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Heat transfer coefficients and other performance parameters for variously positioned and supported thermal loads in ovens with/without water-filled or empty blockages

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Abstract

An in-depth experimental study of heat transfer in ovens has provided basic data that is directly applicable to design. Heat transfer coefficients were measured for thermal loads having either black or highly reflective surface finishes. Approximately 100 different data runs were carried out. These heat transfer coefficients enabled the separation of the heat transfer into convective and radiative components, with radiation being the dominant transfer mechanism for blackened loads. The thermal response of the load to the presence of blockages situated either below or above the load was quantified. This response was only slightly affected by the blockages when they were empty of water, but major effects were observed when the blockages were water filled. Major effects were also encountered when the load was supported from below by cookie sheets. On the other hand, extensive investigation of various positions throughout the oven indicated a very weak effect of load position on the thermal response.

1. Introduction

This is the second and final report of an in-depth experimental study of the heat transfer processes that occur in an oven-like enclosure. The first paper [1] dealt with issues such as the surface finish of the thermal load, of the walls of the oven, and of the sensor that serves to control the temperature of the oven. In addition, experiments were performed to determine the effect of open or closed oven vents on the heating rate of the thermal load. In the present paper, a new set of issues was examined including: (a) blockages situated either above or below the thermal load, (b) the extent and nature of the blockages, (c) the presence or absence of water in the containers which constituted the blockages, (d) the presence of multiple blockages, (e) the positioning of the load either on a sheet or on a rack, and (f) the spatial positioning of the load within the oven. It was thought appropriate to examine the effect of the presence of water inasmuch as water-filled pans or foodstuffs heavily laden with water are commonly encountered in cooking operations.

A major focus of the work was to measure and report definitive information for heat transfer coefficients for thermal loads. These heat transfer coefficients are provided for each of the categories of geometry and operating conditions that were identified in the preceding paragraph. It is anticipated that these heat transfer coefficients would be of direct utility in the design and operation of ovens. The experimental values will be compared to the available literature information [2] that is specific to external natural convection from isolated bodies in large spaces.

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Nomenclature

A	surface area of the thermal load (m^2)
$\bar{h}$	effective heat transfer coefficient (W/m^2 K)
Q	rate of heat transfer to the thermal load (W)
T_{LOAD}	temperature of the thermal load (°C)
T_{OVEN}	oven set-point temperature (°C)
Greek symbols	
ε_{LOAD}	emissivity of the thermal load

Another important focus of the work was to determine the effect of the aforementioned conditions (a)–(f) on the heating rate of the thermal load. In addition, measurements were made of the distribution of the air temperature within the oven enclosure and of the temperatures of the wall bounding the enclosure. Special attention was given to the effects of both the onset and the cessation of boiling of the water in the containers used as blockages.

The literature survey prepared for [1] applies here without change. In particular, there was no literature unearthed by the authors which conveyed information of the nature that is set forth either here or in [1]. It is, therefore, recommended that readers interested in information collateral to that of the present study examine the references listed in [1].

The experimental work reported here and in [1] will be followed on by comprehensive numerical investigations encompassing the three-dimensional geometry, the unsteadiness of the heating, and the three modes of heat transfer which govern the thermal processes which are relevant to the heating of the load. It is expected that the combination of the experimental results and the numerical solutions will serve as a basis of design of ovens in the future.

2. Experimental equipment and procedure

The stove utilized for this experimental study was a commercial model, Kenmore Model #22 94492. It is equipped with an oven having dimensions $48 \times 61 \times 48$ cm^3 ($19 \times 24 \times 19$ in.^{3}), depth by width by height. The oven is heated by a Calrod heating element which is situated just above the oven floor. A shiny-surface temperature sensor is located at the top rear of the oven and centered in the width of the oven enclosure. A vent positioned at the lower edge of the oven door allows ingress of air that passes upwards through the oven and exits through a circular aperture in the right-rear surface burner. The experiments were performed by introducing the thermal load into a room-temperature oven cavity and then initiating the heating process. Therefore, the temperatures of the thermal load and the oven cavity increased simultaneously. All experimental runs were made with an oven set-point temperature of 177 °C (350 °F). The duration of each run was at least 2 h.

