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MODEL FOR THE PREDICTION OF RADIANT HEAT TRANSFER TO A HORIZONTAL CYLINDER ENGULFED IN FLAMES

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ABSTRACT

A model has been developed which estimates the radiant heat transfer to a long horizontal cylinder engulfed in flames. The model estimates the various surface-to-surface and surface-to-volume exchange factors as a function of position around the circumference of the cylinder. The present model considers the two-dimensional case of an infinitely long cylinder, however, only slight modifications are necessary to model the full three dimensional case. The numerical procedures used to carry out the integrations necessary in determining the exchange factors are presented.

This model was developed as part of a study concerned with thermal protection systems for railway tank cars carrying hazardous materials. In some accident situations, a tank car may be engulfed in flames, and it can be shown that the total heat flux to the tank is primarily due to thermal radiation. Radiant heat transfer calculations using this model were performed for two flame sizes with dimensions approximating those of full and fifth scale fire tests conducted on real tanks. The results of these calculations are presented with the flame temperature and absorptivity as parameters.

NOTATION

A - surface area of zone.

D - diameter of the cylinder.

e - exponent.

E - emissive power.

h - height dimension of the fire.

J - radiosity.

K - extinction coefficient of the combustion gases for absorption (scatter neglected); also commmonly known as the mean absorption coefficient.

L - beam length.

n - normal vector.

$\dot{Q}$ - heat transfer rate.

r - radius of cylinder.

R - length of a radius vector between surface elements.

V - volume.

w - width dimension of the fire.

α - surface absorptivity.

δ - Kronecker delta function.

ϵ - surface emmisivity.

Υ - angle from the top of the cylinder to a surface element.

ρ - surface reflectivity.

σ - Stefan-Boltzmann constant.

Θ - angle between the normal vector at a surface element and the radius vector connecting two surface elements.

τ - transmissivity between two surface zones.

subscripts:

e - enclosure surface zone.

g - surrounding gas volume.

i,j - surface zones.

k - volume zone.

m - mean value.

INTRODUCTION

This study is concerned with the variation in radiant heat transfer around the circumference of a long horizontal cylinder engulfed in flames. In the analysis, the cylinder is considered to be of infinite length and is surrounded by radiating combustion gases. It is assumed that the combustion gases occupy a space which can be represented by a rectangle, and that this rectangle is enclosed by walls which represent the infinite stagnant surroundings. With the assumption that the cylinder is of infinite length, the problem becomes two dimensional in the sense that the surface-to-surface, and surface-to-volume exchange factors vary only with the position around the circumference of the cylinder, and not with the axial position along the cylinder.

The rectangular enclosure represents the interface between the combustion gases and the infinite surroundings, and its dimensions approximate those of a pool fire in an infinite stagnant atmosphere. The walls of the enclosure are assumed to act as black surfaces and the cylinder as a gray-Lambert surface. The combustion gases are assumed to act as a gray medium which can be represented by an effective flame temperature and flame absorption coefficient.

Ultimately, the results of this study will be used as part of a mathematical model of a rail tank-car in a pool fire environment. By considering the tank as a long cylinder in a cross flow, it is possible to estimate the convective heat transfer contribution to the total heat transfer. With this approach it can be shown that thermal radiation is the primary mode of heat transfer to the outer surface of the tank, and due to the nature of large pool fires, the rate of radiant heat transfer may vary considerably around the circumference of the tank.

The simplest model of the radiant heat transfer would treat the flame as a black surface surrounding the cylinder radiating at some effective flame temperature. This would be the case for very sooty flames where only a small proportion of the thermal radiation is transmitted. If on the other hand the flame is acting as a absorbing/transmitting medium, the rate of radiant transfer will vary considerably around the circumference of the tank. This variation will be due to both the temperature distributions in the flame, and the geometry of the pool fire. In the present analysis the geometry of the pool fire is considered to be the dominant factor in the variation in radiant transfer around the circumference of the cylinder. No attempt has been made to represent the temperature distribution that exists in a real pool fire.

