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TEST UNIT EFFECTS ON HEAT TRANSFER IN LARGE FIRES

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ABSTRACT

Measurements made in large pool fires with a variety of objects have shown that there is not a one-to-one correspondence between the measured fire temperatures and either the heat flux to the object or the final surface temperature of the object. The measurements indicate that the heat flux depends on the physical size and characteristics of the object. A large, thermally massive object will heat slowly; the radiative heat flux to this cold surface is reduced by the thermal interaction between the object and the flames. A numerical model of this interaction is used to help explain the observed results.

INTRODUCTION

Fires which might occur in a transportation accident involving hazardous materials or in a petrochemical industry accident can put workers or the public at risk. In an attempt to understand and possibly reduce these hazards, there is an interest in determining the response and/or survivability of a variety of items when subjected to large fires.

Regulatory specifications have been given by the IAEA and the US-NRC for conducting simulated transportation accident fire tests of radioactive material (RAM) shipping containers as part of the overall certification testing process. These specifications are relatively simple in that a "uniform fire" boundary condition is used. The US-NRC regulation, 10CFR Part 71.73, specifies exposure "...to a heat flux not less than that of a radiation source of 800 C with an emissivity coefficient of at least 0.9...", "...the surface absorptivity must be either that value which the package may be expected to possess if exposed to a fire or 0.8, whichever is greater." This regulation does not specify the 'fire temperature'; it specifies that the minimum acceptable heat flux to a cold wall is 55.5 kW/m^2. Specifications have been given by the US-DOT, in 49CFR Part 179.105-4, for the evaluation of LPG tank car thermal insulation systems. These call for a flame temperature of 870 C $\pm$ 55 C; an average heat flux of 35.2-30.2 kW/m^2 is required for a calibration specimen to reach a temperature of 430 C during a 12-14 minute calibration.

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Because they are expensive, only a limited number of large tests have been conducted over the years to determine the thermal exposure in an engulfing fire. Several models have been developed for predicting the radiative heat transfer in sooty fires in which the flames are modeled as having a uniform temperature and extinction coefficient. Scale model tests have been tried to reduce costs and increase understanding. The success of either analytical or scale model approaches has been limited by the complexity of real fires and the interaction between the fire and the test unit. A paper by Copley (1967) indicates that eleven dimensionless variables must be matched for a perfect simulation of a fire test.

There has appeared to be a discrepancy between the average heat flux levels estimated from the response of large test objects and that measured with small sensors (Wachtell and Langhaar, 1966, or Anderson, et al, 1974). A number of heavily instrumented, large pool fire tests were conducted at Sandia National Laboratories to provide information on the thermal exposure in fully engulfing fires and the repeatability from fire to fire (Gregory, et al, 1989). From these fires it was observed that:

1) the heat flux to a test object that is "physically large" and "thermally massive" is significantly lower than that measured by small sensors (Gregory, et al, 1987 & 1989, Schnieder, et al, 1989); and,
2) the heat flux distribution on a large horizontal cylinder is different from that predicted by the multidimensional models.

This paper will present a summary of the experimental data developed from some large pool fire tests designed to evaluate the thermal exposure. To examine the influence of the test unit on the heat transfer, an analytical model of the radiation/convection interaction between a large vertical plate and the surrounding fire environment was developed. The model is used to examine the effect of surface temperature, test unit size, gas velocity, extinction coefficient, and volumetric heat release rate on the heat flux to the plate. The results of this model corroborate the results of the experimental work; it correctly predicts the magnitude of the experimentally observed reduction in the heat transfer to a thermally massive object engulfed by flames.

TEMPERATURE AND HEAT FLUX MEASUREMENTS

Most considerations of the fire environment deal with the 'fire temperature' and make the tacit assumption that the higher the temperature the higher the heat flux. The comparison given above of regulations dealing with rail tank cars and RAM shipping containers shows that this can be a misleading assumption. Measurements of both the temperature and the heat flux are necessary to define the thermal exposure in a fire environment. Temperature is important because material operating limits are directly related to temperature or to the time above some temperature. The initial response of the test item is governed by the heat flux and the thermal properties of the item. The heat flux levels indicate how severely an item will be thermally stressed.

Making accurate measurements of temperature and heat flux in fires is a difficult proposition. In most cases, temperatures in the flames are measured with "small" thermocouples. These types of measurements may be in error. An alternate method is the Directional Flame Thermometer described by Fry, 1989; the device is a thin metal plate with a number of radiation shields on one side and the other side exposed to the fire. Its readings are similar to the small thermocouple readings near the center of fires such that the product of the extinction coefficient and the path length to the flame edge is at least 2 to 3. Measurements of heat fluxes often use either Gardon gages (i.e., circular foil heat flux gages) or slug calorimeters. Each of these devices offers potential problems (Keltner, et al, 1989).