During the course of each data run, multiple temperatures were recorded by thermocouples in the thermal load, the air, and the walls of the oven. The data were collected by means of a Hewlett–Packard Model #34970A data logger. The thermocouples and the data logger were calibrated together with the aid of a constant temperature water bath and a NIST-traceable mercury-filled thermometer with 0.1 °F graduations. Collected data were transmitted to a personal computer that facilitated continuous recording and subsequent data processing. Careful attention was given to the influence of thermal radiation on the accuracy of the thermocouple-measured temperatures.

As discussed in [1], it was judged advantageous to use aluminum blocks as the thermal load. The radiation surface conditions of the load consisted of a black-painted finish, an encompassing, tightly fitting shiny aluminum foil covering, or a smooth commercial finish. The expected radiation emissivities of these surfaces are 0.9, 0.05, and 0.15, respectively.

Two types of blockages were employed during the course of the investigation. One type consisted of cookie sheets of dimensions of 45.5×29.3 cm^2 (17.9×11.5 in.^{2}).

Two such sheets were used, which were variously black painted or tightly wrapped with shiny aluminum foil. Both sheets have turned-up edges which allowed them to be filled with water. The other type of blockage was a Pyrex dish with dimensions 22×25.5 cm^2 (8.7×10.0 in.^{2}). The Pyrex dish was variously used either filled with water or left empty. Further details of the geometry and instrumentation can be found in [1].

3. Results and discussion

3.1. Heat transfer coefficients

The result of most immediate practical importance is the presentation of heat transfer coefficients for the thermal load. These heat transfer coefficients were determined from the basic definition

$$\bar{h} = Q/A(T_{OVEN} - T_{LOAD}), \tag{1}$$

where $\bar{h}$ is the effective heat transfer coefficient, Q is the rate of heat transfer to the thermal load, A is the surface area of the thermal load, T_{OVEN} is the oven set-point temperature, and T_{LOAD} is the temperature of the thermal load.

The heat transfer coefficient values reported here were obtained by evaluating all the quantities which appear on the right-hand side of Eq. (1). The rate of heat transfer Q was determined from the temperature versus time measurements recorded for the thermal load. A smooth curve was fitted to various segments of the experimental data. This curve was differentiated with respect to time, and by multiplying the derivative by the mass and the specific heat of the load, the corresponding value of Q was obtained. This procedure was performed at various times during the heating process of the load. Supplementary measurements had demonstrated that at any moment of time, the temperature of the thermal load was spatially uniform. The values of Q determined as just described depended on the time at which the derivatives were taken. Also time dependent was the temperature of the load and, correspondingly, $(T_{OVEN} - T_{LOAD})$. When the ratio of Q to $(T_{OVEN} - T_{LOAD})$ was taken, the result was independent of time. That is, the resulting heat transfer coefficients were quasi-steady.

There are so many independent geometric and operating parameters of the present study that it is appropriate to provide some definition of their range before presenting a tabulation of the heat transfer coefficients. Fig. 1 was prepared to illustrate the geometric parameters which were investigated. The figure is made up of eight diagrams, each defining one of the investigated configurations. For instance, attention may be turned to Fig. 1(a) which shows the thermal load resting on an oven rack with the Calrod heater situated at the very bottom of the oven cavity. Fig. 1(g) portrays the thermal load supported from below on a cookie sheet with a sheet-like blockage below. Figs. 1(a)–(g) have a common format as was just illustrated. In contrast, Fig. 1(h) is a three-dimensional schematic which is included to define the positions LF, RF, LR, and RR, which respectively refer to left front, right front, left rear, and right rear. These positions correspond to various locations in a horizontal plane at which the thermal load was placed during one phase of the investigation.

In certain of the investigated operating conditions, the blockages were filled with water while in other cases, the blockages were empty. In addition, the blockages and sheets had various surface finishes as was also the case for the thermal load and for the oven walls.

The experimentally determined values of the effective heat transfer coefficients are listed in Table 1. The table identifies the operating conditions via the description of the various cases which are elucidated in the next paragraphs. In the second column, the geometric configuration is described by reference to Fig. 1. The next column in the table lists the radiative emissivity of the thermal load. Columns four and five convey the heat transfer coefficient results and the spread of the data with respect to the listed mean values.

Case A includes: (a) a free-standing, geometrically centered load (base case); (b) a free-standing, horizontally centered load located near the top and bottom of the oven; (c) a free-standing, geometrically centered load situated in an oven whose walls have various surface finishes including the natural finish as well as aluminum foil covering any number of the six bounding walls.

