Transient Heat Transfer for Layered Ceramic Insulation and Stainless Foil Fire Barriers

Time-varying flame boundaries to heat transfer applications is a common application in energy and safety related systems. Many such systems could be tested and analyzed; however, the number of thermophysical parameters involved and the possibilities of boundary conditions are endless. Many papers have justified numerical and analytical models based on computational efficiency, with the objective of eventually applying those efficient techniques to real problems. There are, however, a number of standard test situations related to such systems that may be readily studied. The purpose of this study is to be able to optimize a coarse numerical model and range of thermophysical parameters that represent the physics of the real problem in a standard test situation.

The applied thermal problem involves fire barrier safety in the design of buildings, such as hospitals and schools. Public buildings are often primarily constructed of concrete with significant gaps between sections to allow the concrete to expand and contract, due to climate changes or transients. In seismically active regions of the world, gaps may be up to several meters across, requiring some type of thermal fire barrier designed to prevent a fire from spreading for some time. A radiative/conductive fire barrier is first tested with an ASTM standard fire. A numerical model is then applied, which is an optimally coarse finite difference/finite volume formulation applied to the standard transient conduction energy equation with radiative heat flux (Ozisik, 1973) and to the radiative transfer equation (Su, 1993). The numerical model is able to predict thermal performance of the test system, illustrating the utility of the coarse grid model in engineering applications.

Introduction

One standard test method used for transient temperature of concrete expansion joint fire barriers is specified in ASTM Standard E 119, "Standard Methods of Fire Tests of Building Construction and Materials" (1983). The test standard prescribes a fire of given severity and extent, but is written primarily for full, non-loadbearing walls. This standard gives a prescribed minimum ambient (fire side) temperature as a function of time outside the first boundary of the fire barrier. In a slab geometry, a thermal performance criteria is then established at the opposite unexposed boundary of the barrier. The barrier is then rated as satisfying the standard for some stated period of time. Standard prescribes a fire of given severity and extent, but is written primarily for full, non-load bearing walls. The specification is necessary to ensure that the fire-resistive properties of materials and assemblies are measured according to some common standard, and that it can be applied in a variety of cases.

One of the conditions of test acceptance is that the fire barrier not allow passage of the flame or hot gases, sufficient to ignite a cotton sample. A second condition is that the transmission of heat be limited so as not to allow the unexposed barrier surface temperature to be raised more than 139°C (250°F) above its initial temperature. Also, the sample must maintain its structural integrity throughout the experiment. A performance rating is then given as "1-h," "1-2-h," or "2-5-h" according to how long, in hours, the sample continues to meet all of the conditions of acceptance.

As in many high-temperature and combustion applications, ceramic fiber insulation is often used, because it can withstand the high temperatures involved in such situations. For structural sup-
port and for radiative shielding, thin layers of foil are placed on the outside of the ceramic fiber blanket and sometimes between layers of insulation. Aluminum foil would work very well for this purpose in terms of reflectivity, but it cannot withstand the high temperatures. Therefore, layers of stainless steel foil are used as a radiative shield. Since the primary modes of heat transfer in high-performance insulations are radiation and convection, the foil layers also serve the purpose of blocking bulk transport (and convection) of hot gases passing through the porous insulation in the barrier.

Once a candidate fire barrier has been proposed, a detailed heat transfer analysis is performed to calculate the transient temperatures within the barrier. In the analysis, one must look at all modes of heat transfer, including conduction, convection, and radiation. Even though the sample can be modeled as a one-dimensional slab, obtaining an analytical solution for even simplified cases is very difficult. Convection, conduction, and radiation are all considered at the outside boundaries, while only conduction and radiation are used within the slab; the boundary exposed to the fire is termed the exposed boundary, while the boundary opposite the fire is termed unexposed. To solve this problem numerically, accurate finite difference representations of transient conduction and radiative heat transfer must be determined. Then, an iteration between the radiation and the conduction at each time-step is converged at each grid location. Once the temperature calculations at one time-step are complete, new solutions are found for the next time-step. These steps are repeated for a specified amount of time. A coarse grid numerical technique (optimized for the minimum grid resolution) is incorporated here to model the conduction and radiation in a one-dimensional slab, built up of successive layers of foil and insulation, in which the variable direction is perpendicular to the plane of the slab boundaries (and the fire).

