3D THERMAL MAPPING OF CONE CALORIMETER SPECIMEN
AND DEVELOPMENT OF A HEAT FLUX MAPPING PROCEDURE
UTILIZING AN INFRARED CAMERA

by

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ABSTRACT

The Cone Calorimeter has been used widely for various purposes as a bench-scale apparatus. Originally the retainer frame (edge frame) was designed to reduce unrepresentative edge burning of specimens. In general, the frame has been used in most Cone tests without enough understanding of its effect. It is very important to have one-dimensional (1D) conditions in order to estimate thermal properties of materials. It has been implicitly assumed that the heat conduction in the Cone Calorimeter is 1D using the current specimen preparation. However, the assumption has not been corroborated explicitly to date. The first objective of this study was to evaluate the heat transfer behavior of a Cone specimen by examining its three-dimensional (3D) heat conduction.

It is essential to understand the role of wall lining materials when they are exposed to a fire from an ignition source. Full-scale test methods permit an assessment of the performance of a wall lining material. Fire growth models have been developed due to the costly expense associated with full-scale testing. The models require heat flux maps from the ignition burner flame as input data. Work to date was impeded by a lack of detailed spatial characterization of the heat flux maps due to the use of limited instrumentation. To increase the power of fire modeling, accurate and detailed heat flux maps from the ignition burner are essential. High level spatial resolution for surface temperature can be provided from an infrared camera. The second objective of this study was to develop a heat flux mapping procedure for a room test burner flame to a wall configuration with surface temperature information taken from an infrared camera. A
prototype experiment is performed using the ISO 9705 test burner to demonstrate the developed heat flux mapping procedure. The results of the experiment allow the heat flux and spatial resolutions of the method to be determined and compared to the methods currently available.
EXECUTIVE SUMMARY

The Cone Calorimeter has been used widely for various purposes as a bench-scale apparatus. Originally the retainer frame (edge frame) was designed to reduce unrepresentative edge burning of specimens. In general, the frame has been used in most Cone tests without enough understanding of its effect. It has been implicitly assumed that the heat conduction in the Cone Calorimeter is one-dimensional (1D) using the current specimen preparation. However, the assumption has not been corroborated explicitly to date. The first objective of this study was to evaluate the heat transfer behavior of a Cone specimen by examining its three-dimensional (3D) heat conduction.

A series of Cone Calorimeter tests with ceramic fiberboard were conducted in four different configurations by modifying the edge condition and/or the specimen size based upon the current specimen preparation method. A total of forty temperature measurements were taken at the center, middle, and corner of the specimen as well as at various depths. 3D thermal maps were created using the results from the Cone tests to see if it is reasonable to assume 1D heat conduction. Comparison among the configurations allows the following conclusions to be made: 1) Heat conduction is not 1D, but 3D, when using the specimen size, 100 mm by 100 mm (4 inch by 4 inch), regardless as to whether the retainer frame is used or not. This is due to the effect of the edge condition, i.e., heat loss or gain through the edge. 2) Wrapping the side of the specimen with an insulation material (ceramic blanket) shows less edge effect than without insulating the side. Even though the magnitude of the edge effect decreases
considerably, the center temperature data are still affected by the 3D heat transfer behavior. 3) The specimen size, 178 mm by 178 mm (7 inch by 7 inch), provides the best result in terms of approximating 1D heat conduction in the Cone Calorimeter.

According to the comparison of temperature at the center of the specimen for the configurations, the temperature from using a current standard specimen preparation was 20 – 25 K lower than the one from 178 mm by 178 mm specimen when temperatures were measured at 6 mm below from the surface for the incident heat flux, 80 kW/m², not only at 600 seconds but also in an early times such as 100 and 200 seconds. As heat flux is lower, the magnitude of the difference becomes smaller. In case of low heat fluxes such as 20 and 40 kW/m², the temperature difference at the depth is 10 – 15 K. The temperature difference at the surface is predicted to become larger, approximately 30 – 35 K. Therefore, more attention is called when surface temperature from the current Cone test setup is utilized with a 1D application. The difference, 30 – 35 K, can cause the results underestimated.

Ideally, data measured within the central 50 by 50 – mm area of the specimen should be the same as the one at the center if the 1D heat conduction assumption is correct as has been assumed implicitly. However, it turned out that it did not occur with a current standard specimen preparation. In order to minimize an error special attention is needed with a current Cone setup when data are measured at depth not only at the center of the specimen but also near the 50 by 50 – mm area. The data measured at the center might be considered 1D if high accuracy of the 1D data is not required. However, the data taken in the 50 by 50 – mm area not only at the surface but also at depth will be
affected by the edge condition. The temperature data will be up to 40 K higher than temperature at the center for 80 kW/m². For low heat fluxes such as 40 kW/m², the temperature difference between center and 50 by 50 mm area will be up to 15 K. Temperature measurements kept apart from the center would not be as good as the one at the center due to the edge effect even if temperature measurements were taken in 50 by 50 mm area.

From a practical point of view the simulated 1D and 3D data were applied to ignition analysis. As an example two sets of data from using a current standard specimen preparation with the frame that has a 3D effect and the proposed 1D specimen preparation were used following Janssens’ procedure. Overall the analysis did not show a significant difference between the two sets of data. It is fair to say that the ignition analysis did not differentiate 3D data from 1D data. The reason is thought that the simulated 3D data were unrefined due to the fact that the corrected thermal conductivity was an effective 1D property used to approximately compensate for 3D behavior (bias) when using a 1D model.

It is very important to understand the role of wall lining materials when they are exposed to a fire from an ignition source. Full – scale test methods permit an assessment of the performance of a wall lining material. However, fire growth predictions using models have been developed due to the costly expense associated with full – scale testing. The models require heat flux maps provided by the ignition burner flame. Works to date were impeded by a lack of detailed characterization of the heat flux maps.
due to the use of limited instrumentation. To increase the power of fire modeling, accurate and detailed heat flux maps from the ignition burner are essential. High level spatial resolution for surface temperature can be provided from an infrared camera. The second objective of this research was to develop a heat flux mapping procedure for a room test burner flame to a wall configuration using surface temperature measurements taken from an infrared camera. This second part of the research utilizes the 1D temperature data from the Cone Calorimeter tests for the heat flux mapping procedure.

As wall lining material, ceramic fiberboard was heated by a room test ignition burner for a certain duration. One temperature measurement in depth was taken at a location using a thermocouple. The location is called “Location A” for convenience. Immediately after the fire source was removed, cooling temperature data over the surface area including Location A were collected utilizing an infrared camera.

As a part of heat flux mapping one – dimensional heat conduction model was developed. The model can provide temperature profiles not only at the surface but also in depth under a given incident heat flux with appropriate thermal properties of the wall lining material. The 1D temperature data from the model are compared with the 1D temperature data measured from the Cone Calorimeter tests. The comparison allows the resolution of the model to be determined which is ±1 kW/m².

The temperature measurement in depth at Location A was compared with simulated temperatures from the model. Each run of the model provided a different predicted temperature profile for the same exposure duration as in the experiment. The only difference of each run was the incident heat flux. When the measured temperature
data at Location A corresponded to the simulated one, the incident heat flux used in the model has been determined as a best estimate heat flux from the ignition burner at Location A. Then, cooling surface temperature data from the infrared camera at Location A were compared with the modeled temperature under the determined incident heat flux. A residual between the surface temperature data from the infrared camera and the simulated temperature data was calculated. It is the procedure to determine the residual for a best estimate heat flux.

Using the residual at Location A as matching criteria for heat flux, other heat fluxes for other locations were determined by comparing cooling surface temperature histories from the infrared camera with those from the model. Through repeating this heat flux mapping procedure a heat flux map was created by combining the heat flux at all individual locations.

Development of the 1D heat conduction model was a must to simulate temperature profiles under a given incident heat flux. The advantage of using the 1D “direct” approach is that the mathematical complexities and difficulties associated with 3D and 1D “inverse” codes are avoided. To validate the use of a 1D model to determine best estimate heat fluxes from a real fire situation, a commercial 3D conduction code, ALGOR’s finite element analysis, was used for the purpose of comparison. It is acceptable to use the developed 1D heat conduction model in the area of the center line above the burner which showed 1D behavior through ALGOR runs.

A prototype experiment in a wall configuration was performed using the ISO 9705 test burner to demonstrate the developed heat flux mapping procedure. Utilizing
the infrared camera, numerous cooling surface temperature data were collected along the center line above the burner with the spatial resolution of every 1 cm. Through the heat flux mapping procedure, the heat flux maps along the center line was obtained at every 1 cm. Compared to the currently available methods for heat flux maps, i.e., with lines of constant heat flux in 10 kW/m² increments, the developed heat flux mapping method can provide more detailed heat flux maps.
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NOMENCLATURE

$c$  heat capacity at constant pressure \( [J/kgK] \)

$D$  diameter of thermocouple \( [m] \)

$h_c$  convective heat transfer coefficient \( [kW/m^2K] \)

$k$  thermal conductivity \( [W/mK] \)

$\dot{Q}$  heat release rate \( [kW] \)

$q_{cr}$  critical heat flux for ignition \( [kW/m^2] \)

$q_e$  external heat flux \( [kW/m^2] \)

$q_{net}$  net heat flux \( [kW/m^2] \)

$T$  temperature \( [K] \)

$T_0$  initial temperature \( [K] \)

$T_{H}$  Cone heater temperature \( [K] \)

$T_{ig}$  ignition temperature \( [K] \)

$T_s$  surface temperature \( [K] \)

$t$  time \( [s] \)

$t_{ig}$  ignition time \( [s] \)

$x$  distance parallel to surface \( [m] \)

$y$  distance perpendicular to surface \( [m] \)

$\varepsilon_s$  emissivity of surface \( [-] \)
\( \rho \) density \( [kg/m^2] \)

\( \sigma \) Stefan-Boltzmann constant \( [5.67 \times 10^{-11} \, kW/m^2 K^4] \)
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1.0 DISSENGATION OVERVIEW

This dissertation consists of two major sections. The first section (Chapter 2.0) investigates the heat transfer behavior of a specimen in the Cone Calorimeter by examining the current implicit assumption of one-dimensional (1D) heat conduction. The second (Chapter 3.0) proposes a procedure for heat flux mapping from a room test burner utilizing an infrared camera.

Even though Chapter 2.0 and 3.0 are separate, Chapter 3.0 is directly related to Chapter 2.0. The 1D heat conduction model developed in Chapter 3.0 is calibrated utilizing the 1D temperature data at depth in the Cone Calorimeter in Chapter 2.0 as a part of the procedure for heat flux mapping. It is meaningless if the result from the 1D model is compared to the one from the Cone Calorimeter which shows 2D or 3D behavior.

Both Chapter 2.0 and 3.0 conclude with Section 2.7 and 3.8 which discuss the work that will be needed in the future for further studies.

Detailed works which were needed to support and complete the dissertation can be found in the Appendices A, B, G, and H. More data analyses from the Cone Calorimeter tests over time are provided in the Appendices C, D, E, and F.
2.0 INVESTIGATION OF HEAT TRANSFER BEHAVIOR OF A CONE SPECIMEN BY EXAMINING ITS THREE–DIMENSIONAL HEAT CONDUCTION

2.1 Introduction

2.1.1 Problem Statement

The Cone Calorimeter is the most commonly used bench–scale apparatus since its development in the 1980s\(^1\). Currently more data are available compared to the past, but the amount of data is not necessarily proportional to the accuracy of the data. Data with known confidence and limitations will result in better prediction of the effects of fire eventually. The accuracy of bench–scale data or uncertainty of the data is not well known or not well understood. As an example, bench–scale data for a common material with interest can be easily obtained from *The SFPE Handbook of Fire Protection Engineering*. For the most part people who use the data from the Handbook assume automatically that the bench–scale data are one–dimensional (1D). The data are then used as input for fire modeling to predict full–scale fire performance. If the data are not 1D as it was assumed, the uncertainty band of the results from the modeling will end up becoming larger by using the biased data. Therefore it is essential to have a correct understanding of the data which are provided by currently used apparatuses such as the Cone Calorimeter.
The Cone Calorimeter is today’s preferred instrument for measuring bench-scale heat release rate (HRR). Besides the primary purpose of measuring the HRR of specimens, effective heat of combustion, mass loss rate, smoke, soot, and toxic gases are measured in the Cone Calorimeter. Another ability of the Cone Calorimeter is to obtain the ignitability of specimen materials under various incident heat flux exposures.

The size of the specimen is 100 mm by 100 mm with a thickness of up to 50 mm, which is three-dimensional (3D) physically. The retainer frame (edge frame) and the specimen holder are made of stainless steel. According to ASTM-1354, the retainer frame is optional and is used to reduce unrepresentative edge burning of specimens. In general, the frame is used in most Cone tests without enough understanding of its effect. More details will be discussed in Section 2.2.

Regarding the surface uniformity of heat flux from the Cone heater, it is noted that the irradiance within the central 50 by 50-mm area of the specimen is within ±2%. At the corners the heat flux drops off to approximately 30–40% of the central value, see Appendix A.

It is very important to have one-dimensional (1D) conditions in order to estimate thermal properties of materials because all the theories we mostly use are 1D. It has been implicitly assumed that the heat conduction in the Cone Calorimeter is 1D. However, the assumption has not been demonstrated in the literature to date. It needs to be understood when the Cone specimen behavior is 1D and when it is not 1D.
2.1.2 Objective

The objective is to evaluate the heat transfer behavior of a specimen by examining its 3D heat conduction in the Cone Calorimeter. The effect of the edge condition of a specimen including using the retainer frame on 1D heat conduction is investigated.

2.1.3 Approach

Ceramic fiberboard is chosen as a Cone specimen material because it is thermally stable and can be repeatedly exposed to the high heat fluxes. Cone Calorimeter tests with four different configurations are conducted by modifying the edge conditions and the size of the specimen. The temperature profiles at the center, middle, and corner of the specimen as well as at various depths are compared among the configurations to investigate the heat transfer behavior. The 3D thermal maps are created and analyzed to see if it is reasonable to assume a 1D heat conduction. The practical importance of 3D effects is demonstrated via material ignition analysis based on inert material theory.

2.2 Literature Review

Tsantaridis and Östman\textsuperscript{4} compared the test results with and without the retainer frame. They concluded that when using the retainer frame the time to ignition is somewhat longer and the rate of heat release and the specific extinction area are smaller. Fritz and Hunsberger\textsuperscript{5} investigated the Cone Calorimeter test procedures for mattress composites with various specimen preparations and equipment configurations. It was found that the retainer frame reduced the peak heat release rate, which is the same result...
as the work by Tsantaridis and Östman. However, Fritz and Hunsberger recommend use of the retainer frame considering that the frame was an improvement over no frame in that it held the loose specimen components in place during the test. Grayson et al.\textsuperscript{6} developed the Cone Calorimeter test procedure for cables and the materials from which they are made. Their test results showed that using extended cable specimens and sealing and shielding the cable ends did not affect times to ignition, but considerably reduced the heat release generated in early stages of the burn. Kashiwagi and Cleary\textsuperscript{7} investigated the effects of sample mountings on flammability properties of intumescent polymers. Four different mountings were used: 1) specimen and substrate size, 10.5 cm, with grid and the retainer frame, 2) specimen and substrate size, 10.5 cm, without grid but the frame, 3) specimen size, 10.5 cm, substrate size, 15.3 cm, without grid and frame, 4) specimen size, 10.5 cm, substrate size, 15.3 cm, without grid and frame, with 2 cm width Marinite edge around sample. The purpose of a non–metallic frame with Marinite was to contain polymer melt flow. One of their conclusions is that the sample mounting configuration significantly affects the heat release rate curve, particularly peak heat release rate. Unfortunately the comparison of the results from using case 3) and case 4) was not focused on. They concluded that it was not clear what sample mounting condition provides a true measure of the properties which can be applied to predict its behavior in an actual fire.

Work by Toal et al.\textsuperscript{8} showed that sample preparation has a very important bearing upon the test results. They found out that the addition of a metal collar reduces the
maximum rate of heat release (RHR) and wrapping the specimen in foil produces a secondary maximum in the RHR curve.

In addition, the effect of using the grid for intumescent materials was investigated by Mikkola\textsuperscript{9}. Through the Cone tests with a raised grid, straight grid, and without grid, it was found that grids have an effect on time to ignition and the design of a grid is critical with low density products.

The fact that using the retainer frame has an effect on the test results has been reported with a limited number of materials. However, a 3D thermal map of a Cone specimen has not been undertaken to date.

2.3 Repeatability of Ceramic Fiberboard

Cone tests were conducted to investigate the repeatability of the ceramic fiberboard when it is exposed to heat. A total of twelve Cone tests alternating between incident heat fluxes of 10 and 80 kW/m\textsuperscript{2} were conducted. The first test was carried out with new ceramic fiberboards. The result from the 1\textsuperscript{st} test is different from those from the 3\textsuperscript{rd}, 5\textsuperscript{th}, 7\textsuperscript{th}, 9\textsuperscript{th}, and 11\textsuperscript{th} tests for 10 kW/m\textsuperscript{2}. The tendency is more apparent in the case of 80 kW/m\textsuperscript{2}. There is a considerable difference between the result from the 2\textsuperscript{nd} test and those from 4\textsuperscript{th}, 6\textsuperscript{th}, 8\textsuperscript{th}, 10\textsuperscript{th}, and 12\textsuperscript{th} test for 80 kW/m\textsuperscript{2}. Figure 1 shows the temperature profiles at the center of the specimen for 80 kW/m\textsuperscript{2}. The temperature was measured at depth of 6.4 mm from the top surface. For the purpose of stabilizing ceramic fiberboard used in this experimental program in the Cone so that it would be repeatable,
the first Cone test with new ceramic fiberboards is conducted for 95 kW/m$^2$ and the first data are excluded for the data analysis.