Results have been obtained using this model for two cases; the scales of which attempt to represent a full scale rail tank car, and a one fifth scale rail tank car with flame dimensions approximating those encountered in flame tests conducted on real tank cars, (1) and (2). The results of the present model are presented with the flame temperature and the mean absorption coefficient for the flame as parameters.

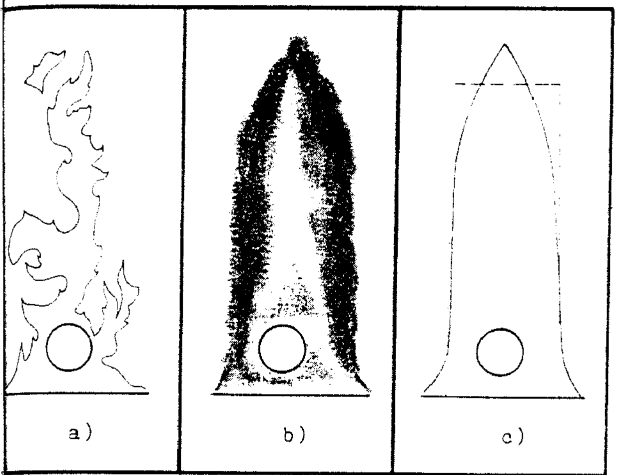
Since reliable measurements of radiant transfer were not taken in either of the fire tests conducted, there are no means of accurately verifying the predictions of the present model. However, radiant heat transfer estimates of the model do fall in the range of heat fluxes which were predicted from the temperature history measurements taken in the fire tests. In addition, the results of this study show some interesting aspects of the scaling effects and the effects of flame temperature and radiating properties.

No studies concerned with the variation in radiant heat transfer around the circumference of a cylinder engulfed in flames could be found by the present authors. However, several studies concerned with the properties of pool fires and the resulting radiant flux to external targets and to the pool surface have been undertaken. These include such studies as those conducted by Modak (3), Orloff (4) and Hertzberg (5).

MODEL DESCRIPTION

As mentioned, the basic model of the radiant heat transfer involves an infinitely long, horizontal cylinder inside a rectangular enclosure. The enclosure is filled with hot combustion gases whose radiating properties can be represented by an effective flame temperature and absorption coefficient.

Figure 1a represents an instantaneous cross section through a large diameter pool fire. The shape shown is representative of a buoyancy-driven turbulent diffusion flame. It has been assumed for the present analysis that the presence of the cylinder does not affect the shape of the flame appreciably. Figure 1b is a time averaged view of the same flame showing the approximate shape expected



gure 1a Instantaneous section through the pool fire.
1b Time averaged view of the fire.
1c Comparison between idealized rectangular shape and the time averaged shape of the pool fire.

a large diameter pool fire. Experiments nducted show that for large diameter pool res (D > 1 m.), the height to diameter tio of the flame is fixed for most hydro-rbon fuels and the value of this ratio is proximately 2. Figure 1c shows the time eraged flame shape compared to the assumed ape, which in the present model is a re-angle.

With the geometry specified, it was essary to divide the system into zones for purpose of doing the radiative heat lance. In the present analysis the walls the enclosure and the pool surface were nsidered as a single surface zone, the mbustion gases as a single volume zone and surface of the tank was divided into a ies of surface zones around the cumference of the tank. The results ported here were obtained using cylinder face zones which spanned 20 degrees of . Each zone has a single uniform perature and uniform radiating properties.

After the zones were defined it was essary to make certain assumptions relat-to the radiative properties of the var-s zones. The walls of the rectangular losure represent the interface between the mbustion gases and the infinite surround-s. Since what ever thermal radiation sses this boundary does not return, the ls of the imaginary enclosure act as black faces. The pool surface which has been luded as part of the enclosure zone will act as a black surface, however, it has n assumed that the true radiative proper-s can be approximated by those of a black face. The temperature of the pool is umed to be equal to the temperature of the inite surroundings. An improvement to the sent model would be to consider the pool face as a separate zone with its own perature and radiating properties.

