s)$ are taken from the solutions (5) and (15), respectively, Δ is the smoke layer thickness.

The residence time t^{res} is equal to:

$$t^{res} = t^* + t^{**} \qquad (21)$$

where t^* and t^{**} are the times for the heat-up and evaporation stages, respectively.

Equation (20) does not have solution if droplet evaporates completely in the layer. In that case, the residence time is taken as $t^* + t_{ev}$ where t_{ev} is the time for complete evaporation of the droplet:

$$t_{ev} = \frac{d_0^2}{k_{ev}} \qquad (22)$$

The total heat absorption rate is obtained by multiplying the heat absorption rate for a single droplet by a number of droplets in the respective group, and summing up over all groups in the discretized spray distribution.

The typical calculated droplet trajectories are presented in Figs. 2 to 4. The difference in the droplet behavior can be observed for the different temperatures of the smoke layer, ranging from 300 to 800 K. The trajectories are presented for different representative droplet diameters and different initial velocities. The smoke layer thickness is taken as 1 m in Figs. 2 to 4.

Generally, the droplets below 500 μm are affected by the increased temperature of the smoke. Compared to the ambient conditions (300 K, Fig. 4), fine droplets of 100 to 200 μm in diameter evaporate quite quickly in the smoke layer with the temperatures above 500 K. This may be seen from Figs. 2 and 3.

A more important parameter for fire engineering applications is the rate of heat absorption by the whole spray. In order to validate the performance of the analytical model, the results are compared to the CFD calculations, which were performed solving exact equations for the two-phase (droplet-gas) flow [7]. The results are presented for the AM24 spray nozzle. The droplet size distribution for such a nozzle [8] is shown in Fig. 5. In this case, the medium droplet diameter in the spray is of the order of 300 μm. Heat absorption rate is plotted in Fig. 6 as a function of the hot layer temperature. It appears that the agreement between the CFD and the simplified model is quite good. Note that the heat absorption rate approaches the plateau at high smoke layer temperatures. This plateau corresponds to the complete evaporation of the spray. Also shown in Fig. 6 is a convective part of the total absorption rate. As expected for water mist spray, major contribution to heat absorption is from vaporization process.

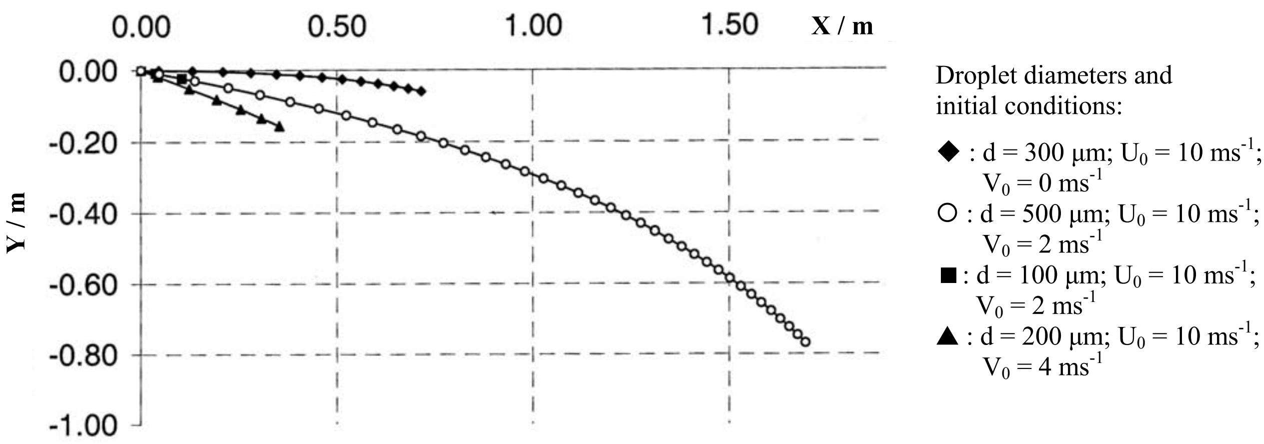

Fig. 2: Selected droplet trajectories in a 800 K smoke layer (layer depth is 1 m)