In general, an accurate estimate of the heat flux cannot be obtained from temperature measurements and vice versa. The problem is demonstrated in Figure 1 with a compilation of calorimetry data from several large fire tests at Sandia using cylindrical and vertical plate designs for the slug calorimeters. The heat transfer rates were obtained with temperature data from slug calorimeters and an inverse heat conduction computer code called SODDIT (Blackwell, et al, 1987). Because these calorimeters are of the transient type, the temperature increases as they absorb energy. The heat fluxes are corrected for radiation heat transfer to a cold wall value using

$$Q_{cw} = Q_{net} + \sigma\varepsilon(T_s^4 - T_{cw}^4) \qquad (1)$$

where T_s is the actual surface temperature and T_{cw} is the cold wall temperature. No cold wall correction has been made for convective heat transfer, although a convective heat transfer coefficient of 57 W/m^2-K was found to give good results in a recent paper (Nakos and Keltner, 1989). Figure 1 shows the average cold wall heat flux plotted versus the average "flame temperature" at the same location (This figure is taken from Schneider, et al, 1989.). The figure shows that the cold wall heat fluxes are higher on the vertical plate than on the 1.4 m horizontal cylinder. Also shown is the blackbody radiative heat flux versus flame temperature.

The agreement between the average values of the cold wall heat flux and the blackbody flux predicted from the average values of the flame temperature is not very good. The measured fluxes are higher than the predicted ones at locations with lower flame temperatures, by as much as a factor of two. Part of this difference is explained by analyses, such as those of Fry (1985), that show the temperatures measured in the fire, especially those close to a cool surface, can be significantly lower than either the actual flame temperature or the effective radiation temperature. Another part of this difference is explained in Schneider, et al, as being due to the large fluctuations in the flame temperatures. In general, locations with lower average temperatures show higher standard deviations. Due to the fourth power relationship between the radiative flux and the temperature, the use of the average temperature underpredicts the average radiative flux in a fluctuating temperature environment.

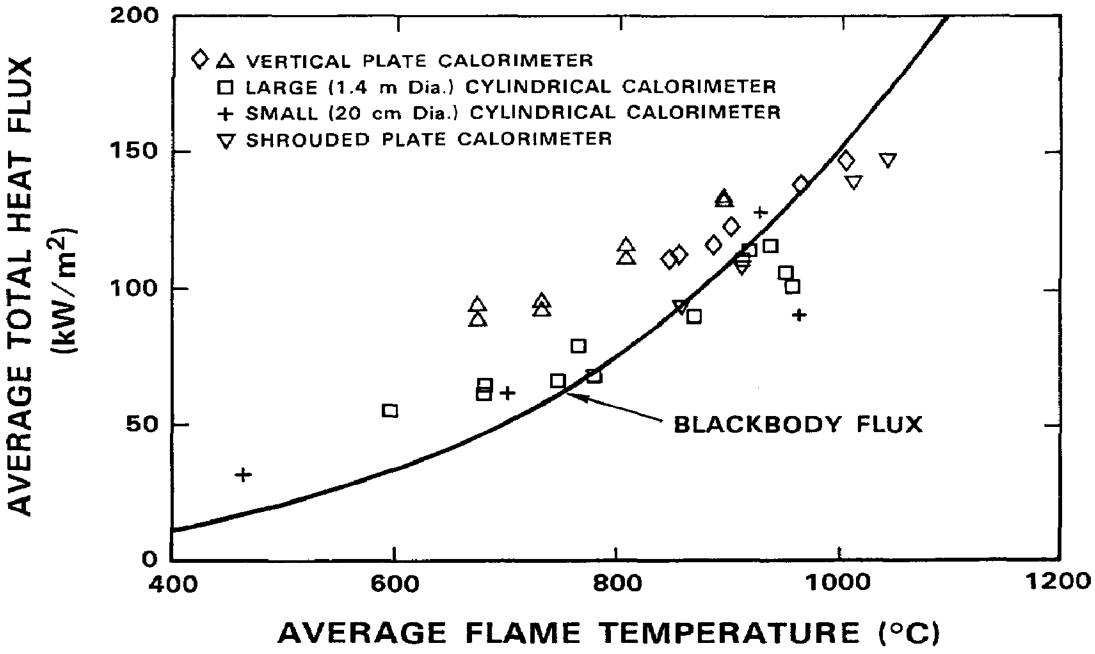

Figure 1. Average Cold Wall Heat Flux as a Function of Average Flame Temperature

A limited number of models have been developed to try to describe the heat flux levels and distributions. These models have been developed for predicting the radiative heat transfer in sooty fires in which the flames are assumed to have a uniform temperature and extinction coefficient. A model for a flame layer has been developed by Fry (1985); it shows how limited flame thickness can affect radiative heat transfer in sooty fires and how the thickness and the presence of a cold wall can affect temperature measurements near the surface. Birk and Oosthuizen (1982), developed a model for a two-dimensional flame volume. Tunc and Karakas (1985), and Wong and Steward (1988), developed models for three-dimensional flame volumes. The ability of these models to predict the heat flux depends on how well the assumption of a uniform, constant temperature models the flame volume.

Measurements in small fires show that the temperature in the "continuous flame zone", as defined by McCaffrey (1979), is relatively uniform. Average temperature data from several instrumentation towers in the "continuous flame zone" of the 9 m by 18 m JP-4 fires conducted at Sandia are shown in Figure 2. The temperature is not constant with elevation. It should be noted that the temperature at any single location varies tremendously with time due to the large scale turbulence and wind effects. The standard deviation of these temperature measurements is large; it ranges from 10-20% at the lower elevations up to 30-50% at stations 6-10 m above the pool surface (Gregory, et al, 1989, and Schneider and Kent, 1989). In an attempt to reduce the effects of the wind, the temperature data have been conditionally sampled (a discussion of the sampling method is given in Schneider and Kent, 1989). Results of this sampling are shown in Figure 3 for the same fires as shown in Figure 2. The conditionally sampled data, which is nominally from periods of 'low winds', has improved consistency. The location of the peak is less than the height of most rail cars or RAM shipping containers.