![Graph showing temperature profiles](image)

**Figure 1** Repeatability of ceramic fiberboard for 80 kW/m$^2$ – Center temperature profiles.

### 2.4 Thermal Maps of a Cone Specimen by Examining its Three-Dimensional Heat Conduction

#### 2.4.1 Specimen Material System Preparation/Test Setup

A series of Cone Calorimeter tests were conducted with four specimen configurations to investigate their thermal behavior. Each configuration is different from the others in terms of the specimen size, and the use of the retainer frame, the specimen holder, and side insulation. The differences among the configurations are described in
Table 1 and Figure 2. Config [I] is the specimen size, 100 mm by 100 mm (4 inch by 4 inch), using the retainer frame and the specimen holder, which is a typical Cone test setup used in most cases. The setup after thermocouples were installed is seen in Figure 3. Config [II] in Figure 4 is the same as Config [I] excluding the frame since the use of the frame is optional. In the case of Config [III] the specimen holder is used without the frame like Config [II]. In addition, the side of the specimen is wrapped with an insulation material (ceramic blanket). The size including the side insulation is approximately 180 mm by 180 mm. The overall view is shown in Figure 5. The main difference in Config [IV] is the specimen size, 178 mm by 178 mm (7 inch by 7 inch), instead of 100 mm by 100 mm (4 inch by 4 inch) as seen in Figure 6. The maximum specimen size is limited to 178 mm by 178 mm due to the presence of the two support rods for the shutter on the Cone Calorimeter. The frame, the holder, and side insulation are not used in Config [IV]. Configs [II] and [III] can be considered intermediate configurations between Config [I] and Config [IV].
Table 1  Comparison of four configurations for the thermal behavior of the Cone specimen.

<table>
<thead>
<tr>
<th>Specimen size</th>
<th>Retainer frame</th>
<th>Specimen holder</th>
<th>Side insulation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Config [I] 100 mm × 100 mm (4 in. × 4 in.)</td>
<td>Yes</td>
<td>Yes</td>
<td>No</td>
</tr>
<tr>
<td>Config [II] 100 mm × 100 mm (4 in. × 4 in.)</td>
<td>No</td>
<td>Yes</td>
<td>No</td>
</tr>
<tr>
<td>Config [III] 100 mm × 100 mm (4 in. × 4 in.)</td>
<td>No</td>
<td>Yes</td>
<td>Yes</td>
</tr>
<tr>
<td>Config [IV] 178 mm × 178 mm (7 in. × 7 in.)</td>
<td>No</td>
<td>No</td>
<td>No</td>
</tr>
</tbody>
</table>

Figure 2  Test setup for four configurations with the same test structure – Section views.
Figure 3  Config [I] – 100 mm by 100 mm (4 in. by 4 in.) with the frame and the holder.

Figure 4  Config [II] – 100 mm by 100 mm (4 in. by 4 in.) with the holder.
Figure 5  Config [III] – 100 mm by 100 mm (4 in. by 4 in.) with the holder and the side insulation.

Figure 6  Config [IV] – 178 mm by 178 mm (7 in. by 7 in.).
However, the structure for the four configurations was the same, *i.e.*, 1) Ten layers of 6.35 mm (¼ inch) thick ceramic fiberboard, PM board, were piled up on top of two layers of 12.7 mm (1/2 inch) thick ceramic fiberboard, HS board. The densities for the PM board and HS board from Thermal Ceramics Inc. are 240 $kg/m^3$ (15 pcf) and 448 $kg/m^3$ (28 pcf), respectively. 2) A total of 40 thermocouples were used to obtain temperatures at 10 interfaces. Four thermocouples per interface, *i.e.*, two at the center (Ctr), one in the middle (Mid), and one in the corner (Cnr), were installed as seen in Figure 7. As for thermocouples 28 – gauge type K thermocouples were used and the limits of error are $±1.1 °C$ or 0.4 %. 3) The manner of thermocouple installation was the same, *i.e.*, the bare wire was placed on the top surface of the lower layer as shown in case of *Surface* in the Appendix B. 4) For configurations where the frame was not used, four brads per layer were inserted at the corners of the specimen as a way of minimizing the air gap between layers. The details are described in the Appendix B.
A series of Cone tests for Configs [I], [II], [III], and [IV] were conducted for 95 kW/m² and then 20, 60, 80, and 100 kW/m² incident heat fluxes. The first test for 95 kW/m² is for the purpose of heat treatment. For each Cone test, the specimen was exposed to the heat for 1800 seconds after the shutter was opened and then removed from the heater. The maximum temperature occurred at 1800 seconds or afterward according to the depth. The cooling data were collected for another 1800 seconds.

2.4.2 Config [I]: 100 mm by 100 mm with the Frame and Holder

In addition to the 40 thermocouples in the ceramic fiberboard, 7 thermocouples were installed on the retainer frame and the specimen holder. Four thermocouples were
attached to the side of the frame: Two of them were placed ¼ inch below the top surface, \textit{i.e.}, at the same depth as the first interface (I1) for the ceramic fiberboard, I1-Fr, in Figure 8. The other two were near the bottom of the frame, \textit{i.e.}, at the same depth as the seventh interface (I7), I7-Fr, in Figure 8. For reference, three thermocouples were placed on the holder, two on the side of the holder, Hol-S, and one at the bottom of the holder, Hol-B, in Figure 8.

![Figure 8](image_url) Locations for the thermocouples in the frame (Top – Section view) and the specimen holder (Bottom – Plan view).

Temperatures for the frame and the holder were compared with the ones for the ceramic fiberboard as seen in Figure 9 and Figure 10 for 30 kW/m$^2$ incident heat flux (IHF). At the height of I1, the order from high temperature to low temperature is Center (Ctr) $\rightarrow$ Middle (Mid) $\rightarrow$ Corner (Cnr) $\rightarrow$ Frame (Fr), which was expected considering the surface non-uniformity of the incident heat flux from the Cone heater, see Appendix.
A. However, the order is totally opposite at the height of I7, Frame → Corner → Middle → Center. In addition, Center temperature at I7 is even lower than the specimen holder temperatures at the side (Hol-S) and bottom (Hol-B) up to 800 seconds. Temperatures for 5, 10, and 20 kW/m² provided exactly the same pattern and only the magnitude of the temperature was different. The fact that the frame temperatures, especially near the bottom of the frame, were high resulted from the high thermal conductivity of the stainless steel frame. Due to the heated frame, temperatures for Corner and Middle become higher than the one for Center at I7.
Figure 9  Temperatures for the frame and the holder compared to the ones for the ceramic fiberboard at I1 – 30 kW/m².

Figure 10  Temperatures for the frame and the holder compared to the ones for the ceramic fiberboard at I7 – 30 kW/m².
The edge effect due to the use of the retainer frame is clearly seen in Figure 11 and Figure 12 for 80 kW/m² IHF. The Corner temperature is greater than the ones for Middle and Center at all interfaces except I1 and I2 at 600 seconds (dotted oval in Figure 11). Temperatures for Middle begin to increase over time, at 800 and 1000 seconds (Appendix C), due to the lateral heat transfer from the hotter Corner. In the same manner as Middle temperatures, temperatures for Center increase over time as well. Finally, Center temperatures are the highest from I1 to I4 at 1800 seconds (Figure 12). The fact that temperatures near the corner were higher than at the center of the specimen indicates that considerable lateral heat transfer from the heated frame may impact on the center temperature rise. The section views at 200, 400, 800, and 1000 seconds can be found in the Appendix C.
Figure 11  Section views in Config [I] at 600 sec – 80 kW/m².

Figure 12  Section views in Config [I] at 1800 sec – 80 kW/m².
Based upon the temperature measurements, it was curious to quantify how much heat flows in the vertical direction in contrast to the lateral direction. In order to estimate the heat flow in both vertical and lateral directions, vertical heat fluxes between interfaces and lateral heat fluxes for [Corner – Middle] and [Middle – Center] were calculated using Fourier’s equation. The temperature-dependent thermal conductivity* of ceramic fiberboard provided by Thermal Ceramics Inc. was used for the calculation. The heat fluxes were described with arrows so that it would be easier to see the direction of heat flows and the magnitude of heat fluxes visually. The width of an arrow is proportional to its magnitude.

Figure 13 shows a 3D thermal map for Config [I] from I1 to I9 at 600 seconds. The vertical heat transfer at Center is dominant from I1 to I4 and the magnitude for the vertical heat flux at Corner is less than half of the one at Center at I2. But the magnitude at Corner is the same as the one at Center at I5. The magnitudes at Corner after I5 towards the bottom are significantly greater than at Center. For the lateral heat fluxes, it is observed that the direction of the lateral heat flows is changed from [Middle → Corner] (from I1 to I2) to [Corner → Middle] (from I3 to I8). In the case of I7 at 600 seconds, the vertical heat flux at Corner is 300 W/m² and the lateral heat flux from Corner to Middle is 200 W/m² while there is no heat flow at Center. This 3D thermal map, especially at I5, I6, I7, and I8, shows the effect of the use of the frame.

* Thermal properties of the ceramic fiberboard have been provided by Thermal Ceramics Inc subcontractor, Dr. Ned Keltner of Ktech Corporation. Thermal conductivity has been determined based on data from ASTM C 201 and Ktech property estimation software.
The arrows for the lateral heat flux between Middle and Center are not seen in Figure 13. However, it should not be overlooked that temperatures at Middle are greater than Center temperatures from I3 to I9. In the case of I5 and I6, the temperature difference is up to 15 K. Thus, there is a possibility that Center temperatures have been affected by the lateral heat flux from Corner towards Center and the significant vertical heat flux near the bottom of the frame over time.

The change of the 3D thermal map over time can be found in the Appendix D.
Figure 13  3D Thermal map in Config [I] at 600 seconds – 80 kW/m$^2$ IHF.
2.4.3 Config [II]: 100 mm by 100 mm with the Holder

The motivation of Config [II] resulted from the thermal behavior of the specimen in Config [I]. Considering the possibility that the heat gain through the retainer frame affects the temperature at the center of the specimen, the test setup for Config [II] was the same as the one for Config [I] and the only difference was without using the frame. It was expected that Center temperature profiles in Config [II] would be less affected than in Config [I] because the frame was removed.

The section views are shown in Figure 14 and Figure 15 for 80 kW/m² IHF. Center temperatures are higher than Corner temperatures from I1 to I3, but Corner temperatures are the highest from I4 to I9 at 600 seconds. Compared to Config [I], temperature rises at Corner are less significant. At 1800 seconds, temperature order from the highest is Center → Middle → Corner from I1 through I8, which indicates that there is not heat gain from the edge. However, the slopes from Middle to Corner are steeper from I1 towards the bottom such as I5 in comparison to Config [I]. Center temperatures in Config [II] are similar with the ones in Config [I] at 600 seconds, but at 1800 seconds Center temperatures in Config [II] are up to 40 K lower than in Config [I]. It is thought to be due to heat loss around the edge of the specimen.

Details of the section views at 200, 400, 800, and 1000 seconds are in the Appendix C.
Figure 14  Section views in Config [II] at 600 sec – 80 kW/m².

Figure 15  Section views in Config [II] at 1800 sec – 80 kW/m².
The vertical and lateral heat fluxes at 600 seconds are illustrated in Figure 16. The magnitudes for the vertical heat flux at Center are close to the ones in Config [I]. The vertical heat fluxes at Corner at I6, I7, and I8 are smaller than in Config [I]. For the lateral heat fluxes, the value at I1, 500 W/m², is noticeable, which results from the lower Corner temperature than in Config [I]. From I5 to I7 less heat moves from Corner to Middle than in Config [I]. Overall the heat flux pattern in Config [II] is not considerably different from in Config [I]. However, Corner temperatures from I2 to I6 in Config [II] are almost 100 K lower than the ones in Config [I]. It indicates that there is heat loss to the surrounding air.

There is a little difference between Middle temperature and Center temperature from I1 to I3. However, Middle temperature is 15 K higher than Center temperature at I6. It would be reasonable to say that Center temperatures in Config [II] are not considered 1D. More details will be discussed in Section 2.4.6.

The lateral and vertical heat fluxes over time are described in the Appendix D.
Figure 16 3D thermal map in Config [II] at 600 seconds – 80 kW/m$^2$ IHF.
2.4.4 Config [III]: 100 mm by 100 mm with the Holder and Side Insulation

It was learned in Configs [I] and [II] that the edge condition either with the retainer frame or without it might have an effect on temperature rise at the center of the specimen. In order to minimize heat gain or loss through the edge of the specimen, the side of the specimen was wrapped with an insulation material (ceramic blanket).

In the case of Config [III] a different pattern of section views from in Configs [I] and [II] is seen in Figure 17 and Figure 18 for 80 kW/m² IHF. Among three temperature measurements at Center, Middle, and Corner Middle temperature is the highest at I1 and Corner is the highest from I2 to I9 at 600 seconds. The pattern is more obvious at 1800 seconds. Middle temperature is the highest from I1 to I3. Compared to Center temperatures in Configs [I] and [II], Center temperature in Config [III] is the highest at all interfaces at 600 and 1800 seconds. Center temperatures in Config [III] are up to 20 K and 30 K higher than in Config [I] at 600 and 1800 seconds, respectively.

More section views over time including 200, 400, 800, and 1000 seconds are described in the Appendix C.
Figure 17  Section views in Config [III] at 600 sec – 80 kW/m².

Figure 18  Section views in Config [III] at 1800 sec – 80 kW/m².
Figure 19 shows a 3D thermal map with vertical and lateral heat fluxes in Config [III]. The vertical heat fluxes at Center are slightly higher than the ones in Configs [I] and [II] at all interfaces. The main difference from Configs [I] and [II] is significantly larger vertical heat fluxes at Corner from I1 to I5. Especially the ones at Corner from I2 to I5 in Config [III] are approximately double the values in Config [I]. In contrast the vertical Corner heat fluxes at I7 and I8 in Config [III] are smaller than the ones in Config [I]. For the lateral heat fluxes, heat flows from Corner to Middle and from Middle to Center. Middle temperatures are 20 K to 25 K greater than Center temperatures near the top surface, from I2 to I4, while it occurs near the bottom, at I5 and I6, in Configs [I] and [II].

More heat flux at Corner in Config [III] than in Configs [I] and [II] is due to the fact that the thermal diffusivity, $\alpha$, of the ceramic blanket is higher than the one of the ceramic fiberboard. Thus, the direction of the heat flow is from the ceramic blanket to the ceramic fiberboard and it causes Corner temperature higher than Center temperature. Rough estimation of the thermal properties for ceramic fiberboard and ceramic blanket is seen in Table 2. The thermal property data were obtained from Thermal Ceramics Inc.

<table>
<thead>
<tr>
<th></th>
<th>Thermal conductivity (W/m.K)</th>
<th>Density (kg/m$^3$)</th>
<th>Specific heat (kJ/kg.K)</th>
<th>Thermal diffusivity (m$^2$/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ceramic Board</td>
<td>0.058 at 533 K</td>
<td>240</td>
<td>0.88 at 533 K</td>
<td>0.00027</td>
</tr>
<tr>
<td>Ceramic blanket</td>
<td>0.068 at 533 K</td>
<td>96</td>
<td>0.88 at 533 K</td>
<td>0.00080</td>
</tr>
</tbody>
</table>
It was realized through the analysis for Config [III] that temperature measurement only at the center would not be sufficient to understand the heat transfer behavior of a specimen. Comparison of Center temperatures among Configs [I], [II], and [III] shows Config [III] is the highest. However, it does not mean that Config [III] provides one-dimensional heat conduction. It is desirable to measure temperatures not only at the center of the specimen but also in the middle at minimum.

Appendix D includes more 3D thermal maps over time.
Figure 19 3D thermal map in Config [III] at 600 seconds – 80 kW/m² IHF.
2.4.5 Config [IV]: 178 mm by 178 mm

Config [IV] was motivated by the results from Config [III] as an extension of Config [III]. Unlike Configs [I], [II], and [III], a bigger specimen size, 178 mm by 178 mm (7 inch by 7 inch), was used without the frame and holder. The heat flux deviation from the center value at the edge of the specimen is shown in Figure 20. It is reminded that the irradiance within the central 50 by 50 – mm area (hatched area in Figure 20) of the specimen should be ±2 % according to ASTM E-1354³.

![Figure 20](image)

From the viewpoint of an overall section view at depth, a consistency between 600 (Figure 21) and 1800 seconds (Figure 22) is observed in Config [IV]. The consistency is that there is little difference between Center and Middle temperatures and Corner temperature is always lowest. Appendix C includes section views at 200, 400, 800, and 1000 seconds.
Figure 21  Section views in Config [IV] at 600 sec – 80 kW/m².

Figure 22  Section views in Config [IV] at 1800 sec – 80 kW/m².
How much heat flows in the lateral and vertical directions is described in Figure 23. The vertical heat fluxes at Center in Config [IV] are almost same as the ones in Config [III]. This slight difference between Configs [III] and [IV] was caused by different vertical heat flux at Corner and lateral heat flux. The vertical heat flux at Corner at I1 in Config [IV], 1800 W/m², is higher than the one in Config [III], 1500 W/m². It is noted that the Corner magnitudes in Configs [I] and [II] are only 1100 and 1200 W/m², respectively. Unlike at I1, the vertical heat fluxes at Corner from I3 to I8 in Config [IV] are lower than the ones in Config [III].