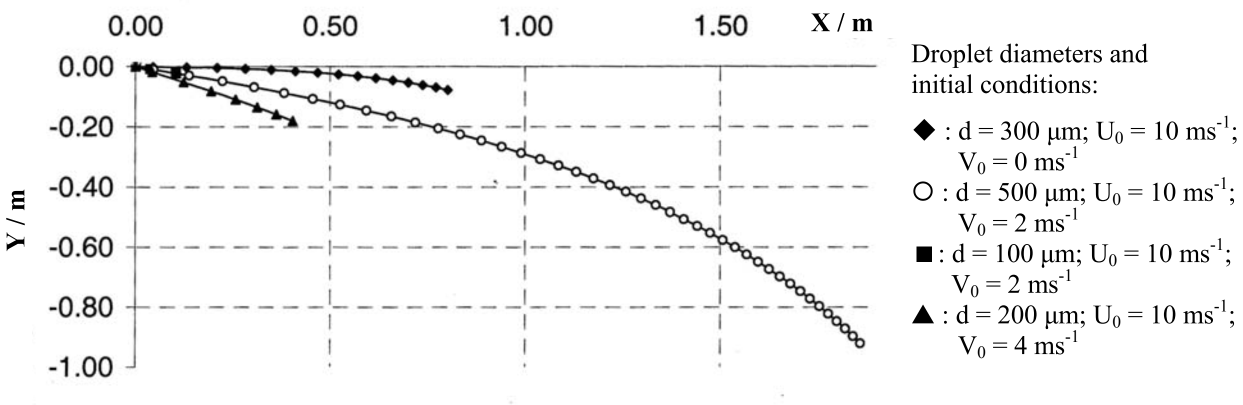

Fig. 3: Selected droplet trajectories in a 500 K smoke layer (layer depth is 1 m)

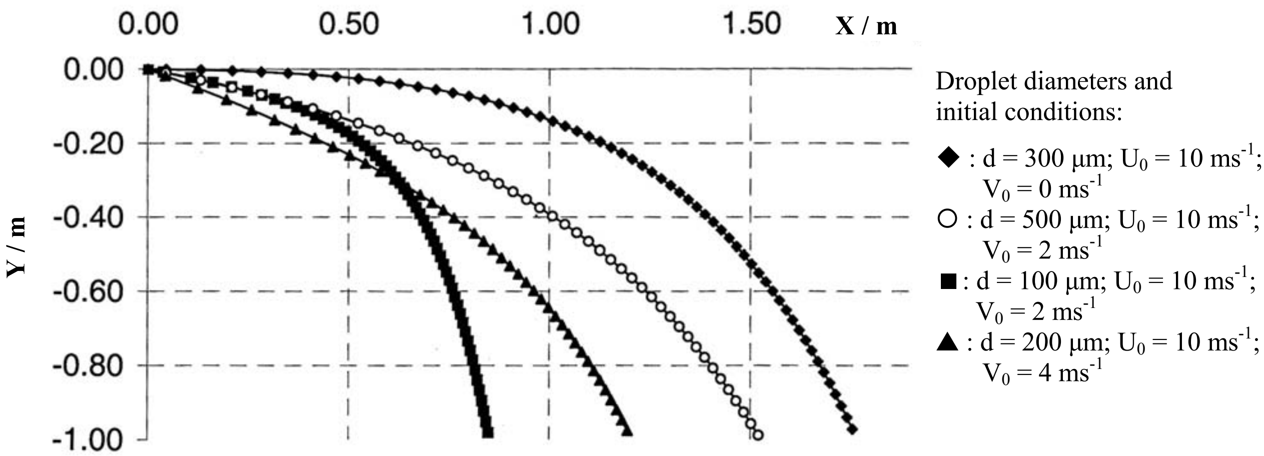

Fig. 4: Selected droplet trajectories in a 300 K smoke layer (layer depth is 1 m)