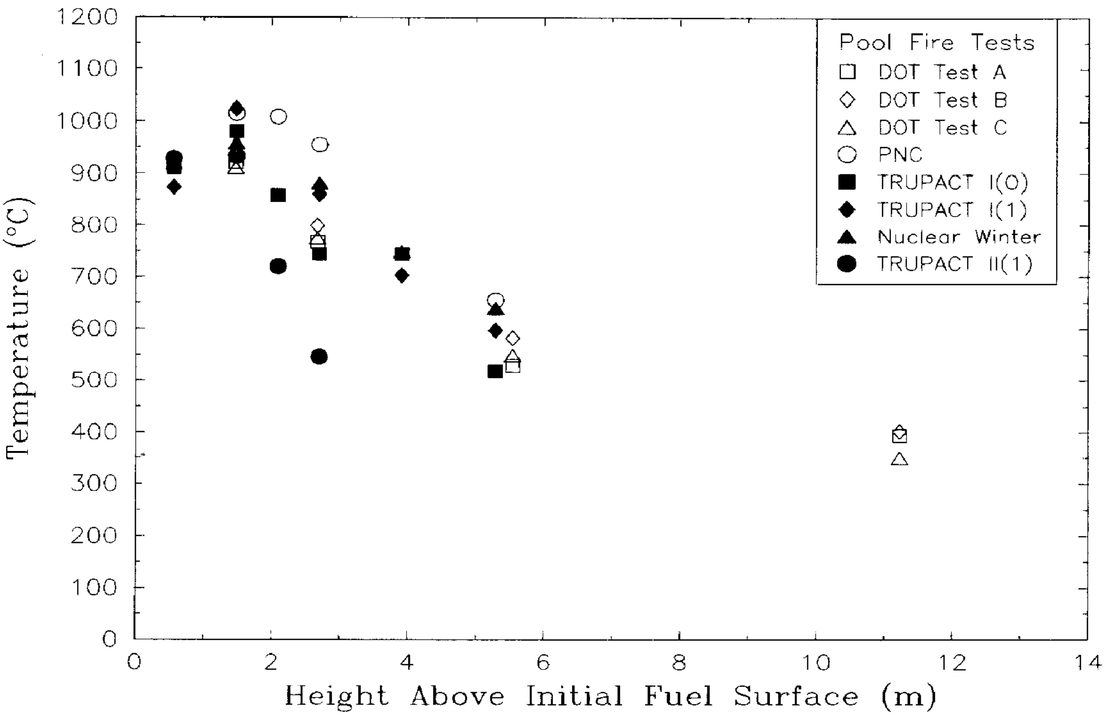

Figure 2. Average Flame Temperatures

One way to compare the temperature data with similar data from other experiments is to scale the elevation with respect to the heat release rate of the fire as outlined by McCaffrey, 1979. For the Sandia fires, the heat release rate is approximately 500 MW. The peak of the curve for the Sandia tests occurs at a scaled elevation of approximately 0.01.

Measurements made in fires show that the average temperature varies significantly with elevation and the variance at any location is large (Bainbridge and Keltner, 1988, Gregory, et al, 1987 & 1989, Schneider and Kent, 1989). The variation of the temperature with elevation helps to explain why the relatively simple models that use a uniform temperature assumption could give results that differ from measurements in a fire. It should be noted that while the temperature variation will affect the heat flux predictions, incorporating even the limited amount of available data concerning the temperature distribution in pool fires into the models would be a tremendous complication.