Regarding the lateral heat fluxes in Config [IV] only a couple of arrows are shown in Figure 23. It appears that there is little lateral heat flow in Config [IV] compared to Configs [I], [II], and [III]. Mostly, the directions of the heat flow are from Corner to Middle and Middle to Center and Middle temperatures are higher than Center temperatures in Configs [I], [II], and [III]. However, Config [IV] exhibits a unique tendency, i.e., the lateral heat flows always from Center to Middle and from Middle to Corner and Center temperatures are higher than Middle temperatures at all the interfaces.

3D thermal maps at 200, 1000, 1400, and 1800 seconds are illustrated in the Appendix D. A consistency is found out for all times in Config [IV] that the magnitudes of vertical heat flux at Corner are slightly smaller than the ones at Center at all interfaces.
Figure 23 3D thermal map in Config [IV] at 600 seconds – 80 kW/m² IHF.
2.4.6 Comparison of the Results from Four Configurations

The name of Cone Calorimeter is from the conical shape of the heater. The diameter of the shell of the heating element is 160 mm$^2$. The specimen size is 100 by 100 mm – rectangular shape. It is well known that the irradiance within the central 50 by 50 – mm area is ±2 %. It was questioned if the non–uniform heat flux is confounded with the edge condition since Middle temperature was measured near the boundary of the central 50 by 50 – mm area (See Figure 24). It was needed to decouple non–uniform surface heat flux and edge condition and ALGOR finite element analysis software$^{10}$ was used as a way of resolving the confounded problem.

The distance from the center to the corner of the 50 by 50 – mm area is 35.35 mm and to the corner of the 100 by 100 – mm area is 70.7 mm. It is noted that the distance from the center to where the thermocouple for Middle was installed is 32.5 mm. The simulated region is 71 mm wide and 64 mm long because of symmetry as described in Figure 25. The width, 64 mm, comes from the depth of 10 layers of ¼ inch PM board.

ALGOR simulation requires heat flux inputs. Surface heat flux in the Cone Calorimeter were measured at the center of the specimen, corner of the 50 by 50 – mm area, and corner of the 100 by 100 – mm area (Figure 25). A total of 71 surface heat fluxes were applied as inputs based upon the linear relationship calculation using the three heat flux measurements.
ALGOR run was done for 600 seconds. The surface temperature profiles from the simulation with non-uniform heat flux were compared with those from the simulation with uniform (constant) heat flux. Comparison of those two temperature profiles at the corner of the 50 by 50-mm area shows 1.4 K difference. The two
temperatures for non-uniform vs. uniform heat flux were compared at where Middle thermocouple was installed and the difference was only 0.4 K. Thus, it would be reasonable to say that the non-uniform surface heat flux and edge condition are not confounded with temperature measurements at Middle.

Now Configs [I], [II], [III], and [IV] are compared based upon the temperature measurements at Center, Middle, and Corner. Overall more attention will be paid to Configs [I] and [IV] than to Configs [II] and [III] since Config [I] is a typical Cone test setup and the behavior in Config [IV] is close to 1D compared to the other configurations.

First, the 3D thermal maps, i.e., vertical heat fluxes with arrows, at the first interface (I1) at 600 seconds were compared for the four configurations as shown in Table 3. It would be ideal if accurate surface temperature measurements were taken and used for the comparison. However, accurate measurements of the surface temperature are quite difficult to make. Therefore, temperature measurements at I1 were used instead of the surface temperatures since I1 is close to the surface. The first row is the type of configuration. The second, third, and fourth rows show the magnitude of the vertical heat flux from I1 to I2 at Corner, Middle, and Center, respectively. The fifth is the second row divided by the fourth to see how much heat flows at Corner as opposed to at Center. The sixth is the third row divided by the fourth to see the heat flow between Middle and Center. The percentage for Corner vs. Center in Config [I] is approximately 50% while the value in Config [IV] is close to 90%. For the vertical heat fluxes between Middle
and Center, the percentage in Config [I], 97 %, is almost same as the one in Config [IV], 98 %.

Table 4 compares the lateral heat fluxes at I1 at 600 seconds for the four configurations. The second and third rows are the magnitude of the lateral heat flux from Middle to Corner and from Center to Middle, respectively. The fourth is the magnitude of the vertical heat flux from I1 to I2 at Center. The fifth is the second row divided by the fourth one, which shows how significant the lateral heat flow is compared to the vertical flow at Center. The sixth is the third divided by the fourth to compare lateral and vertical heat fluxes at Center. The percentages for lateral heat flux [Middle – Corner] vs. vertical heat flux are 14 % in Config [I] and 8 % in Config [IV] while the values for lateral heat flux [Center – Middle] vs. vertical heat flux are 1% in Configs [I] and [IV].

Table 3  
Comparison of the vertical heat fluxes for the four configurations at I1 at 600 seconds.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical heat flux (W/m²)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cnr at I1 → Cnr at I2</td>
<td>1050</td>
<td>1240</td>
<td>1500</td>
<td>1810</td>
</tr>
<tr>
<td>Vertical heat flux (W/m²)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mid at I1 → Mid at I2</td>
<td>1940</td>
<td>2110</td>
<td>2060</td>
<td>2090</td>
</tr>
<tr>
<td>Vertical heat flux (W/m²)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ctr at I1 → Ctr at I2</td>
<td>2000</td>
<td>2040</td>
<td>2060</td>
<td>2150</td>
</tr>
<tr>
<td>Vert. HF (Cnr) ×100 (%)</td>
<td>52</td>
<td>61</td>
<td>73</td>
<td>87</td>
</tr>
<tr>
<td>Vert. HF (Ctr) ×100 (%)</td>
<td>97</td>
<td>103</td>
<td>100</td>
<td>98</td>
</tr>
</tbody>
</table>

38
Table 4: Comparison of the lateral heat fluxes for the four configurations at I1 at 600 seconds.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Lateral heat flux (W/m²)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cnr at I1 ← Mid at I1</td>
<td>-280¹</td>
<td>-460¹</td>
<td>-30¹</td>
<td>-180¹</td>
</tr>
<tr>
<td>Lateral heat flux (W/m²)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mid at I1 ← Ctr at I1</td>
<td>-10²</td>
<td>0</td>
<td>50</td>
<td>-20²</td>
</tr>
<tr>
<td>Vertical heat flux (W/m²)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ctr at I1 → Ctr at I2</td>
<td>2000</td>
<td>2040</td>
<td>2060</td>
<td>2150</td>
</tr>
<tr>
<td>Lat. HF (Cnr ← Mid) ×100 (%)</td>
<td>14³</td>
<td>23³</td>
<td>1³</td>
<td>8³</td>
</tr>
<tr>
<td>Vert. HF (Ctr)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lat. HF (Mid ← Ctr) ×100 (%)</td>
<td>1³</td>
<td>0</td>
<td>2</td>
<td>1³</td>
</tr>
</tbody>
</table>

¹: heat flow direction from Cnr to Mid, not from Mid to Cnr.
²: heat flow direction from Mid to Ctr, not from Ctr to Mid.
³: the percentage using the absolute value of the magnitude.

The behaviors at Corner vertically (Table 3) and laterally (Table 4) are explained with the different edge condition. For the comparison between Middle and Center, the percentages of vertical and lateral heat fluxes in Config [I] are not differentiated from the ones in Config [IV]. The heat flux value itself in Config [I] does not differ significantly from the value in Config [IV]. Thus, the propagation of uncertainties through the heat flux calculation was questioned.

The uncertainty can be calculated from an equation by Taylor¹¹. It is assumed that $x_1,...,z$ are measured with uncertainties $\delta x_1,...,\delta z$ and the measured values are used to
compute the function \( q(x_1, ..., x_n) \). If the uncertainties in \( x_1, ..., x_n \) are independent and random, then the uncertainty in \( q \) is as follows:

\[
\delta q = \sqrt{\left( \frac{\partial q}{\partial x_1} \delta x_1 \right)^2 + \cdots + \left( \frac{\partial q}{\partial x_n} \delta x_n \right)^2}
\]

(1)

As a first step, Taylor suggests calculating the partial uncertainties in \( q \) due to \( \delta x, \delta y, ..., \delta z \) separately, using the following equation.

\[
\delta q_x = \left| \frac{\partial q}{\partial x} \right| \delta x \quad \text{(if } \delta y = \cdots = \delta z = 0 \text{)}
\]

(2)

Then combine these separate uncertainties to give the total uncertainty. It is noted that \( \delta q_x \) is the uncertainty in \( q \) due to \( \delta x \) alone.

For the 3D thermal maps, the heat flux, \( q' \), was calculated according to the Fourier’s equation. Therefore, the uncertainty exists in a function of three variables, thermal conductivity, \( k \), Temperature, \( T \), and distance normal to the surface, \( y \). Using Taylor’s equation the expression for the uncertainty is as follows:

\[
\delta q' = \sqrt{\left( \frac{\partial q'}{\partial k} \delta k \right)^2 + \left( \frac{\partial q'}{\partial T} \delta T \right)^2 + \left( \frac{\partial q'}{\partial y} \delta y \right)^2}
\]

(3)

The total uncertainty in \( q' \), 100 W/m², was estimated according to Equation 3. Considering the uncertainty in \( q'' \), the vertical heat fluxes at Center and Middle in Config [I] are not clearly differentiated from the ones in Config [IV] while it is apparent to see
the difference of the two values at Corner. For the lateral heat fluxes, it would be fair to say that there is no difference between Configs [I] and [IV].

Even though the insignificant differences at I1 in Configs [I] and [IV] were buried in the uncertainty, it is valuable to learn overall heat transfer behaviors of the whole specimen through the 3D thermal mapping, which was illustrated simply in Figure 26. It is clearly shown in Config [I] that heat loss occurs near the upper part of the frame while there is heat gain toward the bottom of the frame. Config [II] shows heat loss through the side overall. On the contrary heat gain is observed in Config [III] in general. Compared to the other configurations there is no heat gain or loss in Config [IV]. It is noted what was described in Figure 26 is the overall tendency for the four configurations and the width of an arrow is irrelevant to the magnitude of the edge effect.

Considering the magnitude of the edge effect for the configurations, the following can be summarized:

- 3D heat conduction occurred when using the specimen size, 100 mm by 100 mm (4 inch by 4 inch), regardless of whether the retainer frame is used or not.
- By wrapping the side of the specimen with insulation the center temperature data were affected less by the 3D heat transfer behavior than without insulating the side.
- The 178 mm by 178 mm (7 inch by 7 inch) ceramic fiberboard specimen provides the best result in terms of approximating 1D heat conduction.
Figure 26  Effect of the edge condition for the four configurations.

Center temperatures at I1 in Configs [I], [II], [III], and [IV] are compared in Table 5. The temperature difference is taken as the temperature for Config [IV] subtracted from the temperature for each configuration. In early stages such as 200 and 600 seconds, temperatures in Config [III] are the same as the ones in Config [IV] while temperatures in Configs [I] and [II] are 20 K lower. After 600 seconds, the temperature difference between Configs [III] and [IV] becomes slightly larger with time. However,
overall Center temperatures in Config [III] are still much closer to the ones in Config [IV] than in Configs [I] and [II]. Temperature profiles for Center, Middle, and Corner at each interface are seen in the Appendix E.

Table 5  
Comparison of Center temperatures at I1 in four configurations – 80 kW/m² IHF.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>200</td>
<td>704</td>
<td>-23</td>
<td>705</td>
<td>-22</td>
</tr>
<tr>
<td>600</td>
<td>831</td>
<td>-23</td>
<td>836</td>
<td>-18</td>
</tr>
<tr>
<td>1000</td>
<td>871</td>
<td>-19</td>
<td>871</td>
<td>-19</td>
</tr>
<tr>
<td>1400</td>
<td>890</td>
<td>-19</td>
<td>887</td>
<td>-22</td>
</tr>
<tr>
<td>1800</td>
<td>902</td>
<td>-17</td>
<td>892</td>
<td>-27</td>
</tr>
</tbody>
</table>

1: \( T_{Config \ [I]} - T_{Config \ [IV]} \)

2: \( T_{Config \ [II]} - T_{Config \ [IV]} \)

3: \( T_{Config \ [III]} - T_{Config \ [IV]} \)

Table 6 presents Center, Middle, and Corner temperatures at I1 at 600 seconds in four configurations. Center and Middle temperatures in Config [I] are more than 20 K lower than the ones in Config [IV]. The temperature difference becomes larger at Corner, 60 K. It is noticed that Center temperature in Config [III] is the same as the one in Config [IV] while Middle temperature in Config [III] is 20 K higher than the one in Config [IV].
Table 6  
Comparison of Center, Middle, and Corner temperatures at I1 at 600 seconds in four configurations – 80 kW/m² IHF.

<table>
<thead>
<tr>
<th>Thermocouple location</th>
<th>Config [I] [K]</th>
<th>Config [II] [K]</th>
<th>Config [III] [K]</th>
<th>Config [IV] [K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Center</td>
<td>831</td>
<td>836</td>
<td>854</td>
<td>854</td>
</tr>
<tr>
<td>Middle</td>
<td>827</td>
<td>837</td>
<td>871</td>
<td>848</td>
</tr>
<tr>
<td>Corner</td>
<td>733</td>
<td>676</td>
<td>863</td>
<td>791</td>
</tr>
</tbody>
</table>

Center temperature between Configs [I] and [IV] at various heat fluxes are compared in Table 7. For heat flux, 80 kW², Center temperature difference, 20 – 25 K, is observed not only at 600 seconds but also in an early times such as 100 and 200 seconds. As heat flux is lower, the magnitude of the difference becomes smaller. In case of low heat fluxes such as 20 and 40 kW/m², the temperature difference is 10 – 15 K.

Table 7  
Comparison of Center temperature difference between Configs [I] and [IV] at I1 – various heat fluxes.

<table>
<thead>
<tr>
<th>IHF (kW/m²)</th>
<th>100 s</th>
<th>200 s</th>
<th>300 s</th>
<th>400 s</th>
<th>500 s</th>
<th>600 s</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>6</td>
<td>9</td>
<td>10</td>
<td>11</td>
<td>11</td>
<td>13</td>
</tr>
<tr>
<td>20</td>
<td>11</td>
<td>13</td>
<td>13</td>
<td>13</td>
<td>13</td>
<td>14</td>
</tr>
<tr>
<td>40</td>
<td>11</td>
<td>13</td>
<td>15</td>
<td>14</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>60</td>
<td>14</td>
<td>17</td>
<td>16</td>
<td>16</td>
<td>16</td>
<td>19</td>
</tr>
<tr>
<td>80</td>
<td>21</td>
<td>23</td>
<td>22</td>
<td>22</td>
<td>23</td>
<td>23</td>
</tr>
</tbody>
</table>
From the results from the comparison of temperature at I1, it was curious to see temperature difference at the surface between Configs [I] and [IV]. Unfortunately accurate surface temperatures were not measured due to the difficulty of its measurement. Based upon Center temperature measurements at depth from I1 to I9 at 600 seconds, surface temperature was predicted using the best – fit line to temperature vs. depth as shown in Figure 27. As temperature measurement was taken closer to the surface, the temperature difference between Configs [I] and [IV] becomes larger. According to the tendency, the temperature difference at the surface is predicted to be 30 – 35 K roughly. Therefore, more attention is called when surface temperature measured in the Cone Calorimeter is utilized with a 1D equation. The difference, 30 – 35 K, can cause the results underestimated.

![Figure 27](image_url)  

Figure 27  Rough estimate of surface temperature at 600 seconds based upon temperature measurements at depth using curve fit – 80 kW/m² IHF.
Ideally, Middle temperature should be the same as Center temperature if the one-dimensional heat conduction assumption is correct as has been assumed implicitly. However, Middle temperature shows a difference from Center temperature according to each configuration. The temperature difference, $\Delta T = T_{Mid} - T_{Ccr}$, is Center temperature subtracted from Middle temperature at each interface. For Config [I], most of temperature differences are positive during the heating period, i.e., up to 1800 seconds, which means Middle temperature is greater than Center temperature. The variation of the temperature difference is up to 70 K as seen in Figure 28. The variation in Config [II] is similar with the one in Config [I]. However, many temperature differences in Config [II] show that Center temperature is greater than Corner temperature. In the case of Config [III], the variation is 40 K, which is approximately half of the value in Configs [I] and [II]. The temperature difference in Config [III] is smaller, but Corner temperature is higher than Center temperature. It means that Config [III] still has an edge effect even though the magnitude of the effect is not as much as in Configs [I] and [II]. On the contrary, Center temperature is the highest compared to Middle temperatures for all times in Config [IV]. Furthermore, the variation for the temperature difference is only 10 K (See Figure 29). Please refer figures for temperature differences in Configs [II] and [III] in the Appendix F.

According to ASTM E-1354, the irradiance within the central 50 by 50 – mm area of the specimen should be ±2 %. The value, ±2 %, is generally thought to be negligible and therefore the 50 by 50 – mm area is considered uniform, which may be acceptable in
some applications. However, special attention is needed with current Cone test setup when data are measured at depth not only at the center of the specimen but also near the 50 by 50 – mm area. The data measured at the center might be considered 1D if a high accuracy of the 1D data is not required. However, the data taken in the 50 by 50 – mm area will be significantly affected by the edge condition. The temperature in the area will be up to 40 K higher than Center temperature for 80 kW/m$^2$. For low heat fluxes such as 40 kW/m$^2$, the temperature in the area will be up to 15 K higher in the upper layers and 15 K lower in the bottom layers than Center temperature. Temperature measurements kept apart from the center would not be as good as the one at the center due to the edge effect even if temperature measurements were taken in 50 by 50 – mm area.
Figure 28  Temperature difference, $\Delta T = T_{mid} - T_{cur}$, in Config [I] – 80 kW/m$^2$ IHF.