Case B includes: (a) and (c) from case A and, additionally, (d) a free-standing, geometrically centered load with cookie sheets below; (e) a free-standing, geometrically centered load with water-filled cookie sheets below; (f) a free-standing, geometrically centered load with a Pyrex dish below; (g) a free-standing load situated at fifteen different locations within the oven.

Case C includes: (h) a free-standing, geometrically centered load with cookie sheets above; and (i) a free-standing, geometrically centered load with water-filled cookie sheets above.

Case D includes: (j) a geometrically centered load resting on a blackened cookie sheet; and (k) a geometrically centered load resting on a blackened cookie sheet with another cookie sheet below.

Case E includes: (l) a geometrically centered load resting on a shiny cookie sheet and (m) a geometrically centered load resting on a shiny cookie sheet with another cookie sheet below.

Case F includes: (a), (d), (f)–(h), and, additionally, (n) a free-standing, geometrically centered load with a Pyrex dish above.

Case G includes: (c).

Case H includes: (j)–(m).

Case I includes: (i).

Case J includes: (o) a free-standing, geometrically centered load with a water-filled Pyrex dish above.

Case K includes: (e) and (p) a free-standing, geometrically centered load with a water-filled Pyrex dish below.

An overall examination of the table indicates that the effective heat transfer coefficients arrange themselves into two loose groupings. Those cases characterized by a high value of the radiative emissivity of the thermal load tend to have relatively higher effective heat transfer coefficients than those cases characterized by low values of the emissivity. These findings reinforce the notion that both radiation and convection contribute to the numerical value of the transfer coefficient. Indeed, the decisive role played by the emissivity of the surface of the load underscores the importance of the radiative component. For the low-emissivity surfaces (cases A–E) the values of the heat transfer coefficients fall in the range from 6 to 11 W/m^2 °C. The compactness of this range is remarkable when it is considered that there is an enormous number of permutations and combinations of both geometrical as well as operating conditions. The