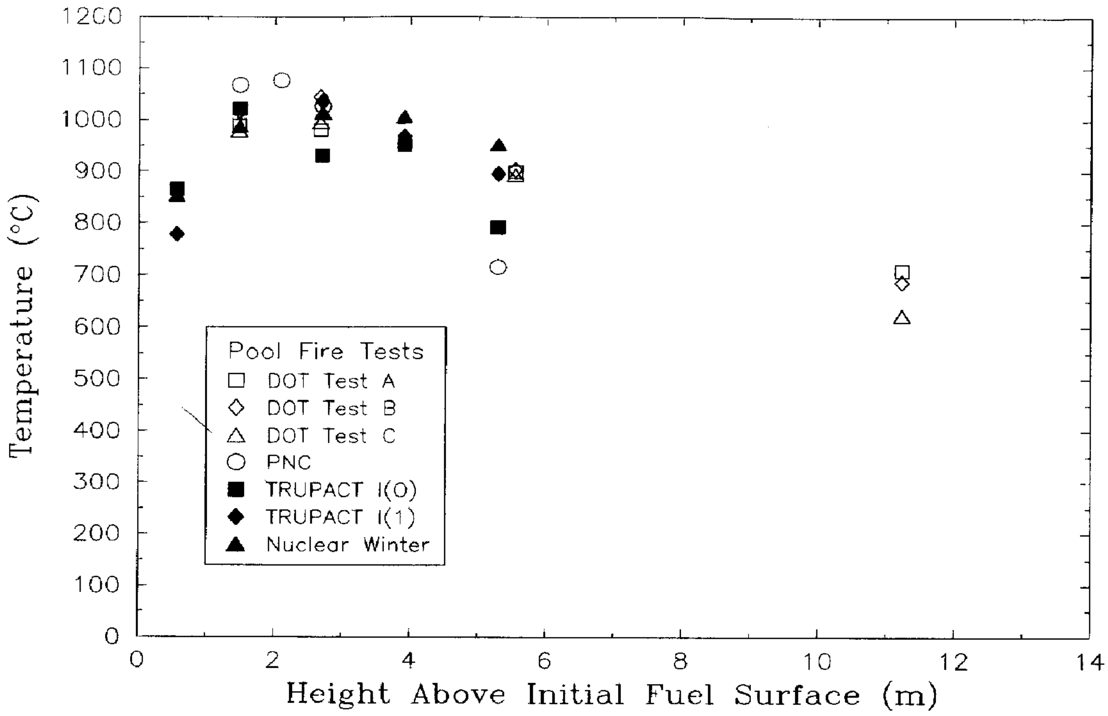

Figure 3. Conditionally Sampled Flame Temperatures

The heat flux to the test unit depends on the temperature distribution in the "region of influence" around the unit. The size of this region depends on the extinction coefficient in the flame zone. Using an extinction coefficient of ~1/m (Longenbaugh and Matthews, 1988), the models would predict that the maximum heat flux would occur on the top of a horizontal tank. Heat flux distributions have been measured with the 1.4 m diameter calorimeter mounted 1 m above the pool surface in the 9 m by 18 m JP-4 fires conducted at Sandia. The average total heat flux (i.e., the measured hot wall heat flux plus the emitted flux, Eqn.1) is consistently highest on the bottom of the calorimeter (000 degrees). These values are listed below along with the conditionally sampled values (Bainbridge and Keltner, 1988).

Table 1: Heat Flux Distribution on the 1.4 m Cylindrical Calorimeter

Angular Station (degrees)	All Data		Conditionally Sampled	
	kW/m^2	% std.dev.	kW/m^2	% std.dev.
000	113	18	107	18
090	71	39	94	21
180	60	42	76	27
270	99	23	96	20

For a number of years there has appeared to be a discrepancy between the values of the heat fluxes measured in large fires with heat flux sensors and the values inferred from test results with large objects that are engulfed in the fires. Some data are summarized in Table 2; brief descriptions are given in the following paragraphs.

Table 2: Heat Flux Levels for Large and Small Objects (kW/m^2)

Reference	Large Objects	Small Sensors
Bader (1965)	NA	154
Anderson, et al (1974)	100,109	139-218
Gregory, et al (1989)	136	161
Wachtell and Langhaar (1966)	85	NA

From measurements made with "heat meters" in a number of fires, Bader (1965), determined that the heat transfer in a fire could be represented by a blackbody source with a temperature of 1283 K which gives a heat flux of 154 kW/m^2. Measurements made with water cooled Gardon gages in fire tests of rail tank cars were ranged from 139 kW/m^2 to 218 kW/m^2 (Anderson, et al, 1974). Gregory, et al (1987 and 1989), obtained average peak heat fluxes of approximately 165 kW/m^2 with 10 cm and 20 cm OD steel calorimeters in a series of three fires in a 9 m x 18 m pool.

A large cylindrical calorimeter has been used in a number of large fires that have been conducted at Sandia National Laboratories. Gregory, et al (1987 and 1989), describe the results of a series of three fires in a 9 m x 18 m pool with a 1.4 m OD x 6 m long steel calorimeter that weighed 10 tons. For these tests the peak heat flux on the bottom was approximately 136 kW/m^2. Heat fluxes to large objects in other large fire tests were reviewed in the appendix of the 1987 report: for a lead filled steel cask, that was ~0.9 m x 1.2 m x 1.5 m, the estimated heat flux on a vertical, finned wall was ~85 kW/m^2 (Wachtell and Langhaar, 1966); for a 3 m OD x 18 m long railcar filled with propane the estimated average flux to the wetted surface was 100 kW/m^2 (Anderson, et al, 1974); for a 1/5 th scale version of the tank car the flux was 109 kW/m^2.

As shown above, the average heat flux to a test object that is "physically large" and "thermally massive" is significantly lower than that measured by small sensors. The heat flux levels as a function of the surface temperature of the calorimeter are shown in Figure 4 for measurements made with 10 cm, 20 cm, and 1.4 m diameter, cylindrical calorimeters. These are the average, hot wall heat fluxes measured in a series of three tests (Gregory, et al, 1987 & 1989). The size effect is apparent in that the peak heat flux to the 1.4 m calorimeter is approximately 20% lower than the fluxes to the smaller calorimeters. The thermal mass effect (surface temperature) is suggested by the fact that all three curves converge at higher temperatures, when the surface temperature no longer influences the surrounding fire