It was assumed that the surface of the cylinder acts as a gray-Lambert surface. This was assumed because the surface of the tanks used in the fire tests, and those found in the railroad industry, can be represented as gray-Lambert to a good approximation. For the purposes of this study the cylinder wall temperature is considered to be constant and uniform, however, the analysis presented can account for different temperatures of each cylinder surface zone as in the case of a real tank car in a pool fire environment, where the tank wall temperatures will vary with time as heat is added to the system.

The combustion gases of large pool fires typically contain large quantities of soot due to the oxygen starved nature of the flame and therefore can be represented as a gray medium to a good approximation. This is because the soot particles act as black bodies which absorb the thermal radiation at all wavelengths, thus tending to make the net radiative properties of the gas similar to that of a gray gas.

The radiative heat balance accounts for the fluxes taking place at the various surface and volume zones. As a result of the radiant heat balance, each zone will have an equation relating its radiosity with the emissive power and radiating properties of the other zones. The general radiant energy balance equation can be shown to have the following form.

$$\sum_j (A_{ij} - \delta_{ij} A_i/\rho_i)\, J_j = - A_i \epsilon_i E_i/\rho_i - \sum_k A_{ki} E_k \qquad (1)$$

The surface exchange factors can be calculated from the following expressions. The basis of these and the related equations can be found in reference (6).

$$A_{ij} = \int_{A_i}\int_{A_j} \frac{\cos\Theta_i \cos\Theta_j \, e^{\int -K\, dR}}{\Pi R^2}\, dA_i\, dA_j \qquad (2)$$

$$A_{ik} = \int_{A_i}\int_{V_k} \frac{K \cos\Theta_i \, e^{\int -K\, dR}}{\Pi R^2}\, dA_i\, dV_k \qquad (3)$$

Because of the geometry of the cylinder surface, each cylinder surface zone only sees part of the enclosure walls and part of the surrounding gases. The cylinder surface zone does not see any portion of itself or any of the other cylinder surface zones. This of course is due to the convex nature of the cylinder. With this in mind it can be shown that only one set of exchange factors need be calculated using equation (2) or (3), the remaining exchange factors can be calculated using the following relation.

$$A_{ig} = A_i - A_{ie} \qquad (4)$$

Equation (4) basically states that all the radiation leaving surface i goes to either the enclosure walls or the surrounding gas. In the present study, equation (2) was used to find the surface-to-surface exchange factors, and then using equation (4) the surface-to-volume factors were evaluated.

With this simple case, it is possible to arrive at the various radiosities by inspection without having to solve equation (1). For the black enclosure walls and the gas volume, the radiosity is equal to the emissive power and therefore it can be shown that the net radiative transfer to a tank surface zone is equal to,

$$\dot{Q}_i = A_i \epsilon_i E_i - \alpha_i (A_{ei} E_e + A_{gi} E_g). \qquad (5)$$

The net radiative transfer from the tank surface is equal to its emissive power minus the absorbed radiation from the gas and enclosure zones. Thus, the only variable in the preceeding analysis requiring extensive calculations is the surface-to-surface exchange factor.

The calculation of the surface-to-surface exchange factors involved the coordinate system shown in Figure 2. Since the cylinder is infinitly long, there is no variation in the radiative constants in the axial direction. For an infinitesimal area element on the cylinder surface located by the radius r, and the angle from the top of the cylinder, and with the flame absorption coefficient considered constant, the integral of equation (2) becomes,

$$\frac{A_{ie}}{dA_i} = \int_{A_e} \frac{\cos \theta_i \cos \theta_j \, e^{-KR}}{\Pi R^2} \, dA_j \qquad (6)$$