Figure 29  Temperature difference, $\Delta T = T_{mid} - T_{cur}$, in Config [IV] – 80 kW/m$^2$ IHF.
A similar pattern with Middle and Center temperatures is observed with Corner and Center temperatures. The only difference is that the magnitude of the temperature difference, $\Delta T = T_{Cnr} - T_{Ctr}$, is much larger than the temperature difference, $\Delta T = T_{Mid} - T_{Ctr}$, as expected due to the edge effect. The variation of the temperature difference in Config [I] is up to 300 K (Figure 30) while the variation for Config [IV] is 70 K (Figure 31). The temperature difference, $\Delta T = T_{Cnr} - T_{Ctr}$, for each configuration is seen in the Appendix F.
Figure 30 Temperature difference, $\Delta T = T_{\text{car}} - T_{\text{cr}}$, in Config [I] – 80 kW/m² IHF.

Figure 31 Temperature difference, $\Delta T = T_{\text{car}} - T_{\text{cr}}$, in Config [IV] – 80 kW/m² IHF.
2.5 An Example of Practical Application: Inert Material Ignition Analysis

The current methods for analyzing ignition data are based on the assumption that piloted ignition occurs when the material reaches a critical temperature, \( T_{ig} \). A number of studies have been published which show procedures to obtain ignition properties. The procedures consider the solid to be inert, \( i.e. \), heats of reaction and phase change can be neglected. The heat transfer problem in the solid phase is treated as one – dimensional heat conduction\textsuperscript{12}.

Since the ceramic fiberboard is inert, no ignition will occur in reality. The benefit of inert materials is that a range of ignition temperature can be readily investigated. Ignition temperatures for common combustible solids vary approximately from 300 °C (573 K) to 500 °C (773 K)\textsuperscript{13}. Thus, an ignition temperature is chosen at 600, 700, and 800 K.

In order to investigate the effect of ignition temperatures, Configs [I] and [IV] are chosen. Config [I] is a typical Cone test setup which undergoes 3D conduction. Config [IV] provides the best 1D heat conduction behavior compared to other configurations. For the ignition analysis it is required to know the ignition temperature at the surface of a material with interest. It is well known that accurate measurements of the surface temperature are quite difficult to make. The surface temperature measurement of the ceramic fiberboard was attempted by embedding thermocouples (See Appendix G), but satisfactory results were not obtained. Thus, ignition temperatures were taken from the
results of the 1D heat conduction model which was developed for the heat flux mapping procedure in Section 3.0. Further details about the 1D model will be mentioned in Section 3.4.

The temperature data from the 1D model is absolutely 1D while the ones from Config [IV] is close to 1D compared to other configurations. The temperature difference between the two data is shown in Figure 32 (blue line with a diamond shape). Surface temperatures in Config [I] are needed as well for the analysis. The same results from the 1D model were compared to the temperature data from Config [I] (pink line with a square in Figure 32). For the purpose of simulating Config [I], the temperature – dependent thermal conductivity which was used in the 1D model (Equation 4) was shifted up and down and the thermal diffusivity was changed accordingly. In case that the original thermal conductivity – 0.006 W/m.K (Equation 5) was used in the model (yellow line with a triangle), the temperatures from the 3D simulation show the least deviation from the ones from the 1D simulation. Thus, this corrected thermal conductivity, \( k(T) \), the original thermal conductivity – 0.006 W/m.K, was used to predict the surface temperature in Config [I]. The corrected equation is an effective 1D property used to approximately compensate for 3D behavior (bias) when using a 1D model.

\[
k(T)_{\text{original}} = -2.15 \times 10^{-12} \times T^3 + 7.36 \times 10^{-8} \times T^2 + 9.87 \times 10^{-6} \times T + 0.032 \quad (4)
\]

\[
k(T)_{\text{corrected}} = -2.15 \times 10^{-12} \times T^3 + 7.36 \times 10^{-8} \times T^2 + 9.87 \times 10^{-6} \times T + 0.026 \quad (5)
\]

where \( k \) is in [W/m.K] and \( T \) is in [K].
Figure 32  Temperature differences between the data from experiment and model with various thermal conductivities.

The estimated ignition times in Configs [I] and [IV] when ignition temperature is at 700 K are shown in Table 8. The differences between the two configurations appear to be insignificant and the percentages are less than 10 %. It would be interesting to see if this difference causes a different result between Configs [I] and [IV] in the ignition analysis.
Table 8 The estimated ignition times in Configs [I] and [IV] and the percentage of two ignition times at 700 K.

<table>
<thead>
<tr>
<th>$q''_e$ (kW/m²)</th>
<th>Config [I] (sec)</th>
<th>Config [IV] (sec)</th>
<th>$\left(\frac{(t_{ig, Config[IV]} - t_{ig, Config[I]})}{t_{ig, Config[I]}}\right) \times 100$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>387</td>
<td>403</td>
<td>4</td>
</tr>
<tr>
<td>30</td>
<td>25.3</td>
<td>16</td>
<td>5</td>
</tr>
<tr>
<td>40</td>
<td>4.9</td>
<td>5.3</td>
<td>8</td>
</tr>
<tr>
<td>50</td>
<td>2.4</td>
<td>2.6</td>
<td>7</td>
</tr>
</tbody>
</table>

Plotting ignition times, $t_{ig}^{-n}$, versus the external heat flux, $q''_e$, shows if the material is thermally thick or non-thick\textsuperscript{14}. If $n$ is close to 0.5, the material is considered thick. But if $n$ is close to 1, the material behaves as a non–thick material and additional data points are needed in the high heat flux range. With the data in Configs [I] and [IV], $t_{ig}^{-n}$ vs. $q''_e$ was plotted to compare the values of $n$. The best fit line to the data ($R^2$) was acquired when $n=0.5$ at $T_{ig}=700$ K in both of Configs [I] (Figure 33) and [IV] (Figure 34). The lines in Figure 33 look same as the ones in Figure 34. But the slopes of the lines between the two configurations are slightly different.
Figure 33  Ignition temperature, 700 K, in Config [I].

Figure 34  Ignition temperature, 700 K, in Config [IV].
According to Janssens’ method\textsuperscript{15} to determine ignition properties, the ignition time, \( t_{ig}^{-0.55} \), versus the external heat flux, \( q''_e \), were plotted in Configs [I] and [IV] when ignition temperature is at 700 K (Figure 35). From plotting \( t_{ig}^{-0.55} \) versus \( q''_e \) the critical heat flux for ignition, \( q''_{cr} \), was determined by calculating the intercept of the line with abscissa in Configs [I] and [IV]. The value for the critical heat flux in Config [I], 18.31 kW/m\(^2\), is turned out to be the same as the one in Config [IV], 18.25 kW/m\(^2\). The slopes of the line were compared between in Configs [I] and [IV]. There is a slight difference with the slopes between the two as seen in Figure 35.

\[ y = 0.0186x - 0.3395 \]

\[ y = 0.0194x - 0.3552 \]

Figure 35 Inert material ignition analysis in Configs [I] and [IV] using Janssens’ method.
Janssens estimates the thermal inertia, \( k\rho c \), from the slope of the straight line. Janssens’s equation for \( k\rho c \) is shown in Equation 6. The corresponding values were calculated based on the ignition analysis of Configs [I] and [IV]. It is noted that the obtained value for \( q''_{cr} \) can be used instead of \( q''_{min} \) for a truly inert material.

\[
k\rho c = h^2_{eff} \left[ \frac{B_{ig}}{0.73q''_{min}} \right]^{1.828}
\]

(6)

where \( h_{eff} = \alpha_s \frac{q''_{min}}{T_{ig} - T_{\infty}} \)

(7)

\[
B_{ig} = \frac{1}{\text{slope}}
\]

(8)

The determined \( k\rho c \) for Configs [I] and [IV] were then compared to \( k\rho c \) used in modeling both Configs [I] and [IV] with the 1D model (Table 9). The thermal properties used in the 1D model were obtained from Thermal Ceramics, Inc. The difference of \( k\rho c \) between Configs [I] and [IV] from Janssens’ method, 0.0012 kJ²/m⁴.s.K² (7.5 %), is similar with the difference of \( k\rho c \) in the model, 0.0013 kJ²/m⁴.s.K² (8 %). The computed \( k\rho c \) from Janssens’ method is slightly different from the \( k\rho c \) in the model, 0.0009 kJ²/m⁴.s.K² (6 %), in both cases of Configs [I] and [IV]. It is noted that this comparison is somewhat circular but allows for a sense of the \( k\rho c \) from the ignition analysis assuming that the separate values are “correct”.

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Table 9 Comparison of the $k\rho c$ calculated using Janssens’ equation with the $k\rho c$ used in the 1D model.

<table>
<thead>
<tr>
<th></th>
<th>Config [I]</th>
<th>Config [IV]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k\rho c$ from Janssens’ method</td>
<td>0.0153 kJ²/m⁴.s.K²</td>
<td>0.0165 kJ²/m⁴.s.K²</td>
</tr>
<tr>
<td>$k\rho c$ used in 1D model</td>
<td>0.0144 kJ²/m⁴.s.K²</td>
<td>0.0156 kJ²/m⁴.s.K²</td>
</tr>
</tbody>
</table>

Through the ignition analysis including $t_{ig}$, $n$ value, $q''_{cr}$, and the slope of the line slight discrepancies were found using the data in Config [I] which has a 3D effect and Config [IV] which data are close to 1D. Overall the analysis presented does not show a significant difference between Configs [I] and [IV]. That is, the ignition analysis did not differentiate 3D data from 1D data. Two things are considered for the possible reason of the similar results in the ignition analysis. One might say that the experimental data from Config [IV] are not 100% 1D unlike the data from the 1D model. However, it was presented how well a 1D inert theory worked with the 1D data from Config [IV]. It indicates that the experimental data from Config [IV] is good enough to be called “1D”. The other reason would be that the simulated data for Config [I] were unrefined due to the difficulty of finding appropriate properties to model Config [I]. Several things were attempted to simulate Config [I], but they were not successful. The corrected thermal conductivity for Config [I] was an effective 1D property to compensate for 3D behavior when using a 1D model. This reason seems to cause that the 1D data were not differentiated from the 3D data through the ignition analysis.
2.6 Conclusions

As a bench-scale apparatus Cone Calorimeter has been used extensively for various purposes. The stainless steel retainer frame (edge frame) is primarily used to reduce unrepresentative edge burning of specimens\(^3\). In general, the frame has been used in most Cone tests without enough understanding of its effect. It has been implicitly assumed that the heat conduction in the Cone Calorimeter is one-dimensional. However, the assumption has not been corroborated explicitly to date. A few studies\(^4,5,6,7,8,9\) have shown that the frame reduced the peak heat release rate and when using it the time to ignition was somewhat longer. The effect on the edge condition of the specimen needed to be investigated directly related to the thermal behavior of the three-dimensional specimen.

Cone Calorimeter tests with ceramic fiberboard were conducted with four different configurations with and without the frame and for two different specimen sizes, 100 mm by 100 mm and 178 mm by 178 mm. 3D thermal maps of the 3D specimen were created and compared among the configurations. It is valuable to learn overall heat transfer behaviors of the whole specimen through the 3D thermal mapping. 3D heat conduction occurred when using the specimen size, 100 mm by 100 mm, regardless as to whether the frame is used. This is due to the effect of the edge condition, \(i.e.,\) heat loss or heat gain through the edge. Thus, strictly speaking the implicit 1D heat conduction assumption is not valid when using the current specimen preparations with ceramic fiberboard. For the purpose of minimizing the edge effect the side of the specimen was wrapped with an insulation material (ceramic blanket). By doing so the center
temperature data were much less affected by the 3D heat transfer behavior than without insulating the side. Overall Center temperatures with the side insulation are still much closer to the ones from the 178 mm by 178 mm specimen than the ones from 100 mm by 100 mm specimen. However, heat gain from Middle to Center needs to be paid attention when the side of a specimen is insulated. The 178 mm by 178 mm ceramic fiberboard specimen provides the best result in terms of approximating 1D heat conduction in the Cone Calorimeter.

According to the comparison of temperature at the center of the specimen for the configurations, the temperature from using a current standard specimen preparation was 20 – 25 K lower than the one from 178 mm by 178 mm specimen when temperatures were measured at 6 mm below from the surface for the incident heat flux, 80 kW/m$^2$, not only at 600 seconds but also in an early times such as 100 and 200 seconds. The temperature difference at the surface would become larger, approximately 30 – 35 K. Therefore, more attention is called when surface temperature from the current Cone test setup is utilized with a 1D application. The difference, 30 – 35 K, can cause the results underestimated.

Ideally, data measured within the central 50 by 50 – mm area of the specimen should be the same as the one at the center if the 1D heat conduction assumption is correct as has been assumed implicitly. However, it turned out that it did not occur with a current standard specimen preparation. In order to minimize an error special attention is needed with a current Cone setup when data are measured at depth not only at the center of the specimen but also near the 50 by 50 – mm area. The data measured at the
center might be considered 1D if a high accuracy of the 1D data is not required. However, the data taken in the 50 by 50 – mm area not only at the surface but also at depth will be affected by the edge condition. The temperature data will be up to 40 K higher than temperature at the center for 80 kW/m$^2$. For low heat fluxes such as 40 kW/m$^2$, the temperature difference between center and 50 by 50 – mm area will be up to 15 K. Temperature measurements kept apart from the center would not be as good as the one at the center due to the edge effect even if temperature measurements were taken in 50 by 50 – mm area.

From a practical point of view the simulated 1D and 3D data were applied to ignition analysis. As an example two sets of data from using a current standard specimen preparation with the frame that has a 3D effect and the proposed 1D specimen preparation were used following Janssens’ procedure$^{15}$. Overall the analysis did not show a significant difference between the two sets of data. It is fair to say that the ignition analysis did not differentiate 3D data from 1D data. The reason is thought that the simulated 3D data were unrefined due to the fact that the corrected thermal conductivity was an effective 1D property used to approximately compensate for 3D behavior (bias) when using a 1D model.

2.7 Future Works

Numerous Cone tests were able to be conducted since ceramic fiberboard was chosen as specimen. One of the conclusions regarding a specimen size may be misinterpreted: *Any specimen with a larger size such as 178 mm by 178 mm (7 inch by 7*
62

inch) would provide the best result in terms of approximating 1D heat conduction in the Cone Calorimeter. It is not the author’s intention. It might be true for some materials, but it has not been proved yet. For approximating the 1D heat conduction in this research, two things happened to be coupled. One is that a larger specimen size was used than the one currently used. The other is that ceramic fiberboard used as specimen is an insulating material.

In order to generalize this conclusion to all materials, there is a way to conduct Cone tests with 178 mm by 178 mm (7 inch by 7 inch) PMMA and then compare the results with the ones from Cone tests with 100 mm by 100 mm (4 inch by 4 inch) PMMA with 39 mm (1.5 inch) width ceramic fiberboard at the side.

It is also interesting to see if Cone results using the retainer frame and specimen holder which were made of an insulation material would have less edge effect than using the frame and holder with stainless steel.

In addition, a modification of Config [III], i.e., using a ceramic fiberboard which has a higher density than PM board at the side of the specimen (PM board), might show less heat gain from the side. For Config [III] ceramic blanket was used to wrap the side which has a much lower density than PM board. The lower density resulted in higher thermal diffusivity, which explains the heat flow from the edge towards the center of the specimen.
3.0 DEVELOPMENT OF A HEAT FLUX MAPPING PROCEDURE UTILIZING AN INFRARED CAMERA

3.1 Introduction

3.1.1 Problem Statement

Accurate fire growth predictions using models such as flame spread algorithms are very important for the evaluation of the performance of wall lining materials in fire. The algorithms require heat flux maps provided by an ignition burner flame. However, few studies have been done and the works to date have limitations in terms of spatial resolution. To increase the power of fire modeling, the accurate and detailed heat flux maps from the ignition burner are critical.

3.1.2 Objective

The objective of this research is to develop a heat flux mapping procedure for a room test burner flame to a wall lining material with the information of surface cooling temperature taken from an infrared camera. One – dimensional (1D) heat conduction model is developed for heat flux mapping from the burner. The resolution of the model can be obtained utilizing 1D temperature data from the Cone Calorimeter tests. It is valuable that the infrared camera can provide high level spatial resolution for surface temperature.
3.1.3 Approach

As a wall lining material ceramic fiberboard is selected due to its thermal stability. A standard room test burner is turned on for a certain duration with a known heat release rate in a flat wall configuration. One temperature measurement in depth is taken at a location using a thermocouple. The location is called “Location A” for convenience. Immediately after the burner is turned off, an infrared camera takes the cooling temperature histories over the surface area including surface temperature at Location A. The measured cooling temperature versus time can be obtained at numerous locations. This database of empirical results is used for comparison to a heat conduction model.

One – dimensional (1D) heat conduction model is developed for heat flux mapping from the ignition burner. The model can provide temperature profiles not only at the surface but also in depth under a given incident heat flux with appropriate thermal properties of the wall lining material. The 1D temperature data from the model are compared with the 1D temperature data measured from the Cone Calorimeter tests. The comparison allows the resolution of the model to be determined.

In order for the 1D model to reproduce the temperature data measured in depth at Location A, the model is repeatedly run for the same exposure time as in the experiment by varying the incident heat flux only. Once the modeled temperature history agrees with the measured one in depth at Location A, the incident heat flux used in the model is determined as a best estimate heat flux from the ignition burner at Location A. Cooling surface temperature history taken from the infrared camera at Location A is then
compared with the one from the model under the determined incident heat flux. Residual between the measured surface temperature and the simulated surface temperature at Location $A$ is calculated. It is the procedure to determine the residual for a best estimate heat flux.