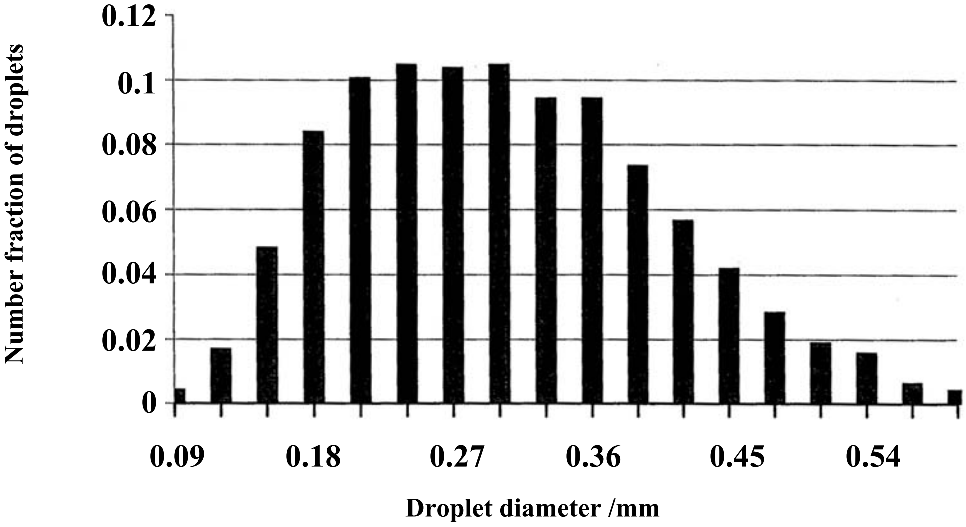

Fig. 5: Droplet size distribution for the AM24 nozzle [8]

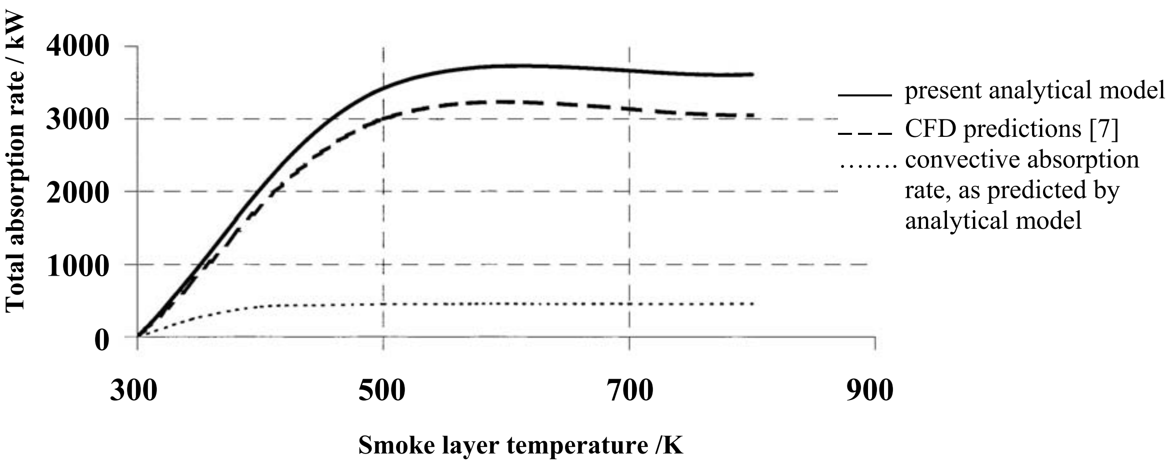

Fig. 6: Heat absorption rate predictions for the AM24 nozzle
(layer depth is 1 m; water discharge rate is equal to ψ = 80 liters/min)

The presented model can also be used to predict temperature history of the smoke layer in compartments. This is accomplished solving the simple energy conservation equation for the smoke layer:

$$M_{sm} c_{pg} \frac{dT_{sm}}{dt} = -\dot{Q}_{spr}(T_{sm}) \qquad (23)$$

where M_{sm} is the mass of the smoke layer. For the solution of equation (23), the rate of heat absorption $\dot{Q}_{spr}(T_{sm})$ must be updated as the solution progresses using the described calculation procedure (equations (2) to (22)).

Typical results are presented in Figs. 7 and 8. The first of these figures plots smoke layer temperature versus time for different water discharge rates. The water discharge rate of $\dot{\psi}$ = 80 liters/min corresponds to the nominal operating conditions for the AM24 nozzle. The other curves in Fig. 7 correspond to the reduced water discharge rates. At the initial stage after sprinkler activation, the temperature decrease rate is roughly linearly proportional to the water discharge rate. All curves in Fig. 7 approach the ambient temperature level (300 K) at infinitely long times.