The ASTM Standard E 119 test setup was used as an experimental validation of the numerical analysis presented here. Although the standard was followed as closely as possible, some of.
the test conditions could not exactly match the standard. The actual furnace temperatures, while measured with relatively small uncertainty, could not be controlled exactly according to the standard. Therefore, the furnace temperature was controlled to always be greater than the standard, anywhere from 1 to 168 K (2 to 297 R). This led to a conservative analysis of the fire barrier temperatures, relative to the standard. A second variance is that the standard is based on testing a full wall (cross-sectional area not less than 9 m²); the actual test area exposed to the fire, however, was only 0.36 m², with dimensions 0.6 × 0.6 m. The aspect ratio (sample width to thickness) for the tests was greater than 20 for all tests; this is well within the one-dimensional slab assumption based on prior research (Saboonchi, 1988).

Literature Review

Many investigators have studied combined conduction and radiation heat transfer, especially as it applies to thermal insulation. King (1978) developed a model for the heat transfer through fibrous insulating materials that could be used to estimate the apparent thermal conductivity as well as heat losses and temperature profiles. Barker (1984, 1985) developed a numerical code, based on exact transient and steady-state solutions of the combined conduction and radiation heat transfer for a gray medium, heated by separate external sources; he also presented (1984) an extensive review of the related literature up to that time. His analysis showed that as the conduction to radiation ratio decreased, the medium reached steady state faster. Experimental and analytical results for transient total heat transfer through fiberglass insulation, applied to residential attics, were compared by Rish and Roux (1987). The results showed that the addition of a radiant foil barrier reduced the total heat transfer by up to 42 percent. Su (1993) also calculated the transient heat transfer through an electromagnetic window with varying system parameters. Using constant values for specific heat, thermal conductivity, and density proved to have little effect on the results, compared to temperature-dependent properties.

Since fibrous insulation material properties are affected by the size, distribution, and orientation of the fibers, Tong and Tien (1983) and Tong et al. (1983) performed analytical and experimental studies, respectively, on insulation heat transfer. In the analytical study, the mean fiber radius was found to have a greater effect than the type of size distribution or the chemical composition of the fibers. In the experimental analysis, extinction coefficients, radiant heat fluxes, and total heat fluxes were measured. The results agreed with the analytical study only qualitatively, due to the nongray and nonuniform characteristics of the insulation. Saboonchi et al. (1988) developed a technique for determining the scattering coefficient and mass extinction coefficient of insulation, and Kumar and White (1995) developed a simple model to account for the interaction between scattered radiation for the individual fibers. It was found that as the fiber separation decreased and the absorption index increased, the scattering efficiency increased.

The effects of natural and forced convection on the heat transfer in insulation materials have also been studied. Bankvall (1978) found that as the aspect ratio (height/depth of insulation) was increased, indicating a longer sample, the heat transfer decreased. Silberstein et al. (1991) related various experimental studies to find that the thermal properties of the insulation were not significantly affected by forced convection, except in cases where poor workmanship allowed multiple air entry paths.

The focus of this study is to analyze the combined conduction and radiation heat transfer of a one-dimensional gray medium, modeling an expansion joint fire barrier. A parametric study of time/temperature profiles was made for various thicknesses, numbers of layers, and material properties. For validation, the numerical results were compared with experimental data.

Experimental Apparatus

Experimental measurements of fire barrier temperature were obtained as a basis for validating the numerical results. The experimental apparatus can be divided into three major components. These include the test specimen or the fire barrier, the furnace which served to simulate the standard fire, and the temperature measurement and data acquisition equipment. Figure 1 shows an illustration of the apparatus used in this experiment. The experimental setup was built and tested performed at the University of Oklahoma, Aerospace and Mechanical Engineering, North campus test cell facility.