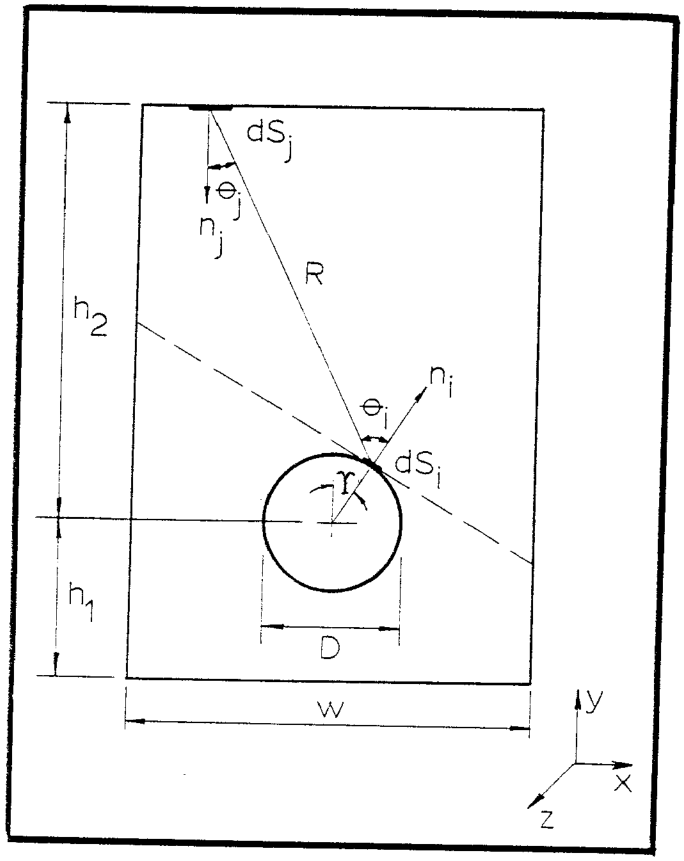


Figure 2 Coordiate system used for surface-to-surface exchange factor calculations.

The quantity on the left side of equation (6) is actually the transmissivity from the surface element dA to the walls of the enclosure. The transmissivity and the surface exchange factor are related by the following expression.

$$A_{ij} = A_i \tau_{ij} \qquad (7)$$

where,

$$\tau = e^{-KL_m}$$

In many studies of radiation with an absorbing medium the concept of a mean beam length is incorporated. The mean beam length is a quantity representing the weighted-average distance that a "beam of light" must travel through the absorbing gases, and is usually assumed to be a function of geometry alone and is therefore calculated directly. However, with this approach certain assumptions are necessary which apply only in the case of small KR values. In the present study, the dimensions involved and the range of K values considered makes that approach inappropriate because the mean beam length becomes a function of both geometry and the radiating properties of the gas.

Equation (6) is integrated over the enclosure surface extending to infinity in either axial direction from the point on the cylinder surface. The limits of integration cover that portion of the enclosure surface which is above a tangent plane to the point of interest on the cylinder surface. This is due to the fact that the cylinder surface element only sees the portions of the gas and the enclosure walls which are above it. For the three dimensional case of a short cylinder, the integration limits would extend only to the ends of the flame filled enclosure, and the enclosure walls would now include the ends of the rectangular box enclosing the short cylinder.

For the case of the infinitly long cylinder, the calculated value of the exchange factor for a given surface element on the cylinder will apply along the entire length of the cylinder due to the two dimensional nature of the system. Therefore, the above mentioned procedure need only be carried out for a number of different circumferential positions on the cylinder surface. For the three dimensional case, the integration would have to be carried out for each cylinder surface element along the axis of the cylinder.

RESULTS AND DISCUSSION

As mentioned, results have been obtained for two scales which approximate those of full and fifth scale fire tests conducted on real tank cars. The dimensions used for the calculations are shown in Figure 3. The results obtained from using this model are presented in Figures 4 to 11.

Figures 4 and 5 show the variation in transmissivity around the cylinders for varying flame absorption coefficients. Figures 6 and 7 show the variation in the resulting heat transfer to the cylinder surfaces when the cylinders have a temperature of 15.5 degrees C. Figures 8 and 9 again show the heat transfer rates, this time when the cylinders are at a uniform temperature of 115.5 degrees C. Figures 10 and 11 present the heat transfer distributions for a fixed flame absorptivity but with varying flame temperatures. For all the radiant heat transfer calculations the cylinder surface emissivity was 0.85.