The effect of compartment size on temperature decrease rate can be estimated from Fig. 8. Sample results are presented for the three different floor areas of the room. The water nozzle operates at its nominal level of $\dot{\psi}$ = 80 liters/min. The rooms with smaller floor area demonstrate higher temperature drop rate, until the temperature levels off to the ambient level. Typically, the thermal time constant of the smoke layer is of the order of 10 min.

Finally, typical droplet heat-up times (time required for the droplet to heat to boiling temperature) are presented in Fig. 9. The heat-up times are presented for different smoke layer temperatures, and for several droplet sizes, which contain most of the AM24 spray volume.

3. CONCLUSIONS

A simple analytical solution has been presented for water droplet motion and evaporation rates in a hot smoke layer. The model allows calculation of the heat absorption rates for water sprays and smoke layer temperature histories to be performed.

As a calculation test, the results are presented for individual representative droplet trajectories in hot layers of different temperatures. The calculations of the total heat absorption rate by the spray is also presented, and a good agreement is found between the simplified model and the results from comprehensive CFD simulations.

The agreement between the analytical and CFD modeling demonstrates that the present model can be applied for fire engineering calculations as a quick and economical alternative to full CFD simulations.

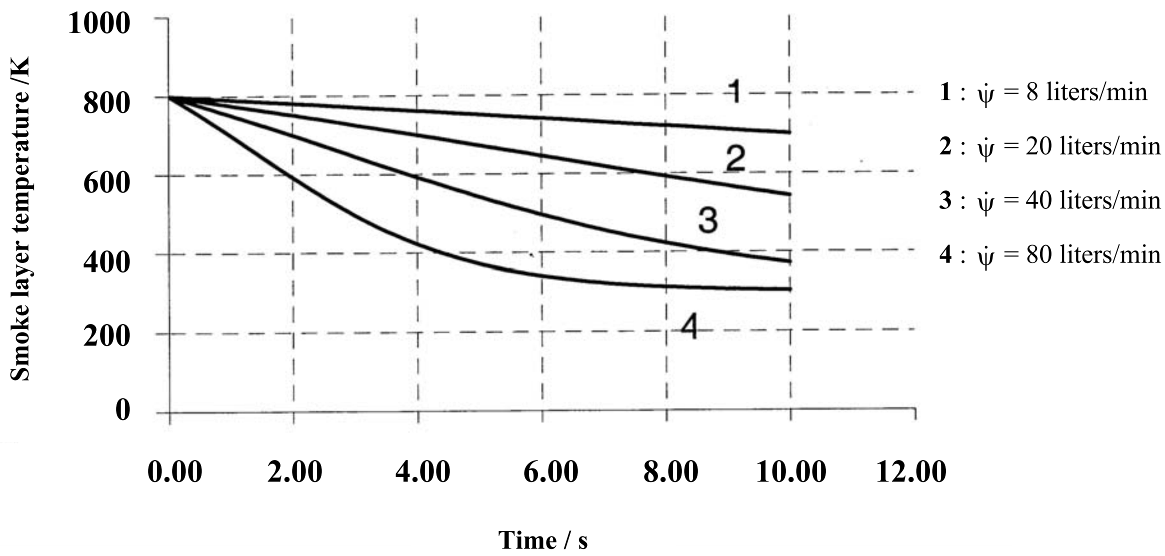

Fig. 7: Smoke layer temperature histories for different water discharge rates
(droplet size distribution corresponds to the AM24 nozzle; layer depth is 1 m)

