HEAT TRANSFER BOUNDARY CONDITIONS IN POOL FIRES

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Abstract

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The paper discusses the use of experimental measurements to validate calculations of heat fluxes from pool fires. Flame emissivity and temperature and the effects of reflected radiation are treated in simple terms. Estimates of convection effects are made and a simple model of surface sooting proposed. More experimental measurements are required to confirm assumptions.

1. INTRODUCTION

Fire tests are commonly used to demonstrate an ability to survive accident conditions. An example of this is the thermal test defined in paragraph 628 of the IAEA Regulations for the Safe Transport of Radioactive Material [1]. The practical test is specified, in part, by the geometry, a 30 minute duration and the fuel for an open liquid hydrocarbon pool fire. Calculation can be used to demonstrate survival of this standard test, in which case heat transfer boundary conditions simulating the effects of the fire must be assumed. The regulation specifies a flame temperature (800°C), a flame emissivity (0.9) and a surface absorptivity (0.8) for situations where more appropriate values cannot be assessed. The value of the convective heat transfer coefficient is not specified but must be justified by the designer.

Ideally, calculation methods should be validated by comparison with experimental results, but the variability of fires and uncertainty in the measured boundary conditions make this difficult. In practice, values for the boundary conditions are inferred from the temperature response of items exposed to experimental fires. This paper discusses the validity of such techniques and provides evidence for the choice of convection coefficients which are otherwise difficult to separate from experimental measurements of total heat flux (radiation and convection).

2. THERMAL RADIATION

Thermal radiation dominates heat input from flames in pool fires. The magnitude of the radiant heat flux depends, in a simple model, on three parameters, the flame emissivity, the flame temperature and the surface temperature. The last is a parameter derived from the calculation and depends on the properties of the body engulfed by the flames. The other two are properties of the fire and depend on geometry and other factors. In practical situations, the flame temperature varies from place to place, particularly when affected by winds, and over the duration of the fire as conditions change, perhaps due to winds again. The flame emissivity is a function of flame thickness and so will also be affected by winds.

2.1 Flame Emissivity

Flame radiation derives from luminous gases and comprises a continuous spectrum of thermal radiation, from soot solids suspended in the gas and band spectra from the component gases, largely nitrogen, carbon dioxide and water vapour. In calculations, the source is generally represented by an artificial surface at a constant temperature and with a simple thermal emissivity. If this is less than unity, care must be taken to avoid non-physical reflections from this surface.

Flames are partly transparent to thermal radiation which is attenuated exponentially when traversing the flame thickness. The effective emissivity of uniform thickness flames is given by Ref. [2].

$$\epsilon_f = 1 - \int_0^{\pi/2} \sin 2\theta . \exp(-\mu\, x_o \sec \theta) d\theta \qquad (1)$$

The variation of effective emissivity with flame thickness (μx_o) is shown in Fig. 1, where x_o is the flame thickness and μ the attenuation coefficient.

Appropriate values of the attentuation coefficient are not generally available and are difficult to measure. A typical value for smoky flames [3] is 1.3 m^{-1}, so flames of 1.0 m thickness are likely to have an effective emissivity, at the body surface, of about 0.85 while the 0.90 emissivity corresponds to a thickness of 1.6 m. Both values are thus consistent with the IAEA-specified pool fire geometry where the fuel extends by 1.0 to 3.0 m beyond the edge of the body.

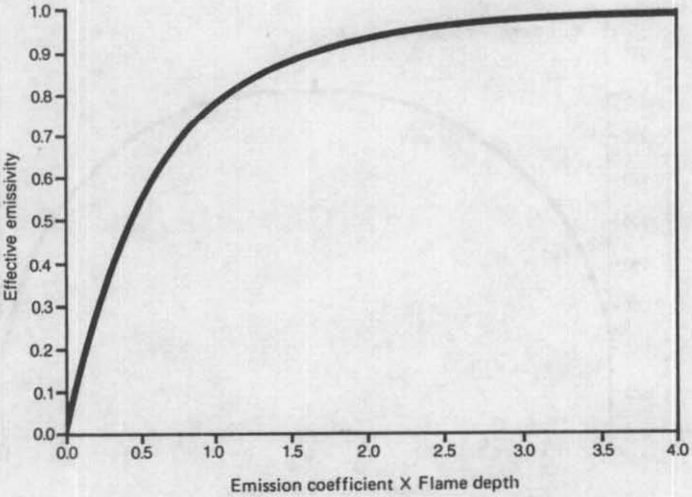


FIG. 1. Variations of effective emissivity with flame depth.