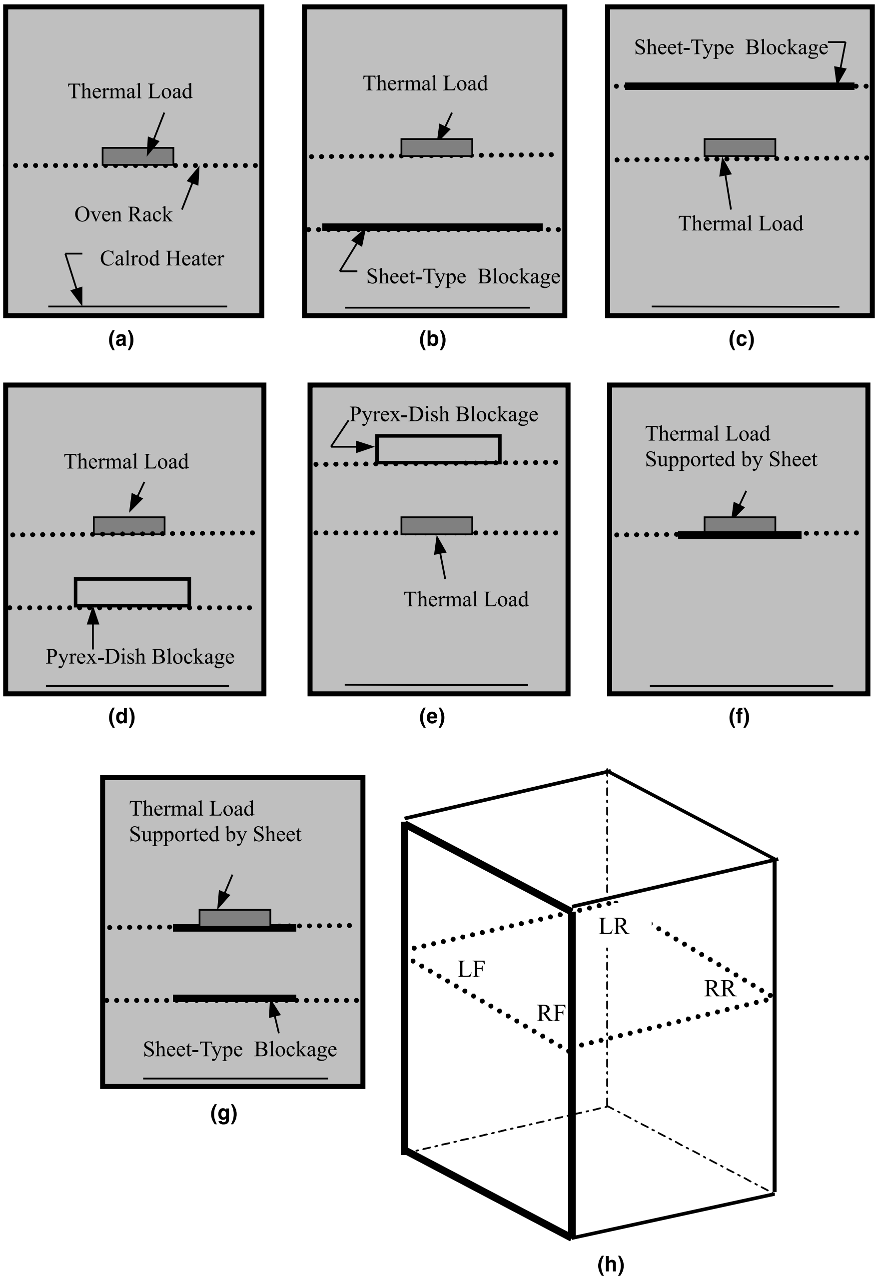

Fig. 1. Investigated geometries.

minimum encountered heat transfer coefficient, namely 6, corresponds to situations in which there is a free-standing thermal load situated at various locations within the oven, with or without blockages below. At the other end of the range, 11, the thermal load is situated in direct contact with a blackened cookie sheet.

For the high-emissivity surfaces, the effective heat transfer coefficients are, for the most part, of the order of the low twenties. This relatively elevated value, compared with the nominal value of 8 for the low-emissivity loads, actually suggests that the radiative effect may outweigh the convective effect. The lowest of the heat transfer coefficients for the black-surface load, about 14, corresponds to a free-standing, geometrically centered load with water-filled blockages below. The highest of the $\bar{h}$ values, 26, relates to a free-standing, geometrically centered load situated in an oven whose walls are variously covered with aluminum foil.

Table 1
Effective heat transfer coefficients

Case	Configuration	ε_{LOAD}	$\bar{h}$ (W/m^2 °C)	Range (%)
A	Figs. 1(a) and (h)	0.15	9.5	±7
B	Figs. 1(a), (b), (d) and (h)	0.05	6.0	±15
C	Fig. 1(c)	0.05	9.0	±18
D	Figs. 1(f) and (g)	0.05	10.9	±13
E	Figs. 1(f) and (g)	0.05	8.0	±8
F	Figs. 1(a), (b), (c), (d), (e) and (h)	0.9	18.7	±16
G	Fig. 1(a)	0.9	25.6	±16
H	Figs. 1(f) and (g)	0.9	18.1	±15
I	Fig. 1(c)	0.9	23.2	±12
J	Fig. 1(e)	0.9	23.2	±3
K	Figs. 1(b) and (b)	0.9	14.2	±11

To provide perspective for these results, it is noteworthy that they are independent of the two sizes of thermal loads used in the course of the experiments. The dimensions of the two thermal loads were 5 cm high by 15.9 cm deep by 18.7 cm wide (2 inches by 6.25 inches by 7.375 inches) and 2.5 cm high by 15.9 cm deep by 18.7 cm wide (1 inch by 6.25 inches by 7.375 inches) respectively. This insensitivity is regarded as a positive outcome. It is also relevant to compare the experimentally determined effective heat transfer coefficients with any appropriate results which may have appeared in the literature. The literature survey unearthed an experimentally based correlation for pure natural convection from heated block-like objects situated in a large, wall-less space filled with quiescent air [2]. When the present operating conditions for the base case were used to evaluate the correlation from [2], a value of the surface-averaged natural convection heat transfer coefficient was found to be approximately 5 W/m^2 K. From Table 1, the smallest value of the effective heat transfer coefficient determined in the present study is 6 W/m^2 K. If the latter is taken to correspond to the case of minimal radiative heat transfer, then the agreement between the present experimental result and the aforementioned correlation is seen to be very good.