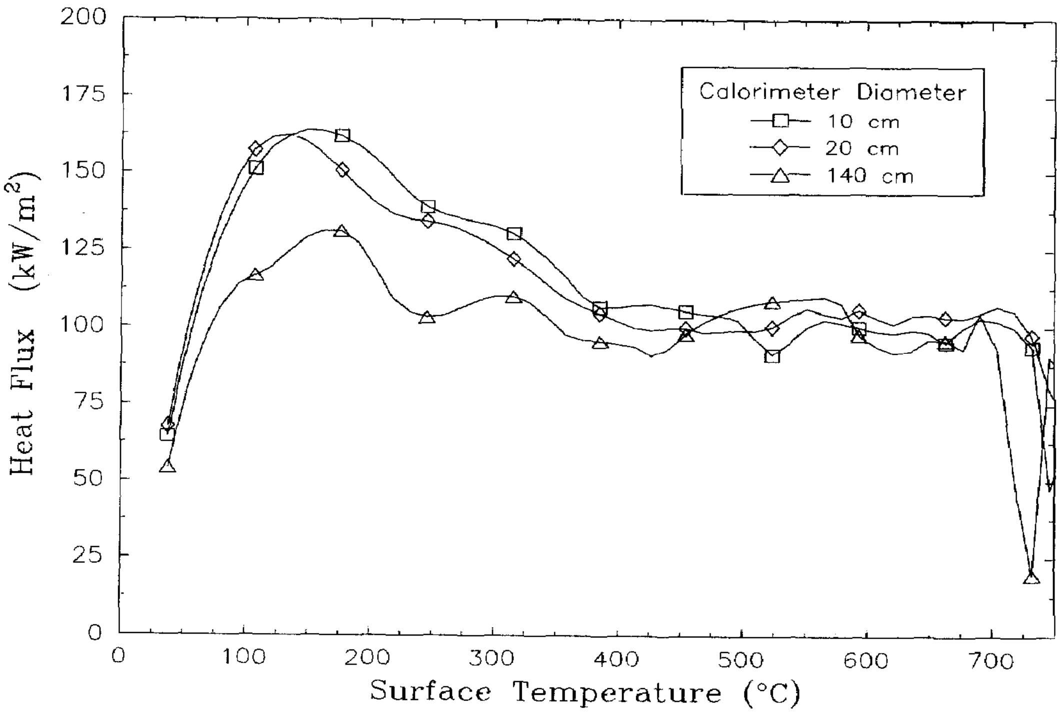

Figure 4. Effect of Calorimeter Size and Mass on the Heat Flux

Previous modeling efforts do not explain the experimentally observed reduction in heat flux to a large thermally massive test object. Mansfield, 1983, had postulated that when the test unit is a large heat sink, localized cooling could result in radiative shielding by the cooler boundary layer and reduce the heat transfer to the unit. This interaction between the flame and the test object has not been modeled in previous works in this area which assume the flame volume is at a uniform temperature.

ANALYTICAL MODEL

In order to try and provide an understanding of the experimentally observed reduction in heat transfer to a thermally massive object, an analytical model was constructed. Many simplifying assumptions were made and their validity checked in order to try to isolate the important physics of the problem. The model consisted of a vertical flat plate at constant temperature completely engulfed by flames of large thickness (Figure 5). The flow field of hot combustion gases and soot upward past the plate was modeled; convective heat transfer between the fluid and the plate was not modeled. Only radiative heat transfer normal to the plate surface (i.e., transverse to the flow direction) was modeled. Viscous boundary layer effects and buoyancy are neglected (uniform fluid velocity). The fluid was modeled as non-conducting with no turbulent mixing and no concentration gradients. The combustion products were treated as dry air with uniform soot distribution and constant properties.