2.2 Reflected Radiation

The effective emissivity discussed above does not imply fractional reflection from the flames, but partial transmission through the flame thickness to ambient temperature surfaces outside. It is thus incorrect (but conservative) to represent the body surface and the flames as parallel surfaces radiating to each other with multiple reflection between them. Fry [1] discusses the possibility of representing the flames by a black body surface of reduced temperature T_r

where $T_r^4 = \epsilon_a T_a^4 + \epsilon_f T_f^4$ (2)

and ϵ_a is the remote ambient surface emissivity (generally unity) attentuated by the flames i.e. $\epsilon_a = 1 - \epsilon_f$

T_a is the ambient temperature outside the flames (K)

T_f is the flame temperature (K).

Thus the body within the flames will never achieve the actual flame temperature (T_f), but will reach an equilibrium at the effective flame temperature (T_r) where radiation to the external ambient surfaces balances the net radiation received from the flames.

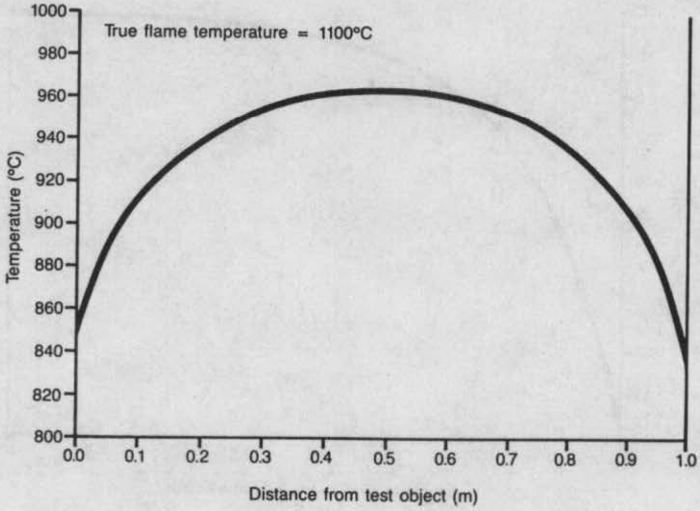


FIG. 2. Temperature measured by thermocouple in a fire.

If $T_a = 20°C$, $T_f = 1100°C$ and $\epsilon_f = 0.90$

$T_r = 1064°C$.

With this formulation and a flame emissivity of unity, the parallel radiating surface model can be used.

2.3 Fin Cavities

In general, the spacing of cooling fins on the surface of a body, such as an irradiated fuel flask, will be small compared with the reciprocal of the attenuation coefficient. The gas occupying the space between the fins will thus be relatively transparent and the effective emissivity small. The fin sides will radiate to each other with relatively little attenuation. Such effects are best modelled with specialised codes representing all these effects with a fine mesh distribution [3].

2.4 Flame Temperature

Measurements of flame temperature in experimental fires will be affected by radiation from cooler surfaces within the flames and from ambient temperature surfaces outside. The effects can be assessed for idealised flame geometries [1] with the results shown in Fig. 2. Thus, even with uniform gas temperatures within the flame thickness, a thermocouple will suggest that a temperature variation exists.

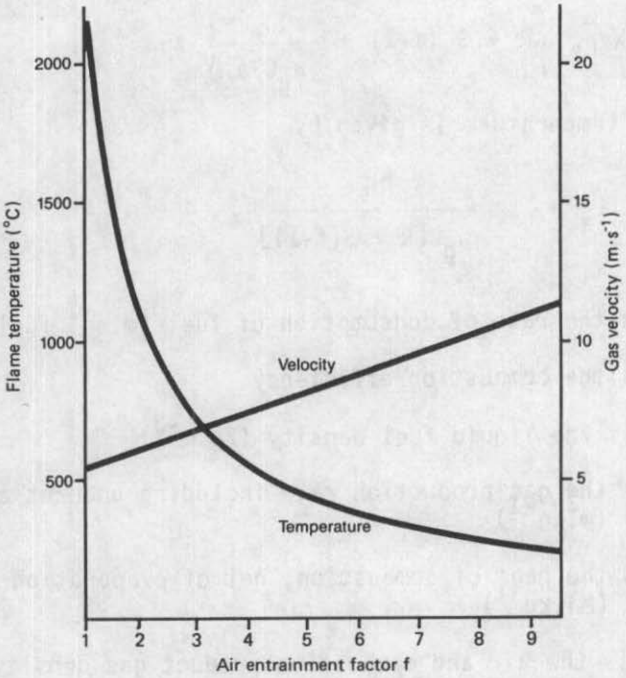


FIG. 3. Kerosene pool fire: parameters, gas velocity and temperature.