The test specimen was composed of seven thin layers (0.32 cm ea., uncompressed) of insulation with eight layers of foil placed between the insulation gaps and on the outer boundaries. thermo-

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**Nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>A</td>
<td>cross-sectional area of slab</td>
</tr>
<tr>
<td>C_p</td>
<td>specific heat</td>
</tr>
<tr>
<td>F_o</td>
<td>Fourier number</td>
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<tr>
<td>h_1</td>
<td>convective heat transfer coefficient at fire boundary</td>
</tr>
<tr>
<td>h_2</td>
<td>convective heat transfer coefficient at unexposed boundary</td>
</tr>
<tr>
<td>I, i</td>
<td>radiant intensity and radiative intensity at node i, with n being the last node</td>
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<tr>
<td>k</td>
<td>thermal conductivity</td>
</tr>
<tr>
<td>L</td>
<td>slab thickness</td>
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<tr>
<td>q_0</td>
<td>conductive heat transfer</td>
</tr>
<tr>
<td>q_r, q_i</td>
<td>radiative heat transfer and radiative heat transfer at node i</td>
</tr>
<tr>
<td>q_f</td>
<td>total heat flux</td>
</tr>
<tr>
<td>S, S_i</td>
<td>radiative source function</td>
</tr>
<tr>
<td>T</td>
<td>temperature</td>
</tr>
<tr>
<td>T_ave</td>
<td>control volume-averaged nodal temperature</td>
</tr>
<tr>
<td>T_i, T_0</td>
<td>temperature and temperature fromprevious time-step at node i</td>
</tr>
<tr>
<td>T_w, T_a</td>
<td>fire temperature and unexposed side ambient temperature</td>
</tr>
<tr>
<td>V</td>
<td>control volume</td>
</tr>
<tr>
<td>x</td>
<td>slab position</td>
</tr>
<tr>
<td>a</td>
<td>thermal diffusivity</td>
</tr>
<tr>
<td>b, o</td>
<td>mass extinction coefficient and scattering albedo</td>
</tr>
<tr>
<td>Δt, Δx</td>
<td>time-step and control volume length increments</td>
</tr>
<tr>
<td>μ</td>
<td>cosine of the radiative propagation angle, θ</td>
</tr>
<tr>
<td>ρ</td>
<td>density (without subscript)</td>
</tr>
<tr>
<td>ρ_1, ρ_2</td>
<td>reflectivity at 1st, 2nd, and successive boundaries</td>
</tr>
<tr>
<td>σ</td>
<td>Stefan-Boltzmann constant</td>
</tr>
<tr>
<td>τ</td>
<td>optical thickness</td>
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</tbody>
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couples were located on the outsides of the insulation, just inside the foil boundaries of layers, and as close as possible to the center (in the area dimension). The insulation material was a ceramic fiber blanket called Durablanket® S, manufactured by Carborundum Company. This material was selected specifically because of its ability to withstand the extreme fire temperatures, low heat capacity, and resistance to thermal shock.

As previously noted, layers of mirror finish, stainless steel foil were used between layers and on the outer boundary to reflect thermal radiation from the fire and to add structural stability to the fire barrier. The thickness of each layer of foil was 0.0076 cm.

The furnace to simulate the fire in a building, as given in ASTM E 119, was constructed of standard sheet steel and angle iron, with outer dimensions of 82.0 × 82.0 × 82.0 cm. The four side walls and bottom of the furnace were heavily insulated, leaving inside dimensions of 60.0 × 60.0 × 70.0 cm. A 60.0 × 60.0 cm exposed opening was left at the top, on which the test specimen could be fixed. Steel bars and fire bricks fixed the horizontal sample in place. Care was taken not to compress the insulation within the exposed area opening. In order to avoid secondary heat loss effects, no structural support, other than the foil, supported the sample from the lower (fire) side. However, in avoiding compression of the sample, some minor “bagging” expansion of the insulation may have occurred at higher temperatures. To account for this observed structural behavior, sensitivity calculations on the position of thermocouples in the thickness dimension were made. The associated systematic uncertainties to thermocouple position are estimated and discussed later.