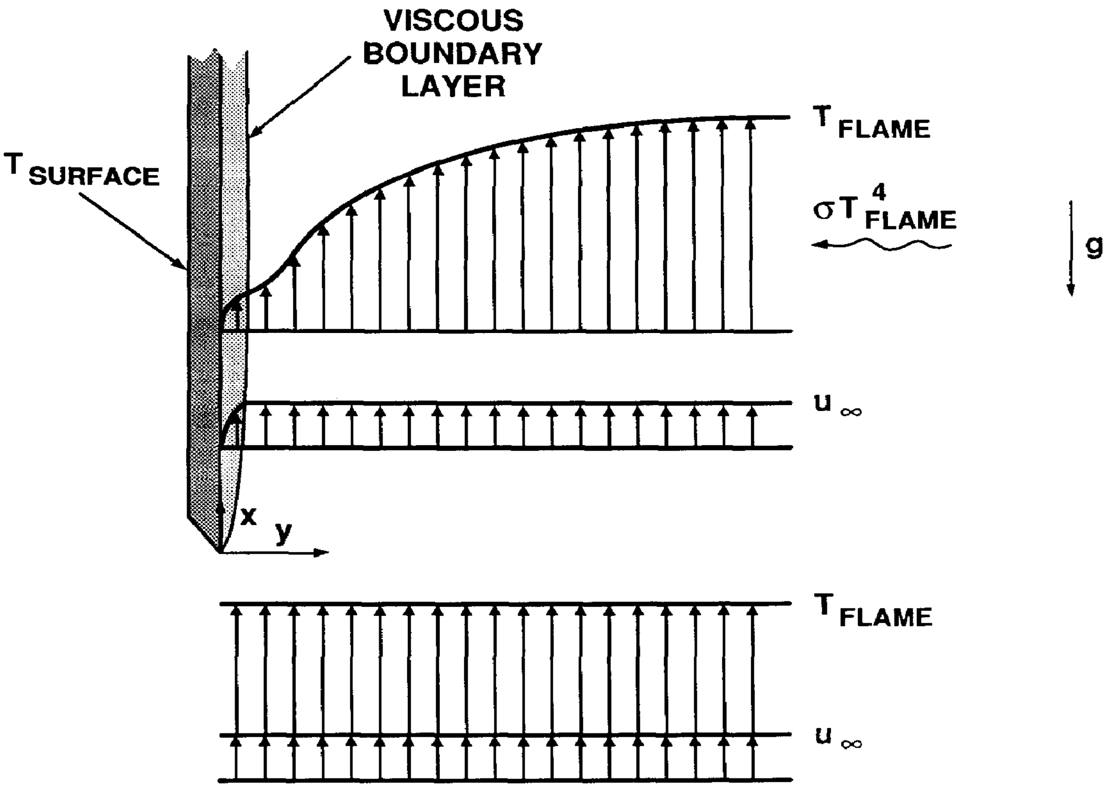

Figure 5. Model Geometry

Thermal radiation to the surface of the plate from the combustion products was modeled assuming 1-D gray gas radiative transfer normal to the surface. The plate was assumed to be a blackbody absorber and emitter. This assumption is most valid during the early stages of the fire when soot, which is close to black, is deposited on the cold surface. As the surface heats up the soot burns off, the emissivity decreases to measured values of 0.8-0.85 when the surface is coated with Pyromark Black paint. Because the interaction between the surface and the flames would be most pronounced for colder surfaces, the assumption that the surface is black appears to be reasonable for the calculations. The far field boundary condition was applied at a distance away from the surface at which any thermal radiation originating from the surface had been attenuated by 99%. For a typical soot extinction coefficient of 1 m^{-1}, the far field boundary condition was applied at a distance of 3 meters out from the surface. The combustion products in the region of interest were modeled with and without a combustion source term. For the cases involving a source term, the far field boundary temperature was calculated assuming zero temperature gradient at the boundary. For the cases without a source term, the far field boundary temperature was set equal to the assumed flame temperature.

Under the above assumptions, the steady-state energy equation reduces to:

$$\rho c_p U_\infty \frac{\partial T}{\partial x} = \frac{\partial}{\partial y} (q_r) + S''' \qquad (2)$$

where q_r is the local radiative flux, S''' is the local volumetric heat source, ρ is the gas density, c_p is the gas specific heat, T is the local gas temperature (a function of x and y), x is the coordinate along the plate surface, y is the coordinate normal to the plate surface, and U_∞ is the uniform freestream velocity. Equation 2 implies conservation of energy at any point throughout the flow field. For a control volume, radiant energy flows from the far field towards the plate in a direction normal to the plate, there is a volumetric heat source, and convective energy flows through in a direction parallel to the plate. A two-flux formulation (Siegel and Howell, 1981) of the 1-D gray gas radiative transport equation was used to solve for the local radiative flux, q_r, based on the temperature field. It should be noted that due to the heavy soot loading the large pool fires of interest are absorption dominated and scattering can be neglected (Longenbaugh and Matthews, 1988).

Calculations were carried out on a 21 by 21 grid using a semi-implicit finite difference scheme to solve (2) and the radiative transport equation. More details of the model may be found in Nicolette and Larson (1990).

While the above model is admittedly simplistic, it does allow us to investigate the magnitude of the effect which a cold surface can have on the heat flux by affecting the local flame temperature. The purpose of the model is not to predict exactly the radiative heat flux from the flames to the plate, but rather to obtain some understanding of the mechanisms involved and how they alter the flux from the far field levels. It is, therefore, an appropriate model for scoping calculations of this nature.

MODEL RESULTS

A large, cold object can significantly alter the local fire environment through radiative cooling of the combustion products (i.e.,gases and soot). The upward flow of the combustion products results in a coupling between the radiation field and the convective flow that produces a radiation boundary layer. Figure 6 shows the development of the radiation boundary layer for the case of a zero source term which is the easist to understand because the far field boundary temperature is equal to the flame temperature. The radiation boundary layer influences a region several meters into the flow whereas the viscous boundary layer is only a few centimeters thick.

A typical flame environment is not constant in time or in space. For the calculations, representative values of parameters such as freestream velocity, gas temperature at the leading