3. CONVECTION

While radiation heat transfer will normally dominate in pool fires, convection can provide a significant contribution to heat input, particularly to a surface with large fin areas. It is important to realise that natural convection effects are not likely to be important as this would imply downward gas flow adjacent to a cool surface, while the buoyant bulk flow is upwards. Forced convection formulations are obviously more appropriate so an assessment of flow velocity is important.

Flame temperature and gas velocity distributions have been assessed for general conditions by Cox and Chitty [4]. This formulation yields the distributions shown in Fig. 3 for a 2m cube. Cox [5] has made measurements of gas velocities in smoke and flame up to 1000°C, revealing values in the range 1 to 5 m.s^{-1}. Buoyancy considerations (see e.g. Corlett [6]) suggest gas velocities of 3 - 10 m.s^{-1}.

Considerations of combustion products and heat of combustion with conservation of mass and energy result in relatively simple equations. The gas velocity is given by

$$V = Xa\rho_k \left\{ R + S\ (f-1) + \frac{bq}{\rho_g CTa} \right\} \ \mathrm{m.s^{-1}} \qquad (3)$$

The flame temperature is given by

$$T_f = T_a + \frac{bq}{\rho_g C[R + S(f-1)]} \qquad \mathrm{K} \qquad (4)$$

where x is the rate of consumption of fuel ($\mathrm{m.s^{-1}}$ fuel depth)

a is the combustion efficiency

ρ_k is the liquid fuel density ($\mathrm{kg.m^{-3}}$)

R is the gas production rate including unburnt air ($\mathrm{m^3.kg^{-1}}$)

q is the heat of combustion, net of evaporation ($\mathrm{MJ.kg^{-1}}$)

ρ_g is the air and combustion product gas density at ambient ($\mathrm{kg.m^{-3}}$)

C is the specific heat of the air and combustion products ($\mathrm{J.Kg^{-1}}$)

T_a is the ambient temperature (K)

b is the non-radiated energy fraction i.e. 1-b is lost by radiation from the flames

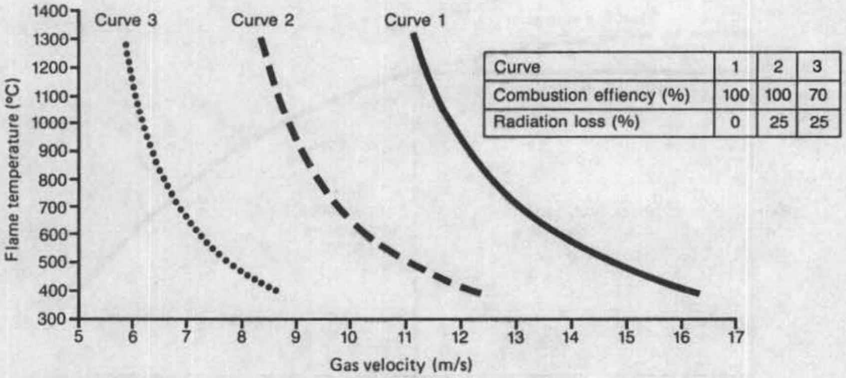
S is the air required for stoichiometric combustion ($\mathrm{kg.kg^{-1}}$)

and f is an air entrainment factor - i.e. (f-1) is the fraction of air not taking part in combustion.