3.2. Free-standing thermal load in the presence of blockages

The presentation of results will continue with a display of data for the case in which the thermal load was positioned on an oven rack and blockages were placed either above or below the load, Figs. 1(b and c). The first blockage investigated was comprised of the two aforementioned cookie sheets placed side by side, with both sheets being completely blackened. One net effect of such a blockage was to more or less subdivide the total height of the oven into two separate subsections one atop the other. This blockage spanned a total width of 58.5 cm (23.0 in.) and a total depth of 45.5 cm (17.9 in.) in the 48×61 cm^2 (19×24 in.^{2}) horizontal cross-section of the oven. In some of the investigated cases, the cookie sheets were filled with water to a height of 1.27 cm (1/2 in.), while in other cases the sheets were left empty.

The thermal load was centered in the oven cavity, both vertically and horizontally. The blockage was positioned either 12.7 cm (5 in.) above or below the rack which supported the load.

The timewise variation of the temperature of the thermal load is presented below in Fig. 2. To facilitate a compact display of the results, two independent ordinate scales have been employed in the figure. These scales are displaced vertically by 40 °C, which corresponds to the vertical gradation between two adjacent horizontal gridlines. In comparing the two sets of results conveyed by Fig. 2, it is necessary to keep the displacement of the ordinates in mind.

The results are grouped according to whether they correspond to the black-painted load or to the foil-wrapped load. In each grouping, curves representing five different operating conditions are shown. One of the curves, labeled *base case*, corresponds to the situation in

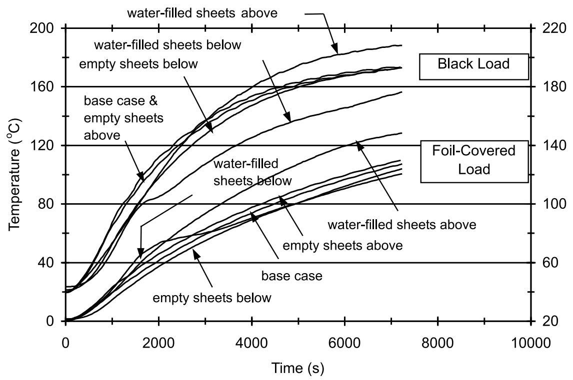

Fig. 2. Thermal response of the load to various blockage positions and to the presence and absence of water in the cookie sheets which form the blockages.

which the thermal load is placed in an oven without blockages. The base case curve is positioned among curves representing blockages situated both above and below the thermal load. In particular, for both the above-positioned and the below-positioned blockages, one of the curves corresponds to water-filled cookie sheets and the other to empty cookie sheets.

An overall examination of Fig. 2 reveals a number of broad generalizations. First of all, it is seen that the heating of the black load is more sensitive to the presence or absence of blockages than is the heating of the foil-covered load. Compared with the heating rate of the base case, the presence of empty sheets above the thermal load gives rise to only a very modest change. However, for the case of empty sheets below, the heating rate of the load is diminished with respect to the base case to the extent that there is a 14 °C temperature difference for the blackened thermal load at intermediate times. The corresponding deviation is about 6 °C for the foil-covered load.

With regard to water-filled sheets above, their presence significantly increases the temperature of the load at intermediate and large heating times for both blackened and foil-covered loads. This finding clearly demonstrates an enhancement of the heating which may be of practical importance. On the other hand, when water-filled sheets are below, there is a tendency for the load temperature to be lower than those of the respective base cases.

To add plausibility to the foregoing results, it may be useful to recall the well-known radiation shield problem that is a feature of most introductory heat transfer courses. That problem is comprised of two parallel walls that face each other and feature an opaque radiation shield placed in the space between these walls. The well-known result is that the radiation shield diminishes the rate of heat transfer between the walls. In the present situation, it was found experimentally that the presence of a blackened sheet-type blockage placed between the heat source and the thermal load resulted in a reduced heating rate of the load. This reduced heating rate can be directly attributed to the radiation shield effect of the sheet-type blockages. It is reasonable to expect that this effect would be more pronounced for the blackened load in that it is generally more sensitive to radiant heat than would be a foil-covered load.

With regard to the role of the empty sheets positioned above the thermal load, there is no precedent in radiation theory for the load to be affected by shields that are not positioned between a heat source and a load. This expectation was verified by the experimental data.