The furnace of the test setup was manually controlled. An inlet on one side was used to pipe natural gas into the system. A blower forced combustion air was forced into the furnace. Combustion products were allowed to escape through a manually position flue vent. The gas was turned on, then a sparking device was used to ignite the flame. The gas was adjusted, manually, to best match the theoretical temperatures. Openings were also left for thermowells, to measure the flame temperature. The thermowells were extended 32.0 and 42.0 cm into the furnace so the flame temperature measurements could be taken at two locations and averaged. The thermowells were made of 1.8-cm diameter standard black steel piping and end caps.

Finally, temperature measurement and data acquisition equipment were used to obtain the temperatures of the oven and the sample. Twenty-gage, type K thermocouples from Omega were located within the thermowells. Data acquisition was done using a standard Dell 386 SX-16 computer, with a Keithley Metabyte DAS-8 data acquisition card and a Keithley EXP-16 I/O expansion board. The thermocouples within the sample were 30-gage, type K, glass insulated, that were placed inside the sample and on the unexposed surface. In the following section, the uncertainties involved in the experiment and the experimental results will be discussed.

Experimental Results

Since the furnace system was controlled manually, a number of preliminary tests were made to give the proper static gas and air setting used; since this was set for the entire time, only the outlet vent damper was varied slowly in small increments to control the flame temperature (quicker variation would not be compatible with the response of the thermowells). Thus the temperature of the fire was controlled to stay above that of ASTM Standard E 119 (1983). Since the control in varying the flame temperature was purposely bound below by the standard, the actual oven temperature remained from 1 to 168 K above the standard temperature throughout the experiment as shown in the results.

As discussed earlier, vertical thermocouple position in the horizontal unsupported sample was an area of uncertainty. Since 20-gage thermocouples were used for portions of the test, an assumed uncertainty for all measurement locations (even those with 30 gage) was taken as ±0.089 cm, and sensitivity calculations performed within that range.

Finally, the experiment was performed only once for a given set of conditions with this level of instrumentation. The reported uncertainty therefore notes only the system bias (without random error) of the measurement system relative to fixed ice bath and boiling water standards.

The data acquisition system consisted of a DOS-based PC with a Keithley/Metabyte DAS-08 A/D converter and a Keithley/ Metabyte EXP-16 MUX board. The EXP-16 was wired to a SMP plug harness by 183 cm of special limits of error, transmission grade, type K thermocouple wire (OMEGA). Individual internal sample measurements were by commercial SMP male terminated, 30-gage, glass insulated, type K thermocouples of the same manufacture, connected to the harness. Figure 2 illustrates the experimental flame temperature compared to the ASTM standard. Error bars on the measured temperature are estimated by linear extrapolation, point by point, based on the water standard noted above. For example, the plotted error bars are for uncertainty estimated to be ±11.4 K and −6.1 K on a temperature of 1394.8 K after three hours of heating.

Another uncertainty, not accounted for in the model, was the thermal degradation by partial oxidation of the reflectivity of the outer stainless foil layer during the test. Also, the effective density used in the analysis was most likely too low, because it did not specifically take into account the foil density (although composite system was varied).

Again in Fig. 2, comparison of the experimental flame temperature and the ASTM standard flame curve as a function of time is given. For the first five minutes, the actual flame temperature matched the standard almost exactly, and stayed close for the first 20 minutes. The largest temperature difference occurred after about 40 minutes, with the measured flame temperature staying

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close to 1350 K for the rest of the time. The temperature difference was 40 K after three hours.

The experimental temperatures at the unexposed boundary are given in Fig. 3. These are the values measured on the outer foil layer of the seven layer fire barrier sample. From the figure there appears to be a 10 to 15-minute time period before the outer boundary temperatures began to rise significantly. For the next 45 minutes, the temperature increase was close to linear, from 290 to 360 K. The temperature increased to 650 K after two hours and 685 K after three. Over the last one-half hour, the temperature increase was somewhat smaller.

**Thermal Analysis**

In order to determine the heat transmission and temperatures though the sample, a thermal analysis was performed. The ceramic fiber sample was considered a one-dimensional slab medium of thickness $L$. The one-dimensional assumption was taken since the cross-sectional dimensions were much larger than the thickness (24:1). The slab was assumed to be an isotropic, homogenous, absorbing, emitting, and scattering medium. Thermal, physical, and radiative properties of the materials involved were taken to be constant. Su (1993) showed that there was very little effect in the results of combined radiation and conduction analysis when the properties were allowed to change with temperature. The convective heat transfer coefficient was varied with temperature difference between the surface and the ambient air outside (model taken as a hot plate facing up, or cold plate facing down with $hL/k = e(Gr_f Pr)^{0.3}$) through the Grashof number.