Using the residual at Location $A$ as matching criteria for heat flux, other heat fluxes for other locations are determined by comparing cooling surface temperature histories from the infrared camera with the ones from the model. All the determined heat fluxes, for all locations, together create the heat flux map.

3.2 Literature Review

3.2.1 Requirement of Flame Spread Models

It is very important to understand the performance of wall lining materials in room fire development. ISO 9705 room/corner test $^{17}$ is used for the classification of wall linings. It is a standard full – scale test in which materials are exposed to a burner at $100 \, kW$ for 10 minutes and then $300 \, kW$ for 10 minutes. Due to the costly expense associated with full – scale testing, there is significant interest in simulating full – scale tests based on the results from bench – scale test apparatus such as the Cone Calorimeter. $^{3}$ Several flame spread algorithms to predict the results of the ISO 9705 test $^{17}$ are available with different levels of complexity by Karlsson, $^{18}$ Janssens et al., $^{19}$ Wright, $^{20}$ and Lattimer et al. $^{21}$ Those flame spread algorithms require heat flux information from the ignition burner flame. Heat flux from the burner was assumed
constant by Karlsson\textsuperscript{18} and Janssens \textit{et al.}\textsuperscript{19} Wright\textsuperscript{20} has taken experimental information on the heat flux distribution from Dillon's work.\textsuperscript{22} In the case of Lattimer \textit{et al.}\textsuperscript{21}, heat flux mapping experiments were conducted to develop heat flux correlations for use in the model. To make the algorithms more robust, it is essential to provide accurate and detailed heat flux maps.

3.2.2 Previous Studies on Heat Flux from the Ignition Burner Flame

Few studies have been carried out to determine the incident heat flux from the ignition burner. Janssens\textsuperscript{19,23} determined heat flux values based on the heat output of the burner and the temperature of the material. Back \textit{et al.}\textsuperscript{24} measured incident wall heat flux distributions on a flat wall configuration with the measurements in 0.15 m increments. Correlation of data for the measured heat flux distributions were developed for wall flame spread modeling. Williamson \textit{et al.}\textsuperscript{25} conducted room/corner experiments to evaluate the effects of ignition source intensity and location on the heat flux distribution and focused on the stand – off distance. Heat flux measurements at nine locations in a corner were taken at heat release rates of 40 kW and 150 kW.

Kokkala\textsuperscript{26} performed experiments for total heat flux distributions of the flame of burners for 40, 100, 160, 230 and 300 kW in an open corner of walls. The contour plots for the flux distributions were based on 100 data points. The flux distributions indicate lines of constant heat flux in 10 kW/m\textsuperscript{2} increments. Dillon\textsuperscript{22} provided heat flux distributions along the walls and ceiling at 100 kW and 300 kW from full – scale room/corner tests. The heat fluxes were calculated from thermocouple readings of steel
plates mounted on the walls and ceiling. 24 gauge thermocouples were fixed to the unexposed surface of steel plates. A total of 96 thermocouples were used at intervals of 0.15 m. Dillon’s heat flux distributions were obtained at every 10 kW/m². Lattimer et al. conducted tests using the full-scale mock corner with an overhead for the fire sizes from 25 kW to 300 kW. The heat flux measurements were taken from twenty heat flux gauges placed 0.15 m or 0.3 m apart. Empirical correlations for heat flux were developed from the data obtained in the experimental part of their work for use in the corner fire growth model developed by them.

3.3 Development of a Heat Flux Mapping Procedure with an Infrared Camera Utilizing One-Dimensional Cone Data

Ceramic fiberboard is used as a wall lining material in the experiment because it is thermally stable and can be repeatedly exposed to the high heat fluxes. Repeatability of the ceramic fiberboard used was described in Section 2.3. The material system for the wall configuration consists of five layers of 6.35 mm (1/4 inch) thick PM board and two layers of 12.7 mm (1/2 inch) thick HS board. The density for PM board is \(240 \text{ kg/m}^3\) and that for HS board is \(448 \text{ kg/m}^3\).

A standard room test burner, the ISO 9705 burner, is turned on for a certain duration with a known heat release rate, \(\dot{Q}\), in a flat wall configuration. A schematic of the experimental set up is illustrated in Figure 36. One temperature measurement in depth is taken at a location using a thermocouple. For the purpose of explanation this
location is called “Location A” for convenience. Immediately after the burner is turned off, an infrared camera takes the cooling temperature histories over the surface area including surface temperature at Location A. Correct surface temperature data for the heating period cannot be obtained with an infrared camera due to the presence of the soot in the diffusion flames. However, the surface temperature data when the flame is removed, *i.e.*, in the cooling period, can be utilized to determine the incident heat flux from the burner flame. Considering the fact that an infrared camera has high level spatial resolution, surface temperature measurements can be taken at numerous locations. This database of empirical results is used for comparison to a heat conduction model.
Figure 36 A schematic of the experimental set up with a standard room test burner in a flat wall configuration.

One – dimensional (1D) heat conduction model was developed for heat flux mapping from the ignition burner. One might question that solving the 1D heat conduction equation is oversimplified to predict temperature profiles. However, it is reasonable considering the material used, ceramic fiberboard, is an inert material. It is essential that the heat conduction model uses actual thermal properties of the material such as thermal conductivity, specific heat, and emissivity as opposed to generic values. The use of actual thermal properties is directly related to the accuracy of temperature
prediction from the model. The model is run with appropriate thermal properties and can provide temperature profiles not only in depth but also at the surface. Each run is made with a given incident heat flux and burner exposure duration.

This model requires the thermal response of the wall lining material which is calibrated using Cone Calorimeter. The 1D temperature data taken from the Cone Calorimeter tests are simulated by the model. It is obvious that the simulated temperature data from the 1D model have to be compared with the measured temperature data from the Cone specimen which shows a 1D heat transfer behavior. It is pointless to compare the 1D simulated data with 2D or 3D measured data in the Cone. It was found out that the current Cone specimen preparations, i.e., the 100 mm by 100 mm specimen size regardless of whether the retainer frame is used or not, result in 3D behavior with ceramic fiberboard. Obtaining 1D temperature data from the Cone tests was described in Section 2.4.

In order for the 1D model to reproduce the temperature data measured in depth at Location A, the model is repeatedly run for the same exposure time as in the experiment by varying the incident heat flux only. Once the modeled temperature history agrees with the measured one in depth at Location A, the incident heat flux used in the model is determined as a best estimate heat flux from the ignition burner for Location A. Cooling surface temperature history taken from the infrared camera at Location A is then compared with the one from the model under the determined incident heat flux. Residual between the measured surface temperature and the simulated surface temperature at
Location $A$ is calculated. It is the procedure to determine the residual for a best estimate heat flux.

Using the residual at Location $A$ as matching criteria for heat flux, other heat fluxes for other locations are determined by comparing cooling surface temperature histories from the infrared camera with the ones from the model. All the heat fluxes at all locations create the heat flux map.

A prototype experiment in a wall configuration is performed using the ISO 9705 test burner$^{17}$ to demonstrate the developed heat flux mapping procedure. The results of the experiment allow the heat flux and spatial resolutions of the method to be determined and compared to the methods currently available.

### 3.4 One – Dimensional Heat Conduction Model

#### 3.4.1 Development of the One – Dimensional Heat Conduction Model

The model solves the one – dimensional heat conduction equation for a semi-infinite solid. The governing equation is

$$
\rho \frac{\partial}{\partial t} \left[ c(T)T \right] = \frac{\partial}{\partial y} \left[ k(T) \frac{\partial T}{\partial y} \right]
$$

The initial and boundary conditions are as follows.

$$
t = 0, \quad T = T_0
$$

$$
y = 0, \quad -k \frac{\partial T}{\partial y} = \dot{q}_n = \frac{F_{s-H} E_s \sigma (T_H^4 - T_s^4)}{E_s + F_{s-H} (1 - E_s)} - h_c (T_s - T_0)
$$
\( y \to \infty, \quad T = T_n \) \hspace{1cm} (12)

The density, \( \rho \), of the ceramic fiberboard does not change significantly with temperature. But the thermal conductivity, \( k \), and specific heat, \( c \), of the ceramic fiberboard are a function of temperature. The temperature dependent thermal conductivity, \( k(T) \), and the specific heat, \( c(T) \), were provided by the manufacturer, Thermal Ceramics Inc.

\[
c(T) = -1.08 \times 10^{-19} \times T^3 - 1.09 \times 10^{-4} \times T^2 + 4.52 \times 10^{-1} \times T + 6.73 \times 10^2 \quad (13)
\]

The emissivity of surface, \( \varepsilon_s \), was measured using the infrared camera. The convective heat transfer coefficient, \( h_c \), was calculated using the following correlations\(^{27}\). The correlation for heated horizontal plate facing up was applied to the model for Cone Calorimeter tests and the correlation for vertical planes was for a wall configuration.

\[
Nu = 0.54(Gr \Pr)^{1/4} \quad \text{for heated horizontal plate facing up} \quad (14)
\]

\[
Nu = 0.59(Gr \Pr)^{1/4} \quad \text{for vertical planes} \quad (15)
\]

The details on how the 1D heat conduction model was coded using a finite difference method can be found in the Appendix H.
3.4.2 Resolution of the Developed One – Dimensional Heat Conduction Model

The 1D model simulated temperatures from Cone Calorimeter tests and the results between experiment and calculation were compared. The varying parameter of this model was incident heat flux such that each run of the model gave a different predicted temperature profile with time. This temperature profile was then compared with the profile from Cone Calorimeter test until a match is made. As discussed in Section 2.4, the 178 mm by 178 mm ceramic fiberboard specimen provides the best result in terms of approximating 1D heat conduction in the Cone Calorimeter compared to temperature profiles from a current standard specimen preparation. Thus, the simulated temperature profiles from the 1D model were compared with the Cone data using the 178 mm by 178 mm specimen.

As an example, Cone Calorimeter test for the incident heat flux, 40 kW/m\(^2\), was simulated with a range of heat flux from 36 to 45 kW/m\(^2\) with 1 kW/m\(^2\) increment. In order to find a match between the results of the experiment and simulations, the temperature difference, \(\Delta T = T_{\text{exp}} - T_{\text{calc}}\), was obtained. The temperature differences for the incident heat flux from 36 to 45 kW/m\(^2\) were plotted in Figure 37. It was obvious that the temperature differences for 36, 37, 44 and 45 kW/m\(^2\) showed a significant deviation from zero. However, it was not easy to distinguish which one is best among 39, 40, 41, and 42 kW/m\(^2\). Therefore, the average value of temperature differences was taken for each heat flux as seen in Table 10. Also, using the temperature difference a residual is calculated as follows.
where $N$ is the number of data points. The lowest value for both average temperature
difference and residual was obtained when the incident heat flux, 41 kW/m$^2$, was used in
the model. It indicates that the best agreement between the experiment and simulation
was found with the choice of 41 kW/m$^2$. However, the average and residual values for 40
kW/m$^2$ were very close to those for 41 kW/m$^2$. Thus, it would be reasonable to say that
the resolution of the 1D model is ±1 kW/m$^2$.

Figure 38 shows the comparison of the experimental and calculated temperature
histories in depth when the incident heat flux, 40 kW/m$^2$, was applied. The temperature
profiles from the 1D model agree favorably with the 1D temperature data from the Cone
Calorimeter test.
Figure 37  Comparison of the temperature differences between 1D Cone data for 40 kW/m² and simulations with various heat fluxes from 36 to 45 kW/m².

Table 10  Comparison of temperature difference between 1D Cone data for 40 kW/m² and simulations with various heat fluxes from 36 to 45 kW/m².

<table>
<thead>
<tr>
<th>IHF</th>
<th>36 kW/m²</th>
<th>37 kW/m²</th>
<th>38 kW/m²</th>
<th>39 kW/m²</th>
<th>40 kW/m²</th>
<th>41 kW/m²</th>
<th>42 kW/m²</th>
<th>43 kW/m²</th>
<th>44 kW/m²</th>
<th>45 kW/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Avg. ΔT (K)</td>
<td>17.3</td>
<td>13.5</td>
<td>9.8</td>
<td>6.2</td>
<td>2.7</td>
<td><strong>-0.8</strong></td>
<td>-4.3</td>
<td>-7.6</td>
<td>-10.9</td>
<td>-14.2</td>
</tr>
<tr>
<td>Residual (K)</td>
<td>0.75</td>
<td>0.59</td>
<td>0.44</td>
<td>0.30</td>
<td>0.19</td>
<td><strong>0.17</strong></td>
<td>0.26</td>
<td>0.38</td>
<td>0.51</td>
<td>0.64</td>
</tr>
</tbody>
</table>
3.5 Application of a 1D Heat Conduction Model on Determining Heat Fluxes from a Real Fire Situation

The advantage of using the 1D “direct” approach is that the mathematical complexities and difficulties associated with 3D and 1D “inverse” codes are avoided. To validate the use of a 1D model to determine heat fluxes from a real fire situation, a commercial 3D conduction code, ALGOR’s finite element analysis, has been used for the purpose of comparison.

Figure 38 Comparison of the simulated data from the 1D model with the measured 1D data from the Cone test for 40 kW/m².
Transient heat transfer in ALGOR’s finite element analysis was used with the same properties as in the 1D model. For surface loads, convection load, radiation load, and heat flux load need to be applied. For convection load,

\[ q'' = h_c (T_s - T_a) \]  

(17)

where \( h_c \): convective heat transfer coefficient

\( T_s \): surface temperature

\( T_a \): ambient temperature

For radiation load,

\[ q'' = F_{rad} \sigma (T_s^4 - T_a^4) \]  

(18)

where \( F_{rad} \): radiation factor, which includes absorptivity, emissivity, and viewing effects

\( \sigma \): Stefan-Boltzmann constant

For heat flux load, input data are required in order to run ALGOR. Lattimer’s heat flux distribution\(^{28}\) as seen in Figure 39 was used as input data. The region outlined by a dotted line, 0.5 m × 1.6 m, has a various heat fluxes from 5 kW/m\(^2\) to 50 kW/m\(^2\) and was simulated for 3D analysis.
Temperature distribution at each time step was obtained through the ALGOR run. As an example of results, surface temperature profiles computed at 5 and 600 seconds are shown in Figure 40 and Figure 41, respectively. Red color represents the highest temperature and blue means the lowest temperature as seen in the color bar with temperature. Orange, yellow, and green colors are in between the two. It was expected that the temperature profile would show smooth transition from high to low temperatures, *i.e.*, from red – orange – yellow – green to blue color. However, inconsistent temperatures were observed in Figure 40 and Figure 41. Some of distinguished ones are marked with oval. Yellow color was found between red and orange colors and small
islands with green color existed between orange and yellow colored regions. These inconsistent temperature profiles would not occur in reality.

It was discovered through the communication with technical support in ALGOR that the computational limitation of ALGOR software could cause the inconsistent temperature profiles to show up. In ALGOR the surface load on the face of a brick element follows the majority rule. This means that if two sides out of four are on surface 2 and the other two are on surface 3 then this face has no load because there is no majority. Also, if two sides are on surface 2, one is on surface 3, and one on surface 4, then it will not see a load either according to the majority rule. As an example, partial view of the mesh (brick elements) is described in Figure 42. The brick element named “A” has 30 kW/m² and “C” brick element has 20 kW/m². However, “B” brick element has 0 kW/m² according to the majority rule since two sides are on 30 kW/m² and the other two are on 20 kW/m².
Figure 40  
Result from 3D ALGOR run using Lattimer’s heat flux distribution – 
surface temperature profile at 5 seconds.
Figure 41  Result from 3D ALGOR run using Lattimer’s heat flux distribution – surface temperature profile at 600 seconds.
Due to the unrealistic temperature profiles shown above, 2D analyses in ALGOR were done using a thin strip shape as shown in Figure 43. A vertical strip was taken from Lattimer’s heat flux distribution, which is a center line above the burner. Also, the horizontal strip in Figure 43 was chosen because of its dense constant heat flux lines.

Even though fewer inconsistent temperatures showed up for the 2D analyses than for the 3D, the 2D analyses with the increment of 10 kW/m² did not completely eliminate the inconsistent temperatures. The increment of 10 kW/m² between contours as heat flux input appeared to be too large to obtain realistic temperature profiles.
The temperature data from 2D analyses as a vertical and a horizontal strip were compared with data from 1D analysis. The heat flux distributions at the top surface along the vertical and horizontal strips are described in Figure 44. Figure 45 shows the comparison at the center line above the burner (where the vertical strip is located in Figure 43) at 600 seconds. The red dotted line stands for temperature data from 1D analysis under the heat flux, 45 kW/m². The blue circle is node temperature in 2D analysis when 45 kW/m² was applied from node V1 to node V10. It was observed that temperature at node V1 is higher than other nodes while node V10 temperature is lower than others. It is because node V1 is located at the boundary of the region for 55 kW/m² and node V10 is attached to the region for 35 kW/m². Except these two node temperatures at the boundary, temperatures from node V2 to node V9 are almost same as 1D temperature data.

However, the compact area with a heat flux of 20 – 50 kW/m² (where the horizontal strip is in Figure 43) showed a different behavior as seen in Figure 46. Even
though the same heat flux, 45 kW/m², was applied to node H1 throughout node H7, node temperatures are not in an order and fluctuate significantly. The two temperatures from 1D and 2D analyses do not agree. A similar tendency can also be found in case of the region for 35 kW/m².

Thus, it is acceptable to use the developed 1D heat conduction model in the area of the center line above the burner in the prototype experiment which showed 1D behavior through ALGOR runs.