Substituting values for kerosene and air with a combustion rate of 6.0 mm per minute, a combustion efficiency of 70% and 25% radiation loss (b = 0.75) yields the following relationships:

$$V = 4.75 + 0.70\ f \qquad \mathrm{m.s^{-1}} \qquad (5)$$

$$T_f = 27 + \frac{2052}{f - 0.476} \quad ^\circ\mathrm{C} \qquad (6)$$



Curve	1	2	3
Combustion effiency (%)	100	100	70
Radiation loss (%)	0	25	25

FIG. 4. Kerosene pool fire: flame temperature and velocities.

$$V = 1423 \left\{ \frac{1}{T_a} + \frac{1}{T_f - T_a} \right\} \text{m.s}^{-1} \qquad (7)$$

These equations are plotted in Figures 3 and 4, showing that with these data assumptions, gas velocities average less than 10 m.s^{-1} through the cross-section of the flames for a wide range of the unknown entrainment parameter. An average flame temperature of 800°C corresponds to a velocity of about 6.6 m.s^{-1}.

The use of these gas velocities with forced convection heat transfer correlations yields a range of possible convection coefficients. For example, the Colburn equation [7] with data for air at 500°C (film temperature) yields values of about 10-12 $\text{w.m}^{-2}.\text{k}^{-1}$ for bodies of dimensions 1 - 2 m and a 5m.s^{-1} gas velocity. This is not inconsistent with some analyses of pool fire experiments which are modelled reasonably well with a coefficient of 10 $\text{w.m}^{-2}.\text{k}^{-1}$.

4. SURFACE SOOTING

In general the surface conditions will vary through the duration of a fire as low conductivity soot accumulates on cool surfaces and appears to burn off at higher temperatures. If this is modelled simply as a condensation-evaporation phenomenon conclusions may be drawn modifying the physics of heat transfer. If a nominal condensation temperature of 500°C is assumed, the heat flux to a cool (less than 500°C) surface is independent of the surface temperature as shown in Fig. 5. The low conductivity soot rapidly attains a surface temperature of 500°C and the thickness will be adjusted automatically to maintain this temperature. Condensation or evaporation will take place

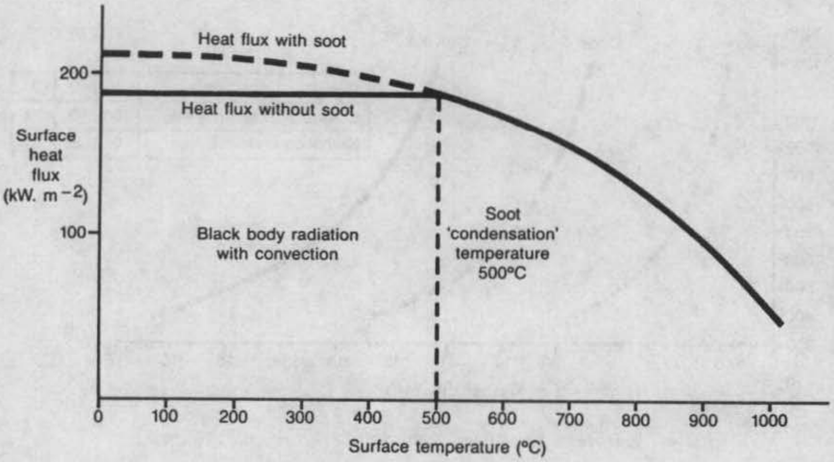


FIG. 5. Simple soot model: effect on heat flux (1100°C flames).

as the soot surface temperature tends to fall or rise. The surface exchanging heat with the flames, by radiation or convection, is thus at a constant temperature and the heat flux from the flames will be constant.

Above the 500°C evaporation temperature, no soot will remain on the surface and the normal physical heat exchange phenomena occur with an increasing surface temperature.

At present there is no data to support this simple model and no accurate knowledge of an appropriate temperature. However, the model illustrates the hazards of using highly sophisticated numerical models while ignoring physical phenomena which are difficult to describe.

5. CONCLUSIONS

This paper, and the references, illustrate the problems of inferring flame conditions from simple measurements of temperature and other parameters. Actual gas temperatures within the flames from a pool fire are likely to exceed 1000°C as evidenced by surface temperatures of this magnitude. However, when averaged around the body within the flames, and over the duration of a fire, flame temperatures close to the 800°C specified for the IAEA thermal test are likely to be obtained, particularly for the larger, Type B packages. Averaging over the flame volumes between fins will also tend to reduce effective temperatures.

More experimental evidence is required to support this contention. A variety of finned surface tests would supply an insight into the effects of radiation and convection. Such work would support the use of relatively simple calculation models, where different gas temperatures are used to represent convection and radiation.

ACKNOWLEDGEMENTS

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