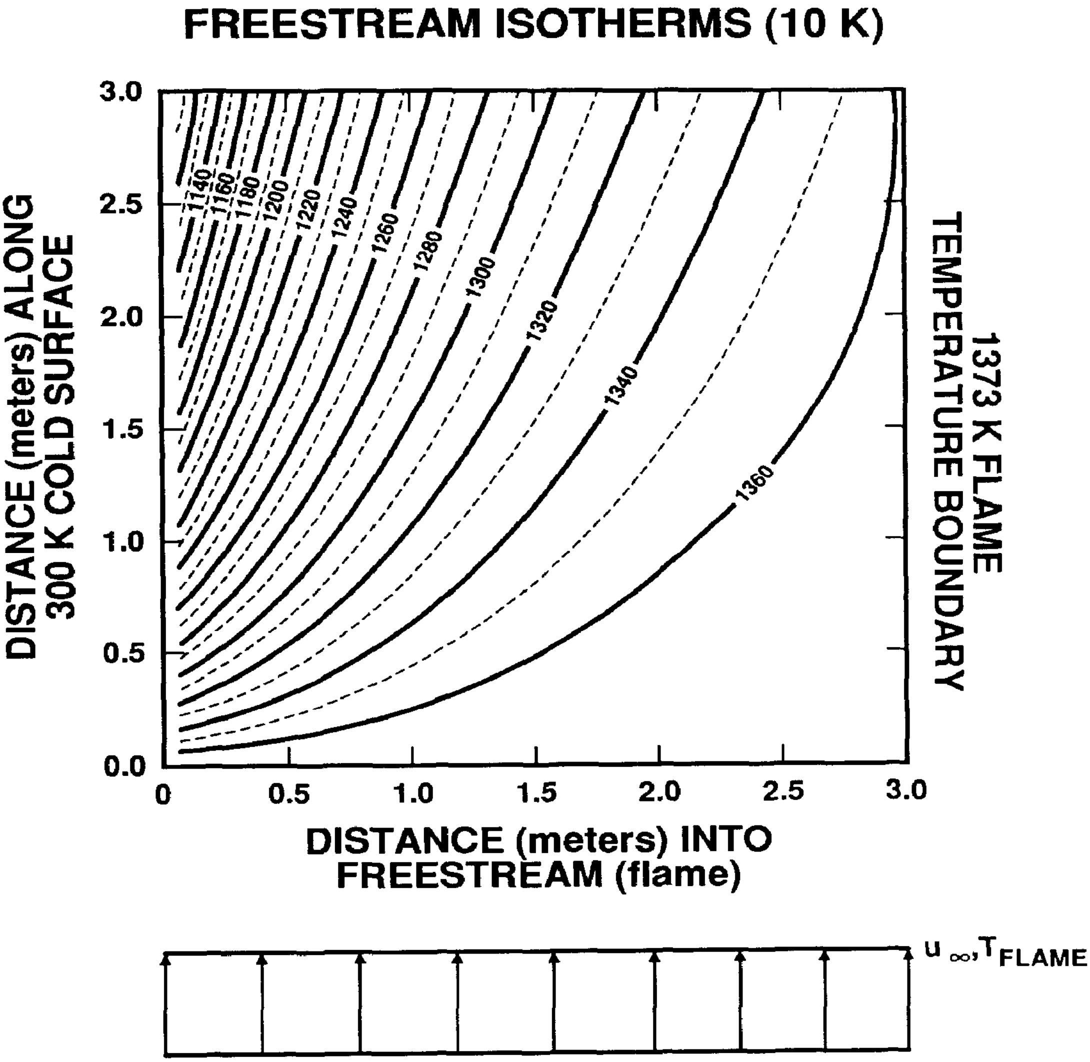

Figure 6. Radiation Thermal Boundary Layer Development Along a Large, Cold Plate

edge of the plate, and absorption coefficient were selected. The selected values are shown in the figures. Uniform combustion source terms and an Arrhenius type source term, which was exponential in temperature, were used. The sensitivity of the results over appropriate ranges of these parameters will be briefly discussed at the end of this section.

Figure 7 shows the calculated radiative heat flux to the plate as a function of plate surface temperature at a location 1 m from the leading edge of the plate. This is representative of the characteristic length of the larger calorimeters used in the experiments (Gregory, Keltner, and Mata, 1989). The effect of the combustion source term is to heat up the combustion gases as they flow upward along the plate, resulting in higher heat fluxes to the surface. However, the presence of the cold surface acts to cool the combustion products as they flow upward. This has the tendency to reduce the radiative heat flux reaching the surface. It can be seen that at 1 meter from the leading edge the presence of a cold surface can reduce the incident radiative heat flux by up to 20% for the parameters selected.

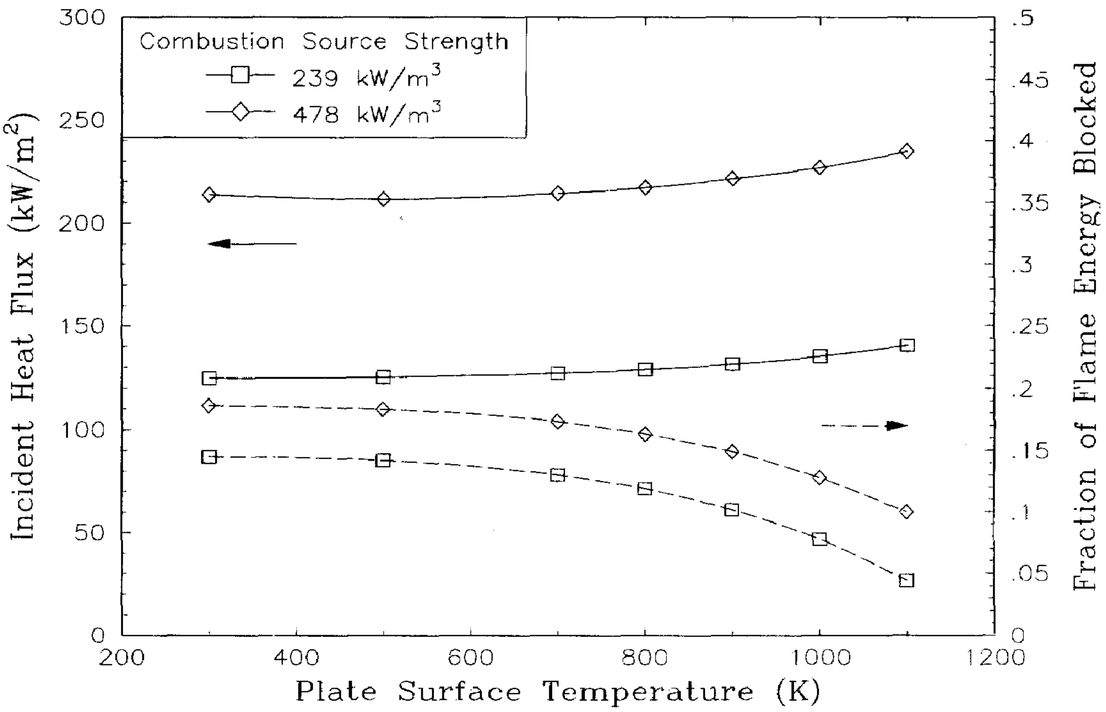

Figure 7. Influence of Cold Surface on the Incident Radiative Heat Flux and the Fraction Blocked: Uniform Heat Source