It is of particular interest to seek an explanation for the experimental fact that the presence of water-filled sheets above the load leads to an enhancement of the temperature of the thermal load. The explanation was provided by an inspection of the timewise temperature variations of thermocouples affixed to the walls of the oven. In the absence of any blockage in the oven, the heating portion of a heating–cooling cycle was slightly shorter than the cooling portion of the cycle. On the other hand, with water-filled sheets in place above the thermal load, the heating portion of the cycle was considerably longer (for example, by a factor of three) than the cooling portion of the cycle. As a consequence, in the presence of water-filled sheets above, the aforementioned longer heating times gave rise to higher heating rates. The explanation of the longer heating times follows immediately from the fact that the temperature of the water in the sheets does not exceed 100 °C (212 °F). The water is an up-facing radiant source, and the limitation on its temperature gives rise to a diminished flux of radiant energy incident on the temperature sensor that is used to provide input to the control system of the oven. The deficit of the radiant flux incident on the sensor, compared with cases in which there are no up-facing, water-filled sheets, causes the oven control system to be in the *on* mode of the cyclic heating period for a longer duration.

It still remains to explain the effect of water-filled sheets on the heating performance when the sheets are positioned below the thermal load. In this regard, it is once again relevant to note that the water temperature is limited to 100 °C (212 °F). This low-temperature radiation replaces direct high-temperature radiation from the Calrod heater when there are no blockages below the thermal load. The impingement of the low-temperature radiation on the blackened load significantly reduces the rate at which radiation is absorbed by the load, thereby explaining the lowering of the temperature of the load compared to the base case. On the other hand, the presence of the low-temperature radiation has a lesser effect on the temperature of the foil-covered load because the high reflectivity of the foil effectively reduces the importance of the radiation mode of heating compared with the convective mode.

A final glance at Fig. 2 reveals that certain of the curves are punctuated by what would appear to be inflection points. The study of this behavior and its relevance is most conveniently addressed by making reference to Fig. 3. In this figure, the curves of Fig. 2 which relate to water-filled sheets have been replotted. In addition, markers, either blackened circles or blackened squares, have been placed to indicate locations that correspond to the inflection points that were noted in the foregoing discussion.

Further inspection of the figure indicates that there are actually two types of inflection points indicated. Those inflection points that occur at relatively short times are marked by a blackened circle while those that occur at larger times are indicated by a black square. The lower inflection points correspond to the onset of

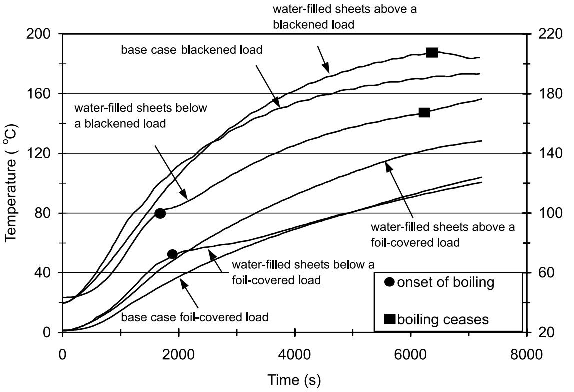

Fig. 3. Thermal response of the load to various water-filled blockage positions.

boiling of the water contained in the sheets, and the upper inflection points represent the cessation of boiling and, thereafter, the sheets are no longer water-filled. It is especially interesting to note that the lower inflection points are present only when the water-filled sheets are situated below the thermal load.

To add plausibility to this latter finding, it should be recognized that the boiling process gives rise to ascending buoyant plumes of water vapor. These plumes provide a new heat transfer mechanism that causes a change in the heating rate of the load. This new convective heat transfer mechanism actually carries relatively cool fluid (vapor whose temperature is approximately 100 °C (212 °F)) to the lower face of the thermal load. In consequence, the rate of heating of the load diminishes compared with the situation which prevailed before the onset of boiling, hence the lower inflection points. The impact of the new convective heat transfer mechanism is greater for the foil-covered load than for the blackened load since the former is more dependent on convection (in contradistinction to radiation) for its heating.

With regard to the absence of lower inflection points for the case in which the water-filled sheets are situated above the thermal load, it is sufficient to note that the buoyant plumes of vapor do not impact the load. Therefore, no new convective mechanism for heating of the load is activated at the onset of boiling.

Attention will now be turned to the upper inflection points. It is noteworthy that the indicated inflection points correspond to cases for which the thermal load had a blackened surface finish. The cases where the load was foil-covered do not show any evidence of inflection. The indicated upper inflection points are actually of a somewhat different nature. For the case in which the water-filled sheets are above the thermal load, the inflection point marks a decrease in the temperature of the thermal load. This decrease is caused by the restoration of the normal on–off operating mode of the oven, in which the periods of heating and non-heating are of similar duration. Prior to this restoration, the duration of the heating period significantly exceeded the duration of the non-heating period as was explained in connection with Fig. 2.