![Diagram](image-url)

**Figure 44** Vertical strip (Left) and horizontal strip (Right) with heat flux distribution at the top surface.
Figure 45  Comparison of temperature data from 2D analysis with 1D analysis at 600 seconds – at the center line above the burner (vertical strip).

Figure 46  Comparison of temperature data from 2D analysis with 1D analysis at 600 seconds – where dense constant heat flux lines are (horizontal strip).
3.6 Prototype Experiment Utilizing an Infrared Camera

3.6.1 Experimental Setup

To demonstrate how the method described in Section 3.3 works, a series of prototype experiments were conducted using an infrared camera. Infrared camera\(^29\) is designed not only to display an image of the heat patterns radiating from surfaces, but also to produce temperature information. The infrared camera provides high level spatial resolution for surface temperature compared to using thermocouples or heat flux gauges. If the infrared camera is placed 150 cm away from a target, 1 cm spatial resolution can be obtained.

The experiment was performed in a wall configuration under a large hood using the ISO 9705 test burner\(^17\). For simplicity a flat wall configuration was used with five layers of 6.35 mm thick ceramic fiberboard on top of two layers of 12.7 mm thick ceramic fiberboard. First interface between first layer and second layer is called I1. In the same manner I2, I3, I4, and I5 stand for second, third, fourth, and fifth interface, respectively. A schematic of the experimental set up with a standard room test burner in a flat wall configuration is shown in Figure 47. The area in a red dotted circle with Location A as the central point was enlarged in Figure 48 (Left) for the detailed instrumentation and experimental set up. The corresponding section view (A-A) of the area is also provided in Figure 48 (Right). It is noted that the diameter of a thermocouple is 0.3 mm.
One Schmidt-Boelter heat flux gauge was used to compare the heat flux determined through the heat flux mapping procedure. The thermocouple at Location A was installed 10 mm away from the edge of the heat flux gauge. A total of twenty five thermocouples were installed in the vicinity of the thermocouple at Location A to confirm that measured temperature data in the area are almost same as the temperature data at Location A. Temperature measurements at the back – face of the wall, *i.e.*, at I5, were made using thermocouples spaced 0.15 m apart in the area along the center line above the burner to make sure the ambient temperature.

![Figure 47](image-url)  

Figure 47  A schematic of the experimental set up with a standard room test burner in a flat wall configuration – Location A.
The Schmidt-Boelter heat flux gauge used is a water cooled total heat flux transducer with $\frac{1}{4}$ inch (6.35 mm) diameter. The manufacturer, Medtherm, reports the calibration uncertainty of $\pm$ 3% for incident radiant flux. The design heat flux range of the gauge is up to 50 kW/m$^2$. The reason for the choice of the small range is to minimize any possible uncertainty of the heat flux gauge itself. The diameter of the gauge, 6.35...
mm (¼ inch), was chosen based upon the spatial resolution from the infrared camera, 10 mm. It was questioned if the gauge diameter would influence the surrounding ceramic fiberboard due to its colder temperature than the heated ceramic fiberboard. A Schmidt-Boelter heat flux gauge surrounded by ceramic fiberboard was simulated using ALGOR software. The gauge is located at the left top corner of Figure 49 (Section view). The simulated gauge is 12 mm long and 3 mm wide because of the symmetry. The total simulated region is 30 mm by 30 mm based upon the thickness of five layers of 6.35 mm thick ceramic fiberboard, 31.8 mm. A fixed water temperature, 25 °C, was applied to the area of the gauge. Incident heat flux, radiative and convective losses occur at the top surface.

**Figure 49** Schematic for the simulation of a Schmidt-Boelter heat flux gauge surrounded by ceramic fiberboard – fixed water temperature.
The temperature profile at 300 seconds from the ALGOR run is seen in Figure 50. The top surface zone influenced by the Schmidt-Boelter heat flux gauge as a cold “spot” is 3 mm from the edge of the gauge. Applying fixed water temperature in the whole area of the Schmidt-Boelter heat flux gauge might be considered too simplified. Another ALGOR run was done using the properties of copper and fixed water temperature instead of applying fixed water temperature only. As described in Figure 51 a smaller region, 10 mm by 12 mm, was chosen as total simulated region while the size of the gauge was the same. The properties of copper were used in the area of the gauge and fixed water temperature was applied along the line (a-b). Incident heat flux, radiative and convective losses occur at the top surface (a-d). Figure 52 shows the temperature profiles when using copper and fixed water temperature at 30 seconds. This temperature profile is little different from the one using fixed water temperature only in the area of the gauge (Figure 53).

The results from both fixed water temperature only and fixed water temperature and copper clearly show an effect of the cold “spot” due to the presence of a Schmidt-Boelter heat flux gauge on the wall. However, the influenced zone is 3 mm from the edge of the gauge. If temperature measurements were taken 10 mm from the edge of the gauge, the distance of 10 mm is far enough that the cold spot should not effect to the surroundings. This value, 10 mm, is also related to the spatial resolution of the infrared camera, 10 mm. Thus, thermocouples were placed 10 mm from the edge of the gauge to avoid any influence caused by the Schmidt-Boelter heat flux gauge.
Figure 50 Simulation of a Schmidt-Boelter heat flux gauge surrounded by ceramic fiberboard using ALGOR – fixed water temperature.
Figure 51  Schematic of the simulation of a Schmidt-Boelter heat flux gauge surrounded by ceramic fiberboard using ALGOR – properties of copper and fixed water temperature.
Figure 52 Simulation of a Schmidt-Boelter heat flux gauge surrounded by ceramic fiberboard using ALGOR – properties of copper and fixed water temperature.
The Schmidt-Boelter heat flux gauge was calibrated in the Cone Calorimeter. The same test setup as the prototype experiment was used, i.e., the Schmidt-Boelter heat flux gauge was mounted flush with the ceramic fiberboard (Figure 54). The range of the incident heat flux from 5 kW/m² to 50 kW/m² was applied. The heat flux measurements with ceramic fiberboard were compared with the ones without ceramic fiberboard (Figure 55). The results with ceramic fiberboard were 15 % higher than the values from the heat

Figure 53    Simulation of a Schmidt-Boelter heat flux gauge surrounded by ceramic fiberboard using ALGOR – fixed water temperature only.
flux gauge alone, which corresponds to the result reported by Persson and Wetterlund\textsuperscript{30}. This 15 % was accounted for when the heat flux mapping procedure was applied.

Figure 54  Heat flux measurement from Schmidt-Boelter heat flux gauge surrounded by ceramic fiberboard in the Cone Calorimeter.

Figure 55  Heat flux measurement from Schmidt-Boelter heat flux gauge in the Cone Calorimeter.
3.6.2 Prototype Experiment Utilizing an Infrared Camera

The goal of the prototype experiment is to show if higher spatial resolution than currently available can be obtained for heat flux. The current heat flux distribution specifies lines of constant heat flux in 10 kW/m² increments.

For the purpose of the experiment the ISO 9705 burner\textsuperscript{17} was turned on for 5 minutes to heat up the ceramic fiberboard at heat release rate of 50 kW. Figure 56 and Figure 57 show the front and side views of the experiment for the heating period. The infrared camera appears in Figure 56. Readers might wonder why the infrared camera was used for the heating period. It is clarified that the camera was not used to record temperature of the wall in the heating period. The reason for the existence of the camera is that it needed to be set 1.5 m from the wall in advance so that cooling temperatures can be recorded immediately. As seen in Figure 57 a stand-off distance of 10 cm was applied in the experiment.

Temperature measurement at the depth of I1 was taken at Location $A$ using a thermocouple (Figure 56). Immediately after the fire source was removed, \textit{i.e.}, after 5 minutes, the infrared camera started to capture the surface temperature data of the heated wall material including surface temperature at Location $A$. Cooling surface temperature data were measured from the infrared camera at numerous locations for another 5 minutes.
Figure 56    Prototype experiment in a wall configuration with a room test burner – Front view in the heating period.

Figure 57    Prototype experiment in a wall configuration with a room test burner – Side view in the heating period.
Two experiments were conducted under the same test conditions for the purpose of repeatability. The results between the two experiments showed high repeatability as shown in Figure 58. It is noted that ceramic fiberboards were exposed to a big fire such as 700 kW beforehand and these data were not included for the data analysis as mentioned in Section 2.3.

![Graph showing temperature histories](image)

**Figure 58** Comparison of temperature histories in depth from two experiments under the same conditions for repeatability.

One Schmidt-Boelter heat flux gauge was used on the wall for the purpose of comparison of the heat flux measurement with the determined one through the heat flux mapping procedure. The measurement from the heat flux gauge was taken at every one second. The measured data showed a significant fluctuation between mainly 20 and 40 kW/m² (Figure 59). Due to the large variation, the average heat flux, 28 kW/m², was
calculated. Based upon the correlation of the heat flux measurement in the Cone Calorimeter between with and without ceramic fiberboard as described in Section 3.6.1, the heat flux, 28 kW/m², was decreased by 15 %, which is equal to 23.8 kW/m². The uncertainty of ±3% was reported by the manufacturer, which corresponds to a range of 23.1 to 24.5 kW/m² for the heat flux measurement. This range of 23.1 to 24.5 kW/m² from the Schmidt-Boelter heat flux gauge is then compared with the heat flux from the heat flux mapping procedure.

![Figure 59](image)

**Figure 59** The measurement from Schmidt-Boelter heat flux gauge on the wall.

A number of thermocouples in depth were used in this prototype experiment to confirm that measured temperature data in the vicinity of the thermocouple at Location A are little different from the temperature data at Location A. However, only one
temperature measurement in depth is sufficient as a part of the heat flux mapping procedure. Several temperature measurements in depth are not necessary in the procedure.

For Location A, the temperature measured at the depth of 11 was compared with the one from the 1D model by varying the incident heat fluxes (Figure 60). In case that the incident heat flux, 21 kW/m², was applied in the model, the simulated temperature agreed favorably with the measured temperature based on the values of the average temperature difference and residual as shown in Table 11. Since a match was found between measurement and simulation, the incident heat flux used in the model, 21 kW/m², was determined as a best estimate heat flux from the ignition burner for Location A.

Considering the uncertainty of the model, ±1 kW/m², as mentioned in Section 3.4.2, the determined heat flux from the model should be expressed as a range of 20 to 22 kW/m². This range of heat flux matches reasonably well with the measurement from the Schmidt-Boelter heat flux gauge, 23.1 to 24.5 kW/m². It is reminded that the data from the Schmidt-Boelter heat flux gauge fluctuated between 20 and 40 kW/m².
Figure 60  Comparison of the temperature differences between the measured temperature data and the simulated data with various heat fluxes at the depth of I1.

Table 11  Comparison of the temperature difference between the measured temperature data and the simulated data with various heat fluxes from 18 to 24 kW/m².

<table>
<thead>
<tr>
<th>IHF</th>
<th>18 kW/m²</th>
<th>19 kW/m²</th>
<th>20 kW/m²</th>
<th>21 kW/m²</th>
<th>22 kW/m²</th>
<th>23 kW/m²</th>
<th>24 kW/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Avg. ΔT (K)</td>
<td>-13.5</td>
<td>-9.2</td>
<td>-5.2</td>
<td>-1.1</td>
<td>2.7</td>
<td>6.4</td>
<td>9.9</td>
</tr>
<tr>
<td>Residual (K)</td>
<td>2.2</td>
<td>1.6</td>
<td>0.9</td>
<td>0.3</td>
<td>0.4</td>
<td>1.0</td>
<td>1.5</td>
</tr>
</tbody>
</table>
Since the incident heat flux was determined at Location A, surface temperature history can be obtained from the model. Cooling surface temperature data recorded from the infrared camera at Location A was compared with the one from the model (Figure 61). It is observed that the simulated surface temperature drops too quickly after the burner was turned off and then near the end of the cooling period the modeled data match the data from the infrared camera.

![Graph showing comparison of temperature data](image)

Figure 61 Comparison of the measured temperature data with the simulated data for the incident heat flux, 21 kW/m².

It was investigated why the surface temperature from the model decreases so rapidly. The 1D heat conduction model was examined thoroughly to see if the model has an error or if a unknown parameter from the experiment was not accounted for in the
model. However, nothing was found in regard of the model. It made question if the surface temperature data from the infrared camera were accurate as they occurred on the wall in reality. It is noted that the infrared camera was calibrated at the manufacturer before it was used in the experiment. In general the response time of the camera is fast. For example, if there is an opening in an oven where the interior is hot and the exterior is cool, the camera will clearly show the differences. However, the application of the camera in the experiment is different. The camera is seeing flames and then all of sudden flames disappear and the camera records cooling temperature of the wall. The rate of temperature change in the transition from hot flames to a cooling wall is very speedy. Considering that the camera is an old one it was questioned there might be a delay time in the sudden transition.

A simple test with the infrared camera was done for a delay time. Cone Calorimeter was set to a high heat flux such as 75 kW/m². The infrared camera measured the surface temperature of the Cone heater, approximately 650 – 700 K. As soon as the infrared camera was pointed toward an object with an ambient temperature, it took 15 – 20 seconds for the camera to catch the ambient temperature. Therefore, the discrepancy between the measurements from the infrared camera and the results from the model during the cooling period can be explained as a limitation of the infrared camera.

At Location $A$ the measured cooling temperature data and the simulated cooling data for 21 kW/m² were used to calculate a residual using Equation 19. Note that $N$ is a number of data. The number of data is 30 since the cooling data from the infrared camera at every 10 seconds from 310 to 600 seconds were used. The value of 4 K for a residual
at Location A was acquired from Equation 19 and was used as matching criteria for determining heat flux along the center line above the burner. The choice of the residual, 4 K, resulted from the temperature measurement accuracy of the infrared camera, ±2°C or ±2%. It is noted that using the “later” times when the infrared camera has caught up for surface temperature from the model is not the best way to develop the residual for heat flux. The residual from using “later” times was in fact inferior to the residual of 4 K because all of the residuals from 18 to 24 kW/m² were less than 3 K which is under the uncertainty of the infrared camera measurements.

\[ E_{rms} = \sqrt{\frac{\sum (T_{exp} - T_{calc})^2}{N}} \]  

(19)

Utilizing the infrared camera, numerous cooling surface temperature data were collected along the center line with the spatial resolution of every 1cm. An overall infrared image of the heated wall after the burner was turned off is seen in Figure 62. For each location, when the residual was found out to be 4 K the incident heat flux used for the simulation was determined to be the very heat flux at the location. By repeating this procedure along the center line, the heat flux mapping at the center line was obtained (Figure 63). Compared to the currently available methods for heat flux maps, i.e., with 10 kW/m² increments, the developed heat flux mapping method can provide more detailed heat flux maps.
Figure 62  An overall infrared image of the heated wall after the burner was turned off.
<table>
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<tr>
<th>Distance from burner (m)</th>
<th>Determined heat flux (kW/m²)</th>
<th>Distance from burner (m)</th>
<th>Determined heat flux (kW/m²)</th>
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Figure 63 Heat flux map along the center line above the burner for 50 kW fire.
3.7 Conclusions

It is very important to understand the role of wall lining materials when they are exposed to a fire from an ignition source. Full-scale test methods permit an assessment of the performance of a wall lining material. However, fire growth predictions using models have been developed due to the costly expense associated with full-scale testing. The fire modeling requires the heat flux maps provided by the ignition burner flame as input data. Works to date were impeded by a lack of detailed characterization of the heat flux maps due to the use of limited instrumentation. To increase the power of fire modeling, accurate and detailed heat flux maps from the ignition burner are essential.

The research was to develop a heat flux mapping procedure for a room test burner flame to a wall configuration utilizing surface temperature measurements from an infrared camera. High level spatial resolution for surface temperature was provided by an infrared camera, which is in contrast to "spot" resolution obtained from thermocouples or heat flux gauges. If the infrared camera is placed 150 cm away from a target, 1 cm spatial resolution can be obtained.

As wall lining material, ceramic fiberboard was heated by a room test ignition burner for a certain duration. One temperature measurement in depth was taken at a location using a thermocouple. The location is called “Location A” for convenience. Immediately after the fire source was removed, cooling temperature data over the surface area including Location A were collected utilizing the infrared camera.
As a part of heat flux mapping one–dimensional heat conduction model was developed. The model can provide temperature profiles not only at the surface but also in depth under a given incident heat flux with appropriate thermal properties of the wall lining material. The 1D temperature data from the model are compared with the 1D temperature data measured from the Cone Calorimeter tests. The comparison allows the resolution of the model to be determined which is ±1 kW/m².

The temperature measurement in depth at Location A was compared with simulated temperatures from the model. Each run of the model provided a different predicted temperature profile for the same exposure duration as in the experiment. The only difference of each run was the incident heat flux. When the measured temperature data at Location A corresponded to the simulated one, the incident heat flux used in the model has been determined as a best estimate heat flux from the ignition burner at Location A. Then, cooling surface temperature data from the infrared camera at Location A were compared with the modeled temperature under the determined incident heat flux. A residual between the surface temperature data from the infrared camera and the simulated temperature data was calculated. It is the procedure to determine the residual for a best estimate heat flux.

Using the residual at Location A as matching criteria for heat flux, other heat fluxes for other locations were determined by comparing cooling surface temperature histories from the infrared camera with those from the model. Through repeating this heat flux mapping procedure a heat flux map was created by combining the heat flux at all individual locations.
Development of the 1D heat conduction model was a must to simulate temperature profiles under a given incident heat flux. The advantage of using the 1D “direct” approach is that the mathematical complexities and difficulties associated with 3D and 1D “inverse” codes are avoided. To validate the use of a 1D model to determine best estimate heat fluxes from a real fire situation, a commercial 3D conduction code, ALGOR’s finite element analysis, was used for the purpose of comparison. It is acceptable to use the developed 1D heat conduction model in the area of the center line above the burner which showed 1D behavior through ALGOR runs.