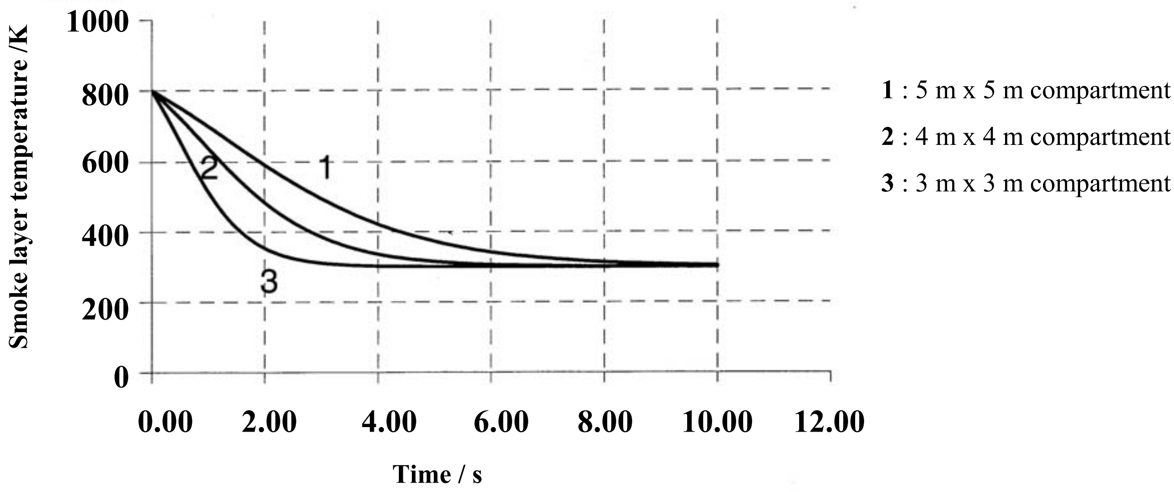

Fig. 8: Effect of the compartment floor area on the smoke temperature drop rate
(droplet size distribution corresponds to the AM24 nozzle; layer depth is 1 m;
water discharge rate is equal to $\dot{\psi}$ = 80 liters/min)

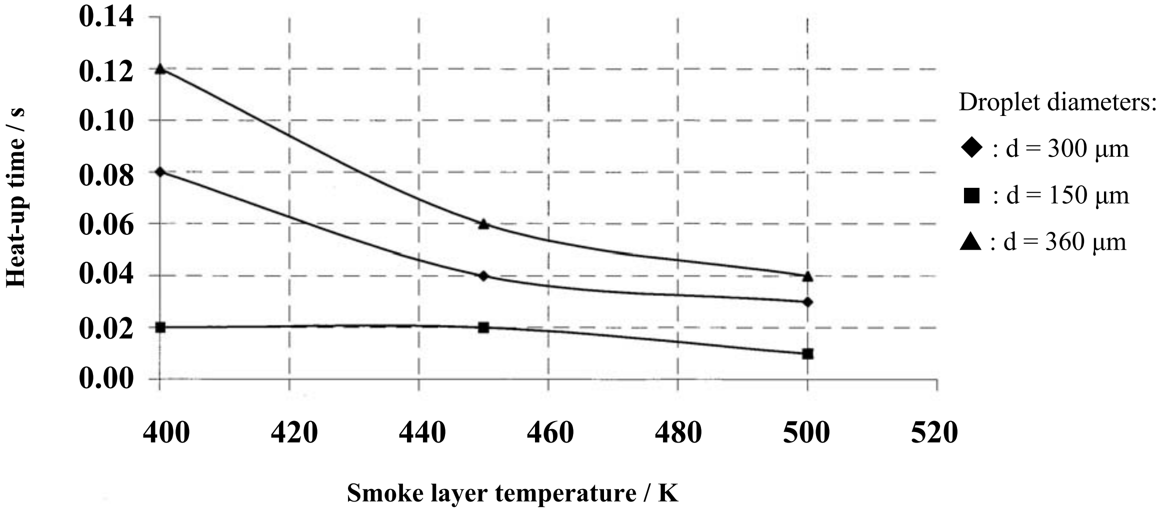

Fig. 9: Heat-up time for various droplets

NOMENCLATURE

c_p	specific heat
C_D	drag coefficient
d	droplet diameter
f (s)	pdf for droplet size distribution
H_{fg}	latent heat of water vaporisation
m	droplet mass
Nu	Nusselt number
$\dot{Q}$	rate of heat absorption
Re	Reynolds number
t	time
T	temperature
U	tangential velocity component of the droplet
V	vertical velocity component of the droplet
y	coordinate normal to the ceiling

Greek symbols

Δ	smoke layer thickness
ϑ	dimensionless temperature
ρ	density
$\dot{\psi}$	water discharge rate

Subscripts

0	initial
g	gas
w	wet bulb temperature
p	particle
spr	spray
sm	smoke

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