These results can also be expressed in terms of the fraction of flame radiation which is effectively blocked by the cooled participating medium (due to the presence of a cold surface). This fraction is defined as:

$$\text{Fraction Blocked} = 1 - \frac{\text{Flame radiative heat flux reaching surface}}{\text{Far field flame radiative heat flux } (\sigma T^4_{flame})} \qquad (3)$$

where σ is the Stefan-Boltzmann constant, T_{flame} is the far field flame temperature (uninfluenced by the cold surface), and a flame emissivity of 1.0 has been assumed. This fraction can also be viewed as:

Fraction Blocked = 1 - Effective flame emissivity (4)

where the effective flame emissivity is less than unity as a result of the radiative cooling of the combustion products by the cold surface. This effective flame emissivity is not solely a function of the combustion products, but is also a function of the length, shape, and temperature of the cold surface. Results in terms of the 'fraction blocked' are also shown in Figure 7 on the right hand ordinate. Note that small surfaces will not show much reduction in the incident radiative heat flux because they are not large enough to influence a significant portion of the fire environment. Similarly, a relatively warm surface will have little effect on the incident radiative heat flux because it can not cool the combustion products to any significant extent.

The effect of a temperature dependent source term (first-order Arrhenius based) is shown in Figure 8 for a location 1 meter from the leading edge of the plate. For this calculation, the incident radiative heat fluxes were reduced by as much as 30% when a large, cold surface was present.

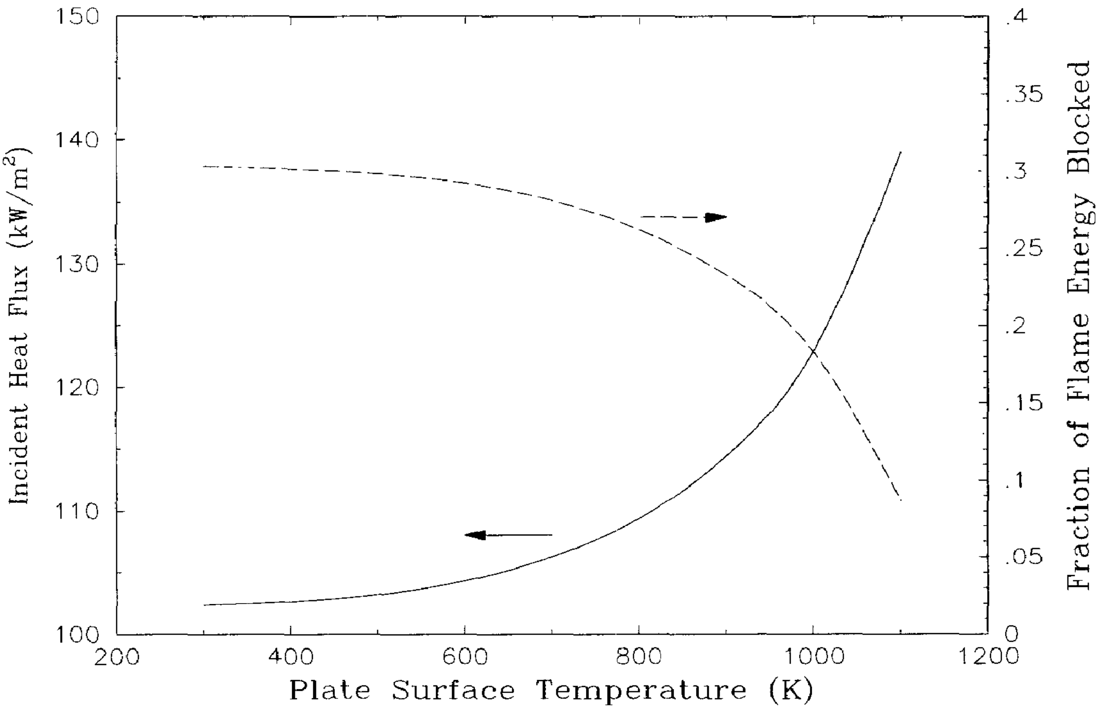

Figure 8. Influence of Cold Surface on the Incident Radiative Heat Flux and the Fraction Blocked: Arrhenius-type Heat Source

The sensitivity of the model to a number of parameters and assumptions has been assessed. The sensitivities are listed in decreasing order of importance:

1) changes in the extinction coefficient,
2) source term levels on the heat flux,
3) changes in the free stream velocity,
4) source term levels on the fraction blocked,
5) modeling of convective boundary layer effects, and
6) modeling of turbulent effects.

Further results and details of the model sensitivities can be found in Brown, et al (1990), or Nicolette and Larson (1990).

SUMMARY

This paper has attempted to show:

1) the need to make measurements of both the temperature in the fire and the heat flux to the test unit;
2) how simple models that assume a uniform temperature in the fire volume can give misleading heat flux predictions, especially during the early part of the test when the test unit is cold;
3) how a model that accounts for the influence of a physically large, thermally massive test unit on the local conditions in a fire can give predictions that are inline with experimental results. (Note that this model is not designed for apriori predictions.)

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