A prototype experiment in a wall configuration was performed using the ISO 9705 test burner to demonstrate the developed heat flux mapping procedure. Utilizing the infrared camera, numerous cooling surface temperature data were collected along the center line above the burner with the spatial resolution of every 1cm. Through the heat flux mapping procedure, the heat flux maps along the center line was obtained at every 1cm. Compared to the currently available methods for heat flux maps, i.e., with lines of constant heat flux in 10 kW/m² increments, the developed heat flux mapping method can provide more detailed heat flux maps.

One Schmidt-Boelter heat flux gauge was used on the wall for the purpose of comparison of the heat flux measurement with the determined one through the heat flux mapping procedure. It was questioned if the gauge diameter would influence the surrounding ceramic fiberboard due to its colder temperature than the heated ceramic fiberboard. A Schmidt-Boelter heat flux gauge surrounded by ceramic fiberboard was simulated using ALGOR software. The results from ALGOR clearly showed an effect of
the cold “spot” due to the presence of a Schmidt-Boelter heat flux gauge on the wall. However, the influenced zone was 3 mm from the edge of the gauge. If temperature measurements were taken 10 mm from the edge of the gauge, the distance of 10 mm is far enough that the cold spot should not effect to the surroundings. This value, 10 mm, is related to the spatial resolution of the infrared camera, 10 mm.

In the prototype experiment the incident heat flux used in the model, 21 kW/m², was determined as a best estimate heat flux from the ignition burner for Location A. Considering the uncertainty of the model, ±1 kW/m², the determined heat flux from the model for Location A was expressed as a range of 20 to 22 kW/m². This range of heat flux matches reasonably well with the measurement from the Schmidt-Boelter heat flux gauge, 23.1 to 24.5 kW/m². It is noted that the measured data from the Schmidt-Boelter heat flux gauge showed a significant fluctuation between mainly 20 and 40 kW/m². Due to the large variation, the average heat flux, 28 kW/m², was chosen for the comparison. Based upon the correlation of the heat flux measurement in the Cone Calorimeter between with and without ceramic fiberboard, the heat flux, 28 kW/m², was decreased by 15 %, which is equal to 23.8 kW/m². By applying the uncertainty of the Schmidt-Boelter heat flux gauge itself, ±3%, a range of 23.1 to 24.5 kW/m² was obtained for the heat flux measurement.
3.8 Future Works

In the prototype experiment the surface temperature from the model decreased more rapidly than temperature measurements from the infrared camera. The discrepancy during the cooling period was explained as a limitation of the infrared camera used in the experiment. A newest infrared camera might be able to solve the problem.

The design heat flux range of the Schmidt-Boelter heat flux gauge used in the prototype experiment was up to 50 kW/m². The reason for the choice of the small range was to minimize any possible uncertainty of the heat flux gauge itself. The heat flux mapping procedure was demonstrated up to 50 kW/m² accordingly. Application of the heat flux mapping procedure for higher heat fluxes than 50 kW/m² need to be demonstrated.

The 1D heat conduction model needs emissivity of a material used in the experiment to predict the surface temperatures. In general, data on emissivity are very limited. In addition, the data currently available are constant, not temperature – dependent. In order to obtain surface temperature with a high accuracy, it is recommended that emissivity as a function of temperature is used in the model.

The stand – off distance, 10 cm, was used in the experiment. The effect of a stand – off distance needs to be investigated.

The heat flux mapping procedure can be applied for common configurations such as a room/corner. In the case, the 1D heat conduction model has to be modified in order to account for re-radiation between walls in a corner.
APPENDICES
APPENDIX A  SURFACE UNIFORMITY OF HEAT FLUX FROM THE CONE HEATER

Cone calorimeter tests were conducted for obtaining temperature profiles over the specimen area. 28 – gauge thermocouples were welded and calibrated with boiling water. Even though the thermocouple readings were within the limits of error, 1.1 °C or 0.4 %, thermocouples can be grouped according to their behavior based upon the range of minimum and maximum temperatures, and average temperature. Twenty five thermocouples were selected, which provided consistent and stable temperature readings. Ten layers of 6.35 mm thick (¼ inch) ceramic fiberboard (PM board) were piled up on top of two layers of 12.7 mm thick (½ inch) HS board. To minimize the air gap between layers, each thermocouple was fitted into a groove on the top surface of the second layer of the PM board, i.e., the first interface (I1). The Cone test setup is shown in Figure 64. For the purpose of analysis, the thermocouples were classified with 6 groups according to its location. A pink dot ( ● ) stands for Center, red dots ( ● ) are for Middle 1, yellow dots ( ● ) are for Middle 2, blue dots ( ● ) are for Edge 1, green dots ( ● ) are for Edge 2, and turquoise dots ( ● ) are for Corner. The distance from the edge of the specimen to a thermocouple bead for Edge 1, Edge 2, and Corner, 7 mm, was chosen based upon the fact that the length before a thermocouple wire is bent should be greater than 20 times of the diameter of thermocouple (D = 0.3 mm).
Figure 64  Cone test setup for obtaining the temperature profile over the specimen area – Plan view (top) and Section view A – A (bottom). Units are in [mm].

The incident heat flux, 10 kW/m², was applied with using the retainer frame. Temperature readings in the same group were compared and a representative temperature profile at each group is shown in Figure 65. There is a big jump from Middle 2 to Edge 1, which explains that Middle 2 is the boundary of the central 50 by 50 – mm area of the specimen. The temperature difference between Center and Corner is significant. The difference is also clearly seen in the section views from the center to the edge over time in Figure 66. It is noted that the temperature values for the section views were calculated
based upon 30 second average. One section for Center – Middle 1 – Edge 1 is compared
with the other section for Center – Middle 2 – Corner at a time as seen in Figure 67
through Figure 70. Considering the fact that the temperature difference between Center
and Corner is significant it needed to be confirmed if the incident heat flux from the cone
calorimeter heater is uniform.
Figure 65  Temperature profiles at various locations of the specimen area.

Figure 66  Section view [Center – Middle 1 – Edge 1] over time.
Figure 67  Comparison of the side and corner directions at 200 seconds.

Figure 68  Comparison of the side and corner directions at 600 seconds.
Figure 69  
Comparison of the side and corner directions at 1000 seconds.

Figure 70  
Comparison of the side and corner directions at 1800 seconds.
First of all, it needed to be confirmed if the incident heat flux from the Cone Calorimeter heater is uniform. It was found that the center of the heater was not in line with the one of the specimen. Thus, the specimen has been re-centered under the Cone heater by moving the Cone platform. The reading from the heat flux meter in the Cone Calorimeter was confirmed. The heat flux meter was checked at 14 flux levels from 10 to 100 kW/m² and showed linearity in its flux versus voltage relationship (R²=0.999).

Based upon ASTM E1354³, the irradiance should be uniform within the central 50 by 50 – mm area of the specimen to within ±2 % in the horizontal orientation and to within ±10 % in the vertical orientation. Data were acquired every 0.1 second and the value for 2 second average was used. The standard deviation was calculated from the 20 points at each location. Heat flux maps and the standard deviations are shown in Figure 71 through Figure 76. The range of ±2 % was calculated from the heat flux at the center. For 80 kW/m², the hatched area with a dotted line, i.e., the central 50 by 50 – mm area, is within ±2 %. For 10 and 40 kW/m², the values with asterisk (*) are not within the range. However, it is fair to say that the central 50 by 50 – mm area is within ±2 % considering the standard deviation.
Figure 71  Heat flux map for 10 kW/m².

Figure 72  Standard deviation for 10 kW/m².
Figure 73  Heat flux map for 40 kW/m$^2$.

Figure 74  Standard deviation for 40 kW/m$^2$.  

±2 %  
(38.22–39.78)
Figure 75  Heat flux map for 80 kW/m².

Figure 76  Standard deviation for 80 kW/m².
APPENDIX B  CONTACT RESISTANCE BETWEEN LAYERS OF CERAMIC FIBERBOARD

It is ideal to obtain temperatures at depth with a single piece of ceramic fiberboard as described in Figure 77 (Left). However, ten layers of ceramic fiberboard, Figure 77 (Right), were used instead due to the difficulty of a thermocouple installation in a single piece.

![Figure 77 Description of the temperature measurement at depth.](image)

A thermocouple consists of its bead, bare wire, and insulated wire. The insulated wire was fitted into a groove so that it does not cause an air gap between the upper and lower layers of an interface. For the bare wire, it was placed on the top surface of the lower layer and was compressed down by the weight of the retainer frame when the frame was used. Without the edge frame, the bare wire on the top surface caused a significant air gap. Thus, the bare wire was embedded in the ceramic fiberboard and the bead was the only thing that was flushed with the surface. It needed to be confirmed if the temperature when embedding the bare wire of a thermocouple, Figure 78 [Left], is the same as the one when placing the bare wire on top surface of ceramic fiberboard, Figure 78 [Right]. Cone tests with specimen size, 178 mm by 178 mm (7 inch by 7 inch), were
conducted under the same conditions except for the installation of the bare wire. The detail for how the 178 mm by 178 mm specimen size was determined is described in 2.4.

Figure 78 Bare wire embedded [Left] and bare wire placed on top surface [Right].

In order to measure temperature of ceramic fiberboard at a location, it was first done that two thermocouples were placed at the center of the specimen; one was put at the rear face of the upper layer (U) while the other was at the top surface of the lower layer (L) at each interface as seen in Figure 79. I1 stands for Interface 1 between the first and second layers of ceramic fiberboards. The bead at the rear surface was placed 1 mm apart from the center to avoid touching with the one at the center on the top surface. Temperature differences from two thermocouples at the center, i.e., \( T_U - T_L \), for 80 kW/m² are shown in Figure 80 and Figure 81. The temperature difference for I5 can be considered small. However, the temperature differences from I1 to I3 are significant, especially in the beginning. The peak for I1 in Figure 80 occurred right after the shutter was opened. After the peak, the temperature difference decreases with time until when
the specimen was removed from the heater, 1800 seconds. Decrease of the temperature difference is thought due to the change of the main heat transfer mode with time. Convection and/or conduction are assumed to be dominant in the early stage, but radiation would be the major heat transfer mode afterwards.

![Diagram]

Figure 79  Location of two thermocouples at the center at each interface – Section view.

When the bare wire was embedded, in the case of Embedded, the maximum temperature difference, \( T_U - T_L \), is 120 K at I1, 40 K at I2, and the difference becomes smaller with depth. However, the temperature difference is approximately 5 K at I1 in the case of placing the bare wire on top surface of ceramic fiberboard, Surface case. These behaviors are shown in Figure 80 and Figure 81. What caused the difference between two cases, Embedded and Surface, needed to be investigated. It is recommended to bring the thermocouple wires away along an isotherm for at least 20 wire diameters to reduce conduction errors\(^31\). The diameter for 28-gauge thermocouples is 0.3 mm. Thus, the wire should not be bent up to 6 mm from the bead. In the case of Embedded, it is thought that there is heat flow from the hotter wire to the bead in the upper layer of an interface and from the bead to the cooler wire in the lower layer, which possibly caused a significant temperature difference between two beads. The heat flow direction is described with a red arrow in Figure 82. The maximum air gap between layers is less
than 0.5 mm considering the diameter of the bead and the fact that the ceramic fiberboard (PM board) deforms around the bead when the board layers are compressed with the frame. For the purpose of comparison, case of *Surface* is illustrated in Figure 83.
Figure 80  Temperature difference for 80 kW/m$^2$ at each interface – *Embedded*.

Figure 81  Temperature difference for 80 kW/m$^2$ at each interface – *Surface*.
Temperatures using the embedded bare wire, in the case of *Embedded*, are compared with those when the bare wire is placed on top surface, *Surface* case. As shown in Figure 84, the temperature for the upper layer of the first interface, $T_{I1, U}$, for *Embedded* is the highest while the other three temperatures fall into a similar range. It is interesting that the temperature for the lower layer, $T_{I1, L}$, for *Embedded* is greater than the ones for *Surface*. It had been expected that $T_{I1, L}$, for *Embedded* would be lower than $T_{I1, L}$, for *Surface* due to the heat flow from the bead to the cooler wire as mentioned above. The reason for this unexpected behavior is thought to be that the radiation from the hotter bead of the upper layer to the cooler one of the lower layer is greater than the heat loss through the bare wire in the lower layer. For the case of *Surface*, it is thought that the contact resistance would be minimal since there is no heat loss through the wire, the board deforms around the bead, and the two bead temperatures are similar as seen in...
Figure 84. As a conclusion, the way of installation for a thermocouple in case of *Surface* is chosen for the Cone tests.

How to minimize a possible air gap between layers needed to be explored since the case of *Surface* was chosen for the cone test setup. First, using a ceramic glue, Durapot 809 by Cotronics Corp., between layers was attempted. The temperatures were compared with the ones when the boards were stacked. Temperature with glue was lower than without glue (stacked) and the temperature difference grows larger with depth. This behavior is possibly due to a low thermal conductivity (increased thermal resistance) of the ceramic glue. Thus, it was turned out that using glue is not appropriate since it creates another layer even though the thickness is thin.
Second, four brads were used in the corner of the specimen as a way of compression by pinning. A brad is 0.9 mm in diameter and 12.7 mm (1/2 inch) in length. How three layers of ceramic fiberboard were compressed by using a brad is illustrated in Figure 85. It was recessed under the surface in order to reduce the possibility of its being a local heat source. Location of four brads for each layer is shown in Figure 86. The PM board is rigid enough as to not be bent at the center due to being compressed in the corner. Very tight contact between layers was observed compared to the stacked case without brads. Cone test results with and without brads for 10 and 80 kW/m² were almost the same. Stacking ceramic fiberboards on the wall would not work in a wall configuration. On the contrary, using brads would work well as a test setup in a wall. Thus, brads are used for the cone test setup to minimize the air gap.
Figure 85  How three layers of the ceramic fiberboard are compressed by a brad.

Figure 86  Layout for five thermocouples and four brads in the corner. Even layers (Left) and odd layers (Right).
APPENDIX C  SECTION VIEWS FROM CENTER TO CORNER AT EACH INTERFACE
Figure 87  Section views in Config [I] at 200 sec – 80 kW/m².

Figure 88  Section views in Config [I] at 400 sec – 80 kW/m².
Figure 89  Section views in Config [I] at 600 sec – 80 kW/m$^2$.

Figure 90  Section views in Config [I] at 800 sec – 80 kW/m$^2$. 
Figure 91  Section views in Config [I] at 1000 sec – 80 kW/m².

Figure 92  Section views in Config [I] at 1800 sec – 80 kW/m².
Figure 93  Section views in Config [II] at 200 sec – 80 kW/m².

Figure 94  Section views in Config [II] at 400 sec – 80 kW/m².
Figure 95  Section views in Config [II] at 600 sec – 80 kW/m².

Figure 96  Section views in Config [II] at 800 sec – 80 kW/m².
Figure 97  Section views in Config [II] at 1000 sec – 80 kW/m².

Figure 98  Section views in Config [II] at 1800 sec – 80 kW/m².
Figure 99  Section views in Config [III] at 200 sec – 80 kW/m².

Figure 100  Section views in Config [III] at 400 sec – 80 kW/m².
Figure 101  Section views in Config [III] at 600 sec – 80 kW/m².

Figure 102  Section views in Config [III] at 800 sec – 80 kW/m².
Figure 103  Section views in Config [III] at 1000 sec – 80 kW/m².

Figure 104  Section views in Config [III] at 1800 sec – 80 kW/m².
Figure 105 Section views in Config [IV] at 200 sec – 80 kW/m².

Figure 106 Section views in Config [IV] at 400 sec – 80 kW/m².
Figure 107  Section views in Config [IV] at 600 sec – 80 kW/m².

Figure 108  Section views in Config [IV] at 800 sec – 80 kW/m².
Figure 109  Section views in Config [IV] at 1000 sec – 80 kW/m².

Figure 110  Section views in Config [IV] at 1800 sec – 80 kW/m².
APPENDIX D  THREE – DIMENSIONAL THERMAL MAPS OVER TIME
Figure 111  3D thermal map in Config [I] at 200 seconds – 80 kW/m².
Table I

<table>
<thead>
<tr>
<th>Location</th>
<th>Temperature (K)</th>
</tr>
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<tbody>
<tr>
<td>I1</td>
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<td>I2</td>
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<td>I3</td>
<td>526 K</td>
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<tr>
<td>I8</td>
<td>301 K</td>
</tr>
<tr>
<td>I9</td>
<td>298 K</td>
</tr>
</tbody>
</table>

Figure 112 3D thermal map in Config [I] at 600 seconds – 80 kW/m².
Figure 113  3D thermal map in Config [I] at 1000 seconds – 80 kW/m².
Figure 114  3D thermal map in Config [I] at 1400 seconds – 80 kW/m².
Figure 115  3D thermal map in Config [I] at 1800 seconds – 80 kW/m².
Figure 116  3D thermal map in Config [II] at 200 seconds – 80 kW/m².
Figure 117  3D thermal map in Config [II] at 600 seconds – 80 kW/m².
Figure 118 3D thermal map in Config [II] at 1000 seconds – 80 kW/m².
Figure 119  3D thermal map in Config [II] at 1400 seconds – 80 kW/m^2.
Figure 120 3D thermal map in Config [II] at 1800 seconds – 80 kW/m².
Figure 121 3D thermal map in Config [III] at 200 seconds – 80 kW/m².
Figure 122  3D thermal map in Config [III] at 600 seconds – 80 kW/m².
Figure 123  3D thermal map in Config [III] at 1000 seconds – 80 kW/m².
Figure 124 3D thermal map in Config [III] at 1400 seconds – 80 kW/m².
Figure 125  3D thermal map in Config [III] at 1800 seconds – 80 kW/m².

unit [kW/m²x10⁻¹]
Figure 126  3D thermal map in Config [IV] at 200 seconds – 80 kW/m².
Figure 127 3D thermal map in Config [IV] at 600 seconds – 80 kW/m².
Figure 128  3D thermal map in Config [IV] at 1000 seconds – 80 kW/m².
Figure 129 3D thermal map in Config [IV] at 1400 seconds – 80 kW/m².
Figure 130 3D thermal map in Config [IV] at 1800 seconds – 80 kW/m².
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Figure 132  Temperature profiles in Config [II] at I1 – 100 kW/m².
Figure 133  Temperature profiles in Config [III] at I1 – 100 kW/m².