When using the energy equation and radiative transfer equation it is important that the boundary conditions be properly described. In this case, one boundary was heated by a flame, while the other boundary was exposed to a constant ambient temperature. The boundary nearest the flame will be referred to as the fire side boundary, and the boundary on the opposite side of the flame will be referred to as the unexposed boundary. For the fire side, natural convection, radiation, and conduction from the oven were considered. On the unexposed boundary, radiation, conduction, and convection were again considered. Within the barrier, only conduction and radiation were analyzed. Convection was neglected inside the porous slab since the fiber insulation (and foil layers when used) prevented air inside the barrier from moving freely. The foil layers, located on both the fire side and unexposed boundaries, as well as between layers, were only considered for their reflective properties. Using a lumped system analysis, according to (Ozisik, 1985, p. 103), it was found that the entire layer of foil would reach 96.4 percent of the fire temperature within 0.01 hours (due to the small thickness). Therefore for the analysis, the foil was assumed to have the temperature of the adjacent insulation.

The energy equation was applied to this problem by dividing the slab into coarse grid multiple control volumes, using a finite difference approximation. For the cases in which only one layer of insulation was used, the thickness was divided into five control volumes and six equally spaced nodal locations. Five volumes was determined to give accuracy within thermocouple tolerances by successively reducing the number of control volume which would reproduce benchmark results with the model (approximately 20) when compared to published exact solutions (Barker). When multiple layers were used, each layer was again divided into five control volumes. Visual illustrations of the thermal energy balance on the fire side boundary, the interior nodes, and the unexposed boundary, from Caplinger (1997), are given in the Appendix, using simplified heat transfer coefficients for free convection on a horizontal plate (hot plate facing up or cold surface facing down. (Ozisik, 1985, p. 437).

The transient conduction energy equation includes conduction and radiation; the development of the analytical form of the equation may be found in Ozisik (1973, Chapter 12). The finite difference model derivation from the analytical statement of the problem used here may be found in Su (1993). Here, we simply state the physical basis of the control volume model. For the flame-side first finite difference volume, a half volume of the slab includes the flame side boundary. The sum of the heat transfer coming in from the left and right of the control volume equal the energy storage inside. The rate of heat being added from the left combines the convection and radiation from the fire. The rate of heat being added from the right side includes both internal conduction and radiation (no convection). The rate of energy storage is equal to the sum of these terms. This energy rate balance can be described mathematically as

$$h_i(A(T_{sw} - T_i) + (1 - \rho_i)\sigma A(T_{sw}^4 - T_i^4) + k A \frac{T_2 - T_1}{\Delta x} - \frac{(q + q_f)}{2} A = \rho C_v V \frac{(T_{sw} - T_{avg})}{\Delta t}$$

(1)

where $V = A\Delta x/2$, and $T$ and $To$ refer to the temperatures at the present and previous time-steps.

To evaluate the energy storage term in this coarse grid, space-averaged temperatures were needed for the present and previous time-steps. This is essential for a coarse grid model. Therefore, the storage term was given as follows:

$$\rho C_v V \frac{(T_{sw} - T_{avg})}{\Delta t} = \rho C_v A \frac{\Delta x}{2} \frac{3T_1 + T_2}{4} \frac{3T_{avg} + T_2}{4} \frac{\Delta t}{4}$$

(2)

The radiation term was given by the difference of the fourth powers of the fire temperature and boundary temperature. To use a tridiagonal matrix solver for the temperatures in the energy equation, it was necessary to linearize the radiation term as follows:

$$(T_{sw}^4 - T_i^4) = (T_{sw})^4 + (T_{sw})^3(T_{sw} - T_1)(T_{sw})^2(T_{sw} - T_2)$$

(3)

Because of the number of time-steps taken (and resulting small temperature changes) and the successive iteration between radiation and temperature solutions, this linearization has negligible
effect on the results. The tridiagonal solver, however, greatly accelerates the computation speed.