The second of the upper inflection points corresponds to the case in which the water-filled sheets are below the thermal load. Careful inspection of the relevant curve in Fig. 3 indicates that that inflection point corresponds to a slight increase in the heating rate of the thermal load. This change in the heating rate can properly be attributed to the rapid rise of the temperature of the sheets when the water boils away. This temperature rise transforms the sheets into high-temperature radiators, with a consequent higher rate of delivery of thermal energy to the down-facing surface of the thermal load.

The absence of the upper inflection points for a foil-covered load is related to the relatively weak radiation transport.

To add further perspective to these results, it is useful to make reference to temperature measurements, in addition to those of the thermal load, which included the air, oven sidewalls, and the sheets. Fig. 4 has been prepared in this connection. This figure corresponds to the blackened thermal load and to the water-filled sheets situated beneath the load. The figure conveys information for the timewise temperature variation of the load, of the air above the load, and of the sheets. The thermal load shows both the lower and upper inflection points that had been already exhibited in Fig. 3. The air temperature distribution reveals an early period of sustained rise which corresponds to the oven operating in the preheat mode during which the temperature controller calls for continuous heating. The cessation of the continuous heating regime corresponds to the time at which

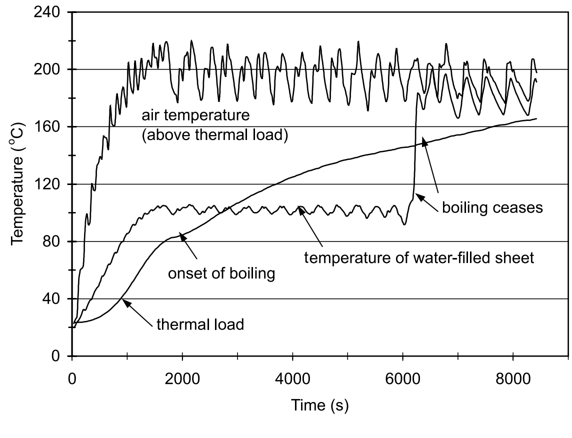

Fig. 4. Temperatures of the thermal load, water-filled sheet, and air for the case of a blackened thermal load with water-filled sheets beneath the load.

boiling begins. Thereafter, the heating follows a cyclic pattern that is mirrored by the periodic rise and fall of the air temperature. Concurrently, the temperatures of the water-filled sheets (measured on the bottom face of the sheets) take on a periodic rise and fall about a steady mean value of approximately 102 °C (216 °F). Abruptly, at about 6000 s, the temperatures of the sheets rise precipitously, signaling the cessation of boiling and the disappearance of liquid water from the sheets. The rapid rise of the sheet temperature brings it into close conformity with the air temperature. The consequence of the increase of the sheet temperature is a more rapid heating of the thermal load, hence the upper inflection point in the temperature variation of the thermal load.

To complement the information conveyed in Fig. 4, measured sidewall temperatures are exhibited in Fig. 5 along with that of the thermal load. As was the case for the air temperature rise during the initial heating period, so also is there a sustained rise in all of the temperatures measured on the sidewalls. These sustained temperature rises cease at the moment when boiling begins. Thereafter, a periodic regime sets in where the various sidewall temperatures oscillate about a constant mean value. At the moment when the boiling ceases, indicated by the upper inflection point in the thermal load's temperature variation, the sidewall temperatures exhibit marked changes. The temperatures in the lower portion of the sidewall decrease while those in the upper portion increase. The increase of the upper wall temperatures results from the increased rate of arriving radiation brought about by the higher temperature of the sheets that radiate to that portion of the walls. On the other hand, the decrease of the temperatures on the lower portion of the wall is caused by a change in the duration times of the heating and non-heating portions of the oven's thermal cycle.