Figures 4 and 5 indicate that the transmissivity can vary considerably around the circumference of the cylinder. The radiant heat transfer calculations are based on these factors and therefore the radiant heat transfer also varies around the cylinder.

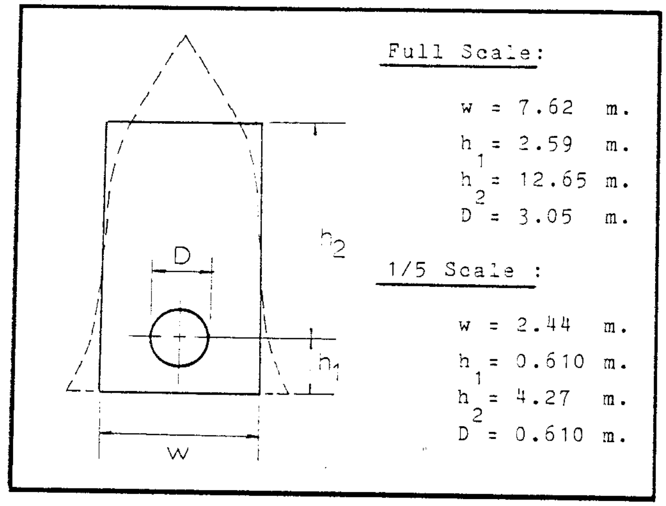


Figure 3 Enclosure and fire dimensions used in model calculations.

Figures 6 to 9 show the predicted heat transfer rates for different K values and two different cylinder wall temperatures. These results of course apply for a steady state situation where the tank wall temperatures are not changing with time. For a real tank partially filled with lading, this steady state assumption would not apply.

Figures 10 and 11 present the heat transfer predictions with various flame temperatures. Of course, results indicate that the heat transfer rates are very strongly related to the flame temperature, and therefore show the importance of knowing the mean effective radiating temperature of the fire.

No attempt was made to account for the affects of convection in the present analysis. However, by considering a cylinder in a cross flow, the convective heat transfer coefficient, and its variation around the circumference, can be approximated. This approach to calculating the convective contribution to the total heat transfer indicates that it is small (less than 20 %) for the range of flame absorptivities expected in real fire conditions.

The most interesting results are the apparent scaling effects. The studies that were cited attempted to model the effects of an engulfing flame on a full and fifth scale tank car, and although the geometry of the tanks were similar the fire conditions were not. Due to the non similarity of the two flames considered, the variation in the predicted radiant heat transfer rates was more evident on the cylinder representing the fifth scale tank. This makes comparison between the two studies difficult, if not impossible. For true similarity, in terms of the fire conditions, it would have been necessary to properly scale the dimensions of the tank and the fire, as well as the radiating properties of the fire. This of course would not be a simple task.

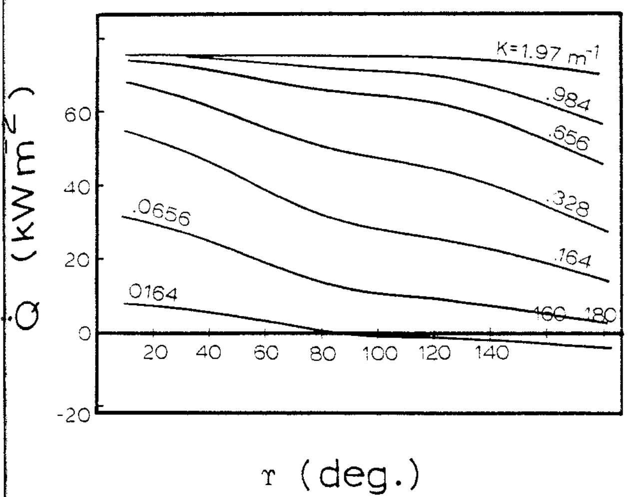


Figure 8 Variation in radiant heat transfer to cylinder surface with varying flame absorption coefficient: full scale

Cylinder surface temperature at 315.5 deg. C.
Flame temperature at 870 degrees C.
Infinite surroundings at 15.5 degrees C.