The interior nodal equations are identical, except for the nodal references. The heat being added from the right and left both included conduction and radiation. The energy storage term was similar to that used in the first boundary, and the energy balance is best described by the following equation:

\[
kA \frac{(T_{i-1} - T_i)}{\Delta x} + A \frac{(q_{i-1} + q_i)}{2} + kA \frac{(T_{i+1} - T_i)}{\Delta x} - A \frac{(q_{i+1} + q_i)}{2} = \rho C_p \Delta x \frac{(T_i - T_0)}{\Delta t},
\]

(4)

After the interior and fire side boundary equations have been formulated, the unexposed boundary equation was determined. It was much the same as the fire-side boundary equation, except for the nodal designations and an ambient temperature that remains constant. An equation for the energy balance on the unexposed boundary is given as follows:

\[
h_2A(T_s - T_{a2}) - (1 - \rho_s)\sigma A(T_{a2}^4 - T_s^4) + kA \frac{(T_{n-1} - T_n)}{\Delta x} + \frac{(q_{n-1} + q_n)}{2} = \rho C_p \Delta x \frac{(T_{avg} - T_0_{avg})}{\Delta t},
\]

(5)

where \( n \) indicates the last node and \( T_{avg} \) was defined in a similar manner to the first boundary.

To calculate the radiative heat transfer values in the energy equation it was necessary to numerically determine the radiative heat transfer at each nodal location. For the radiation analysis a combination of finite difference and a variation of discrete ordinates was used. Radiative intensity was determined from the formulation of the radiative transfer equation, found in Ozisik (1973, Ch. 8), which is given as

\[
\mu \frac{\partial I(\tau, \mu)}{\partial \tau} + I(\tau, \mu) = (1 - \omega)I_s[T] + \frac{\omega}{2} \int_{-1}^{1} I(\tau, \mu')d\mu',
\]

(6)

where \( I_s[T] = n^2 \sigma T^4/\pi \) (assuming refractive index \( n = 1 \)), \( \omega = \sigma/\beta, \Delta \tau = \rho B dx, \mu = \cos \theta, \) and \( \theta \) is defined as the angle of radiation propagation with respect to the surface normal of the material. The right-hand side of the equation is called the source function \( S(\tau) \). Therefore, the radiative transfer equation can be simplified to

\[
\mu \frac{\partial I(\tau, \mu)}{\partial \tau} + I(\tau, \mu) = S(\tau)
\]

(7)

where \( \mu > 0 \) for radiative intensity from the positive direction relative to planes perpendicular to \( \tau (\mu = 0) \) and \( \mu < 0 \) for intensity from the negative direction.

The boundary intensities in the insulation are given physically by the effective emitted intensity plus reflected incident intensities from the stainless steel foil into that direction

\[
I^+(0, \mu) = (1 - \rho_i) \frac{\sigma T_i^4}{\pi} + \rho_i I^-(0, -\mu); \quad \mu > 0 \quad (8a)
\]

\[
I^-(\tau_0, -\mu) = (1 - \rho_2) \frac{\sigma T_0^4}{\pi} + \rho_2 I^+(\tau_0, \mu); \quad \mu > 0 \quad (8b)
\]

where \( \tau_0 \) is the optical thickness of the slab, and \( -\mu \) has been used to make all directions \( \mu > 0 \).

The radiative transfer equation for the insulation was then put in discretized form, and a form of discrete ordinates, found in Sutton and Kamath (1986), was used to calculate boundary intensities

\[
\frac{I^+_i - I^+_{i-1}}{\Delta \tau} + \frac{I^-_{i-1} + I^-_i}{2} = \frac{S_i + S_{i-1}}{2}; \quad \mu > 0 \quad (9a)
\]

\[
-\mu \frac{I^+_i - I^+_{i+1}}{\Delta \tau} + \frac{I^-_{i+1} + I^-_i}{2} = \frac{S_i + S_{i+1}}{2}; \quad \mu > 0 \quad (9b)
\]

An initial guess was made for the source term that considers only the emission term

\[
S_i = (1 - \omega) \frac{\sigma T_i^4}{\pi}.
\]