Figure 134  Temperature profiles in Config [IV] at I1 – 100 kW/m².
Figure 135  Temperature profiles in Config [I] at \( I_2 = 100 \text{ kW/m}^2 \).

Figure 136  Temperature profiles in Config [II] at \( I_2 = 100 \text{ kW/m}^2 \).
Figure 137  Temperature profiles in Config [III] at I2 – 100 kW/m².

Figure 138  Temperature profiles in Config [IV] at I2 – 100 kW/m².
Figure 139   Temperature profiles in Config [I] at I3 – 100 kW/m².

Figure 140   Temperature profiles in Config [II] at I3 – 100 kW/m².
Figure 141    Temperature profiles in Config [III] at I3 – 100 kW/m$^2$.

Figure 142    Temperature profiles in Config [IV] at I3 – 100 kW/m$^2$.  

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Figure 143  Temperature profiles in Config [I] at I4 – 100 kW/m².

Figure 144  Temperature profiles in Config [II] at I4 – 100 kW/m².
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Figure 146    Temperature profiles in Config [IV] at I4 – 100 kW/m².
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Figure 148  Temperature profiles in Config [II] at 15 – 100 kW/m².
Figure 149 Temperature profiles in Config [III] at I5 – 100 kW/m².

Figure 150 Temperature profiles in Config [IV] at I5 – 100 kW/m².
Figure 151  Temperature profiles in Config [I] at 16 – 100 kW/m².

Figure 152  Temperature profiles in Config [II] at 16 – 100 kW/m².
Figure 153  Temperature profiles in Config [III] at I6 – 100 kW/m².

Figure 154  Temperature profiles in Config [IV] at I6 – 100 kW/m².
Figure 155    Temperature profiles in Config [I] at 17 – 100 kW/m^2.

Figure 156    Temperature profiles in Config [II] at 17 – 100 kW/m^2.
Figure 157    Temperature profiles in Config [III] at $17 \cdot 100 \text{ kW/m}^2$.

Figure 158    Temperature profiles in Config [IV] at $17 \cdot 100 \text{ kW/m}^2$.  

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Figure 159 Temperature profiles in Config [I] at $18 - 100 \text{ kW/m}^2$.

Figure 160 Temperature profiles in Config [II] at $18 - 100 \text{ kW/m}^2$. 
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Figure 164   Temperature profiles in Config [II] at 19 – 100 kW/m².
Figure 165  Temperature profiles in Config [III] at 19 – 100 kW/m².

Figure 166  Temperature profiles in Config [IV] at 19 – 100 kW/m².
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Figure 170  Temperature difference, $T_{Mid} - T_{Cir}$, in Config [IV] – 80 kW/m².
Figure 171  Temperature difference, $T_{\text{car}} - T_{\text{cr}}$, in Config [I] – 80 kW/m$^2$.

Figure 172  Temperature difference, $T_{\text{car}} - T_{\text{cr}}$, in Config [II] – 80 kW/m$^2$. 
Figure 173  Temperature difference, $T_{cnr} - T_{cpr}$, in Config [III] – 80 kW/m$^2$.

Figure 174  Temperature difference, $T_{cnr} - T_{cpr}$, in Config [IV] – 80 kW/m$^2$.  

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APPENDIX G  MEASUREMENTS OF SURFACE TEMPERATURE

Accurate measurements of the surface temperature are quite difficult to make. The surface temperature measurement of the ceramic fiberboard was undertaken by embedding thermocouples since the ceramic fiberboard (PM board) deforms around the bead and the bare wire. It is noted that the bead and the bare wire are located in a horizontal plane approximately 0.3 mm below the plane of the exposed surface. Compared to the temperature readings in depth, the surface temperature profiles are slightly noisier, but the variation, approximately 10 K, is not significant. The surface temperature profiles for Configs [I] and [IV] are shown in Figure 175 and Figure 176, respectively.
Figure 175  Temperature profiles at depth in Config [I] – 100 kW/m².

Figure 176  Temperature profiles at depth in Config [IV] – 100 kW/m².
Radiative heat flux between the Cone heater and a specimen surface is needed for a front–face boundary condition. Figure 177 illustrates a network of radiative exchange between the Cone heater and a specimen surface.

Figure 177  Network of radiative exchange between the Cone heater and a specimen surface.
\[
\dot{q}_{rad} = \frac{\sigma T_H^4 - \sigma T_s^4}{1 - \varepsilon_s} \frac{1}{A_s F_{s-H}} + \frac{1 - \varepsilon_s}{A_s \varepsilon_s}
\]

\[
\dot{q}_{rad} = \frac{\sigma (T_H^4 - T_s^4)}{\varepsilon_s + F_{s-H} (1 - \varepsilon_s)} \frac{1}{A_s F_{s-H} \varepsilon_s}
\]

\[
\dot{q}_{rad} = A_s F_{s-H} \varepsilon_s \sigma (T_H^4 - T_s^4)
\]

\[
\dot{q}_{rad} = \frac{A_s F_{s-H} \varepsilon_s \sigma (T_H^4 - T_s^4)}{\varepsilon_s + F_{s-H} (1 - \varepsilon_s)}
\]

\[
\dot{q}_{rad}^* = \frac{F_{s-H} \varepsilon_s \sigma (T_H^4 - T_s^4)}{\varepsilon_s + F_{s-H} (1 - \varepsilon_s)}
\]
One-dimensional heat conduction model has been coded using a finite difference method.

For interior nodes,

\[ -k(T) \frac{T^n_j - T^{n+1}_j}{\Delta y} - k(T) \frac{T^n_j - T^{n}_j-1}{\Delta y} = \rho c(T)(\Delta y) \frac{T^{n+1}_j - T^n_j}{\Delta t} \]

\[ T^{n+1}_j - T^n_j = \frac{\alpha(T)\Delta t}{(\Delta y)^2}(T^n_{j+1} + T^n_{j-1} - 2T^n_j) \]

\[ T^{n+1}_j = T^n_j + Fo(T)(T^n_{j+1} + T^n_{j-1} - 2T^n_j) \]

\[ T^{n+1}_j = (1 - 2Fo(T))T^n_j + Fo(T)(T^n_{j+1} + T^n_{j-1}) \]
where \( Fo(T) = \frac{\alpha(T)\Delta t}{(\Delta y)^2} \), \( \alpha(T) = \frac{k(T)}{\rho c(T)} \)

For the surface with incident heat flux, convective loss, and radiative loss,

\[
-k(T)\frac{T_j^n - T_{j+1}^n}{\Delta y} - h_c(T_j^n - T_s^n) + \frac{F_{s-H} \varepsilon \sigma (T_H^n)^n - (T_j^n)^n}{\varepsilon_s + F_{s-H}(1 - \varepsilon_s)} = \rho c(T)\left(\frac{\Delta y}{2}\right)\frac{T_j^{n+1} - T_j^n}{\Delta t}
\]

\[
T_j^{n+1} = T_j^n + 2Fo(T)(T_{j+1}^n - T_j^n) - Fo(T)(2\Delta y)\frac{h_c}{k(T)}(T_j^n - T_s^n) + Fo(T)\left(\frac{2\Delta y}{k(T)}\right)\frac{F_{s-H} \varepsilon \sigma (T_H^n)^n - (T_j^n)^n}{\varepsilon_s + F_{s-H}(1 - \varepsilon_s)}
\]

\[
T_j^{n+1} = (1 - 2Fo(T) - 2Fo(T)Bi(T))T_j^n + 2Fo(T)(T_{j+1}^n + Bi(T)T_s^n) + Fo(T)\left(\frac{2\Delta y}{k(T)}\right)\frac{F_{s-H} \varepsilon \sigma (T_H^n)^n - (T_j^n)^n}{\varepsilon_s + F_{s-H}(1 - \varepsilon_s)}
\]

where \( Bi(T) = \frac{h_c\Delta y}{k(T)} \)

For the adiabatic back-face,

\[
0 - k(T)\frac{T_j^n - T_{j-1}^n}{\Delta y} = \rho c(T)\left(\frac{\Delta y}{2}\right)\frac{T_j^{n+1} - T_j^n}{\Delta t}
\]

\[
T_j^{n+1} = T_j^n + Fo(T)(2T_{j-1}^n - 2T_j^n)
\]

\[
T_j^{n+1} = (1 - 2Fo(T))T_j^n + 2Fo(T)(T_{j-1}^n)
\]
Contact resistance between layers of ceramic fiberboard was taken into account by including radiation and conduction through an air gap. Figure 178 shows an example when an air gap is at Node 9 and 10. For the radiative heat flux, a network of radiative exchange between Node 9 and 10 is described in Figure 179.

Figure 178 Schematic of nodes for contact resistance between layers of ceramic fiberboard.
where $\varepsilon_9 = \varepsilon_{10} = \varepsilon_{CFB}$

$$\dot{q}_{rad}^\prime = \frac{\sigma T_9^4 - \sigma T_{10}^4}{\frac{1 - \varepsilon_9}{A_9} + \frac{1}{A_9 F_{9-10}} + \frac{1 - \varepsilon_{10}}{A_{10} F_{10-9} A_{10} \varepsilon_{10}}}$$

$$\dot{q}_{rad}^\prime = \frac{\varepsilon_{CFB} \left( T_9^4 - T_{10}^4 \right)}{2 - \varepsilon_{CFB}}$$

$$\dot{q}_{rad}^\prime = \frac{\varepsilon_{CFB} \sigma \left( T_9^4 - T_{10}^4 \right)}{2 - \varepsilon_{CFB}}$$
For Node 9,

\[-k(T)_{\text{Air}} \frac{T^n_9 - T^{n+1}_9}{\Delta y} - k(T)_{\text{CFB}} \frac{T^n_9 - T^n_8}{\Delta y} - \varepsilon_{\text{CFB}} \sigma \left( \frac{(T^n_9)^4 - (T^n_{10})^4}{2 - \varepsilon_{\text{CFB}}} \right) \Delta y \left( \frac{\Delta y}{\Delta t} \right) \]

\[T^{n+1}_9 - T^n_9 = \frac{1}{\rho_{\text{CFB}} c(T)_{\text{CFB}}} \frac{2\Delta t}{\Delta y} \left[ -k(T)_{\text{Air}} \frac{T^n_9 - T^{n+1}_9}{\Delta y} - k(T)_{\text{CFB}} \frac{T^n_9 - T^n_8}{\Delta y} - \varepsilon_{\text{CFB}} \sigma \left( \frac{(T^n_9)^4 - (T^n_{10})^4}{2 - \varepsilon_{\text{CFB}}} \right) \right] \]

\[T^{n+1}_9 = \frac{1}{\rho_{\text{CFB}} c(T)_{\text{CFB}}} \frac{2\Delta t}{\Delta y} \left[ -k(T)_{\text{Air}} \frac{T^n_9 - T^{n+1}_9}{\Delta y} - k(T)_{\text{CFB}} \frac{T^n_9 - T^n_8}{\Delta y} - \varepsilon_{\text{CFB}} \sigma \left( \frac{(T^n_9)^4 - (T^n_{10})^4}{2 - \varepsilon_{\text{CFB}}} \right) \right] \]

\[T^{n+1}_9 = \frac{k(T)_{\text{CFB}}}{\rho_{\text{CFB}} c(T)_{\text{CFB}}} \Delta t \left[ T^n_9 - T^{n+1}_9 \right] - \frac{k(T)_{\text{CFB}}}{\rho_{\text{CFB}} c(T)_{\text{CFB}}} \Delta t \left[ T^n_9 - T^n_8 \right] - \frac{k(T)_{\text{CFB}}}{\rho_{\text{CFB}} c(T)_{\text{CFB}}} \Delta t \left[ T^n_9 - T^n_{10} \right] \]

\[T^{n+1}_9 = F_0(T) \frac{k(T)_{\text{Air}}}{k(T)_{\text{CFB}}} \left[ T^n_9 - T^{n+1}_9 \right] - F_0(T) \left[ T^n_9 - T^n_8 \right] - F_0(T) \left[ T^n_9 - T^n_{10} \right] \]
For Node 10,

\[-k(T)_{CFB} \frac{T_{10}^{n} - T_{11}^{n}}{\Delta y} - k(T)_{Air} \frac{T_{10}^{n} - T_{9}^{n}}{\Delta y} + \varepsilon_{CFB} \sigma \left( T_{9}^{n} - T_{10}^{n} \right) = \rho_{CFB} c(T)_{CFB} \left( \frac{\Delta y}{2} \right) T_{10}^{n+1} - T_{10}^{n} \]

\[T_{10}^{n+1} - T_{10}^{n} = \frac{1}{\rho_{CFB} c(T)_{CFB} \Delta y} \frac{2 \Delta t}{2} \left[ -k(T)_{CFB} \frac{T_{10}^{n} - T_{11}^{n}}{\Delta y} - k(T)_{Air} \frac{T_{10}^{n} - T_{9}^{n}}{\Delta y} + \varepsilon_{CFB} \sigma \left( T_{9}^{n} - T_{10}^{n} \right) \right] \]

\[T_{10}^{n+1} = T_{10}^{n} - \frac{1}{\rho_{CFB} c(T)_{CFB} \Delta y} \frac{2 \Delta t}{2} \left[ k(T)_{CFB} \frac{T_{10}^{n} - T_{11}^{n}}{\Delta y} - \frac{1}{\rho_{CFB} c(T)_{CFB} \Delta y} \frac{2 \Delta t}{2} k(T)_{Air} \frac{T_{10}^{n} - T_{9}^{n}}{\Delta y} + \frac{1}{\rho_{CFB} c(T)_{CFB} \Delta y} \frac{2 \Delta t}{2} \varepsilon_{CFB} \sigma \left( T_{9}^{n} - T_{10}^{n} \right) \right] \]

\[T_{10}^{n+1} = T_{10}^{n} - \frac{k(T)_{CFB}}{\rho_{CFB} c(T)_{CFB} \Delta y} \frac{2 \Delta t}{2} \left[ T_{10}^{n} - T_{11}^{n} \right] - \frac{k(T)_{CFB}}{\rho_{CFB} c(T)_{CFB} \Delta y} \frac{2 \Delta t}{2} \left[ k(T)_{Air} \left[ T_{10}^{n} - T_{9}^{n} \right] + \frac{k(T)_{CFB}}{\rho_{CFB} c(T)_{CFB} \Delta y} \frac{2 \Delta t}{2} \varepsilon_{CFB} \sigma \left( T_{9}^{n} - T_{10}^{n} \right) \right] \]

\[T_{10}^{n+1} = T_{10}^{n} - \frac{2k(T)_{Air}}{k(T)_{CFB}} \left[ T_{10}^{n} - T_{9}^{n} \right] + F(T) \frac{2 \Delta y}{k(T)_{CFB}} \frac{2 \Delta t}{2} \frac{\varepsilon_{CFB} \sigma \left( T_{9}^{n} - T_{10}^{n} \right)}{2 - \varepsilon_{CFB}} \]

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In order to obtain temperature profiles with a given incident heat flux, $q_i^*$, the procedure below is repeated. In advance, the number of cell in $y$ – direction, $J$, is decided and $\Delta y$ is calculated. $\Delta y = \frac{D}{J}$ where $D$ is depth, thickness of ceramic fiberboard. At the time step $n$,

1. $k(T)_{CFB}$ and $k(T)_{Air}$ are calculated only at the surface and air gap.

2. $\alpha(T)_{CFB}$ and $\alpha(T)_{Air}$ are calculated for all nodes.

3. $\Delta t$ is calculated for all nodes. $\Delta t = \frac{(\Delta y)^2}{2\alpha(T)}$

4. $\Delta t$ is determined based upon the minimum value of $\Delta t$ since $\Delta t$ for ceramic fiberboard is much greater than $\Delta t$ for air gap.

5. $Fo$ is calculated for all nodes. $Fo(T) = \frac{\alpha(T)\Delta t}{(\Delta y)^2}$

6. Confirm if $Fo$ meets the stability criteria, $Fo \leq \frac{1}{2(1 + Bi)}$, where

$$Bi = \frac{h_{total} \Delta y}{k}, \quad h_{total} = h_{conv} + h_{rad}.$$  

7. The maximum value of $Fo$ is chosen since $Fo$ for ceramic fiberboard is much smaller than $Fo$ for air gap.

8. With the values above, $T_j^{n+1}$ is calculated for all nodes.
For the calculation of temperature, 5 worksheets are engaged as seen in Figure 180: 1) Temperature 2) $k(T)_{CFB}$ and $k(T)_{Air}$ 3) $\alpha(T)_{CFB}$ and $\alpha(T)_{Air}$ 4) $\Delta t$ 5) $Fo(T)$.

**Figure 180** Calculation of temperature using temperature dependent thermal conductivity and thermal diffusivity.
REFERENCES