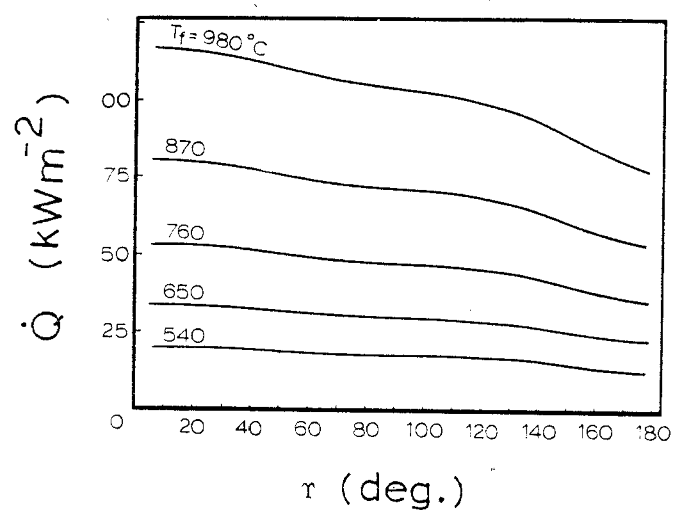


Figure 10 Variation in radiant heat transfer to cylinder with flame temperature as a parameter: full scale.

Flame absorption coefficient equals 0.656 1/m.
Cylinder wall and infinite surroundings at 15.5 degrees C.

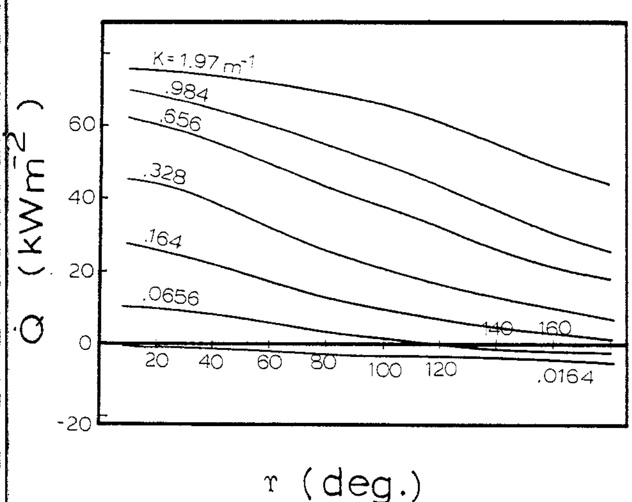


Figure 9 Variation in radiant heat transfer to cylinder surface with varying flame absorption coefficient: fifth scale

Cylinder surface temperature at 315.5 deg. C.
Flame temperature at 870 degrees C.
Infinite surroundings at 15.5 degrees C.

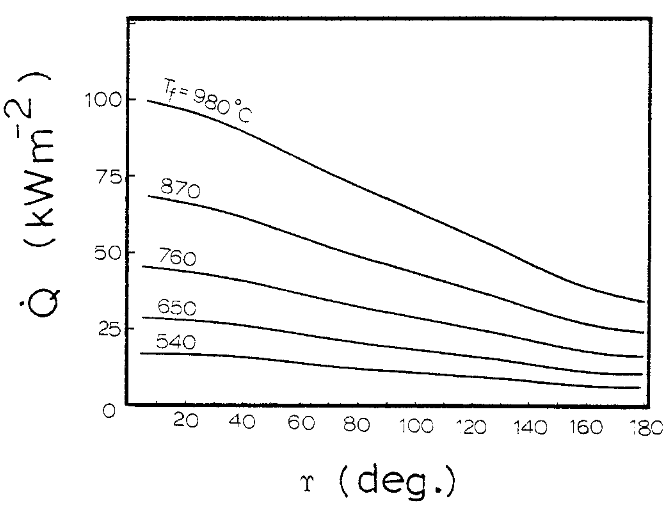


Figure 11 Variation in radiant heat transfer to cylinder with flame temperature as a parameter: fifth scale.

Flame absorption coefficient equals 0.656 1/m.
Cylinder wall and infinite surroundings at 15.5 degrees C.