(10)

Then, after the intensity for the first iteration was calculated, the in-scattering integral could be evaluated. Both terms for the source function were then used for the following iterations:

\[
S_i = (1 - \omega) \frac{\sigma T_i^4}{\pi} + \omega \int_{-1}^{1} I d\mu.
\]

(11)

Once the intensities had been calculated, the radiative heat transfer could be determined. The radiative heat transfer perpendicular to the plane of the slab was defined as follows:

\[
q_r = 2\pi \int_{0}^{\pi} \int_{-\theta}^{\theta} I \cos \theta \sin \theta d\theta d\phi
\]

(12)

where \( \mu = \cos \theta \) and \( -d\mu = \sin \theta d\theta \). This may be resolved in terms of positive and negative intensities as in Ozisik (1973)

\[
q_r = 2\pi \left[ \int_{0}^{1} \mu I^+_1 d\mu - \int_{-1}^{0} \mu I^-_{1'} d\mu \right].
\]

(13)

**Sensitivity Analysis and Results**

The numerical computations were made for many different cases to determine how changing barrier characteristics and material properties affected the temperature distribution. Comparisons were made to published pure conduction, pure radiation, and combined results in Barker (1984) with agreement within the convergence tolerance. The two most important cases were for varying the thickness of insulation, to try to find an optimum thickness and changing the number of layers of foil between insulation layers. Other material properties that were varied include density, thermal conductivity, and specific heat.

The convergence criteria used in the computer code was \( \Delta T < 0.01 \) (between iterations for conduction and radiation) and \( \Delta S < 1.0 \times 10^{-5} \) (between iterations for radiation). The following is a list of default properties used in the numerical code, unless otherwise specified: \( \Delta \tau = 0.01 \) h, \( \Delta x = \text{thickness}/\text{number of control volumes}, \beta = 38 \text{ m}^2/\text{kg} \cdot \text{°C}, \gamma = 4864 \text{ thickness}, \rho_i = 0.6, \rho_2 = 128 \text{ kg/m}^3, \kappa = 6.20 \times 10^{-2} \text{ W/(m °C)}, \) and \( C_p = 1170 \text{ J/(kg °C)} \). Variations of this code have been used in a number of cases, such as Su (1993) and Barker (1984), in which the results agreed with published data.

The most critical factor that determines the effectiveness of a fire barrier is the thickness of insulation. In Fig. 4, a single-layer sample with foil only on the outer boundaries was examined. All of the material properties were held constant while the material thickness was varied from 0.318 to 5.08 cm by multiples of two.

As expected, when the thickness was increased, the temperatures for the entire three-hour time period decreased. For the thinner material cases, the curves were not as steep over the first 1.5 hours, but then become close to parallel. One important fact from these curves is that as the insulation thickness was doubled, the benefits of decreased temperature became additive rather than doubling.
The second factor examined was the number of foil layers between insulation layers. It is important to know how many layers of foil are necessary to provide the greatest benefit in blocking the thermal radiation. For this case, each sample was taken to be 2.54-cm thick with foil located on the outer boundaries in every case. The material properties were held constant and the sample was divided in halves, twice; giving one, two, and four-layer samples, with foil between each. Figure 5 shows the temperature distribution, in these cases, for the unexposed boundary. The difference between the two and four-layer cases was nearly indistinguishable, while the one-layer cases varied less than 5 K. The profiles show that the multiple foil layer cases did not perform much better than the single-layer case, indicating the need for foil only on the boundaries. Additional foil layers are only necessary if the outer layer were to break down over time or to physically preserve the model conditions (no convective flow through and dimensional stability).

The next series of figures shows the effects of changing material properties while maintaining a constant thickness and number of layers. Three charts are given for variations in effective density, thermal conductivity, and specific heat, respectively, for a 2.54-cm thick slab with only one layer of insulation. These results track trends of the pure conduction case, except for density; this is due to the fact that density also changes the optical thickness for the radiation in a nonlinear fashion. Figure 6 shows temperature profiles for a three-hour time period with densities of 64, 128, and 256 kg/m³. The most significant trend evident in these curves is that as the density increased, the slope of the curve decreased. After 3.0 hours, the temperature of the 256 kg/m³ case was 35 K less than the 128 kg/m³ case and 85 K less than the 56 kg/m³ case. As density was increased, the heat transmission was less and it reached steady state earlier. The transient temperatures for thermal conductivities of 4.04 × 10⁻³, 6.20 × 10⁻³, and 8.36 × 10⁻³ W/(m K) are given in Fig. 7. From these curves it was determined that as thermal conductivity increased, the temperatures for the entire time period increased. As expected, higher thermal conductivity allowed more heat to pass through the barrier. In Fig. 8, temperature curves are given for three different specific heat values of 1.17, 2.01, and 2.85 kJ/(kg K). Specific heat appears to play a role only for the initial times. As it was increased, the heat transmission over the first hour was slower.

It was necessary to compare the numerical results with those obtained experimentally to determine a correlation. Since the experimental case contained seven layers of insulation separated by foil, this was used in the numerical computation. The material properties were optimized over a limited range to find the best fit to the experimental time-temperature curve. The optimization was done systematically, by holding two of the thermal properties constant and finding the best match for the other property. This was done until the best match was found for all three properties. The property values that brought about the best match to the experimental results were: density = 256 kg/m³, thermal conductivity = 7.93 × 10⁻³ W/(m K), and specific heat = 2.93 × 10³ J/(kg K). The temperature position was offset by 0.089 cm inside the boundary to account for possible thermocouple position uncertainty that might be due to vertical “bagging” of the unsupported horizontal sample or due to the finite size of the thermocouple wire junction (single AWG 30 wire is 0.0254 cm in diameter).
The plot of the experimental and computed temperature profiles at the unexposed boundary is given in Fig. 9. The experimental flame temperature curve (Fig. 2) was used as input data to the numerical model. The figure shows that for the first 1.5 hours the numerical solution matched the experimental data very closely, overestimating the temperatures from 2–50 K. The curves then cross, and for the next 1.5 hours the temperatures were underestimated. After 3.0 hours, the temperature difference was 27 K. The numerical solution came to a steadier upward trend faster than the measured temperatures. Note that as in many heavily insulated systems, with active heat rate addition, no steady state is achieved; the process ultimately would end in destruction of the sample.

The better experimental fit by increased density can be explained by the fact that the foil was not explicitly taken into account in the conduction energy equation. Including the foil would increase the effective density and thermal conductivity of the composite system. The conductivity was increased slightly to match the test data. Since there was such a large range of temperatures estimating a constant specific heat was difficult. The specific heat had to be increased significantly, which could also be due to a low mean temperature value, initial moisture in the sample, or transient response of the furnace walls.

**Conclusions**

In this study, transient thermal analysis was studied relative to an experiment for foil and insulation fire. The transient temperature profile of ASTM E 119-1983 was used as a basis for the experiment and validation of the numerical code. The experimental fire barrier was composed of seven layers of ceramic fiber insulation material, with layers of stainless steel foil placed between each layer and at the outer boundaries. Interior sample and unexposed boundary temperature measurements were made in five-minute intervals (to match the standard) for a period of three hours at two locations within the furnace and on the outer boundary of the test sample. The furnace temperatures were found to be consistently higher than the standard curve, which was necessary for a conservative analysis. A numerical model was developed based on the transient conduction energy equation for the fire barrier. The one-dimensional slab was divided into six nodes per layer of insulation (including the boundaries), and temperatures were calculated for varying conditions over a three hour period, according to the standard ASTM fire. From these calculations, it was found that as the insulation thickness was doubled, there was an additive (but diminishing) benefit of decreased boundary temperatures. When the number of foil layers was varied, there was very little difference in the calculated temperature profiles. In order to get the best match of numerical to experimental values, the density, thermal conductivity, and specific heat were all increased.

With the small changes noted above (foil not being considered in the effective density, the use of constant properties, and other factors) the numerical model agreed well in predicting the experimental trend in temperature through a three-hour test of a real fire barrier.

**Acknowledgment**

This work was sponsored in part by U.S. Department of Energy under contract DE-FG36-95GO101112.
References