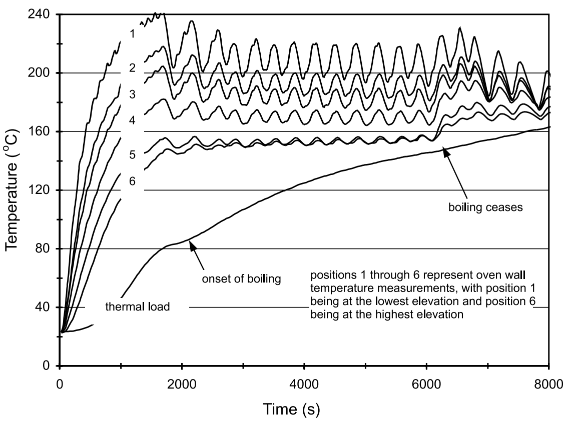

Fig. 5. Temperature variations of the oven sidewall for the case of a blackened thermal load with water-filled sheets beneath the load.

The preceding blockage-related results pertained to sheet-like blockages (for example, cookie sheets). Additional experiments were performed with a different type of blockage, namely, a Pyrex baking dish. The dimensions of the Pyrex dish have been presented earlier in the paper. A major difference between the sheet-type blockages and the Pyrex-dish blockage is the considerable larger space between the outboard edges of the blockage and the oven walls. The results for the heating of the thermal load in the presence of Pyrex-dish blockages are presented in Fig. 6. The figure conveys information for the base case (no blockages) as well as for both water-filled and empty blockages both below and above the thermal load as illustrated in Figs. 1(d and e).

As seen in the figure, the presence of the blockages gives rise to diminished load temperatures for all of the investigated cases. The largest depression of the load temperature (~17 °C, ~30 °F) occurs when there is a water-filled dish beneath the load. This behavior is a result of the limitation of the water temperature to 100 °C (212 °F) and the corresponding diminution of the radiant emission of the water surface compared with that of the Calrod heater. A second factor is the presence of cool, buoyant plumes of water vapor rising from the boiling surface and impinging on the downward face of the load. When the Pyrex dish is below and empty of water, there is a diminished radiation to the load but to a lesser extent than that of the case for the water-filled Pyrex dish below. For the purpose of comparison, there is also displayed the load temperature variation in the presence of a single, empty, blackened cookie sheet placed beneath the load. It is interesting to observe that the single empty cookie sheet and the empty Pyrex dish have a similar effect on the heating of the load.

When the Pyrex dish is positioned above the load, there is a decrease in the load temperature which is the same regardless of whether or not there is water or no water in the dish. The insensitivity to the presence of water can be explained by the moderate radiant inter-

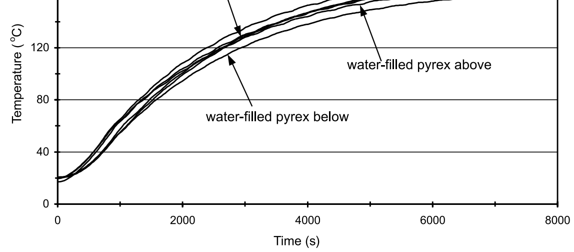

Fig. 6. Thermal response of the blackened thermal load in the presence of Pyrex-dish blockages.

action between the bottom of the relatively small dish and the top of the thermal load.

3.3. Thermal load situated atop a sheet-like blockage in the presence and absence of other blockages below

In the foregoing presentation of results, attention was given to a free-standing thermal load (situated on a standard oven rack). Now, the focus will be shifted to thermal loads which are supported from below by a sheet, for example a cookie sheet, as illustrated in Figs. 1(f and g). The supporting sheet was either blackened or foil-covered. Blockages, also either blackened or foil-covered, are situated beneath the load and its supporting sheet. Two types of surface finishes of the load were investigated, blackened and foil covered. The results of this investigation are presented in Figs. 7 and 8, respectively, for the blackened load and for the foil-covered load.

Prior to a discussion of the results conveyed by the figures, it is appropriate to elucidate the nomenclature which appears therein. The surface finish of the sheet on which the load rests is indicated by the notation: *on S* or *on B*. In turn, if there is a blockage beneath the load, its surface finish follows the designation of the surface finish of the supporting sheet. For instance, *on B*, *B below* refers to a load which is resting on a black-surfaced sheet with a blockage below which is also black surfaced.

Examination of Figs. 7 and 8 reveals certain generalizations. For one, it is clear that the blackened thermal load always yields a higher heating rate than does a foil-covered load, regardless of any of the other investigated parameters. Furthermore, the cases related to the blackened load are less sensitive to the various operating conditions than are the cases related to the foil-covered load. In both of the figures, more rapid heating is accomplished when the load is situated atop a blackened supporting sheet than when it is situated atop a foil-covered sheet.

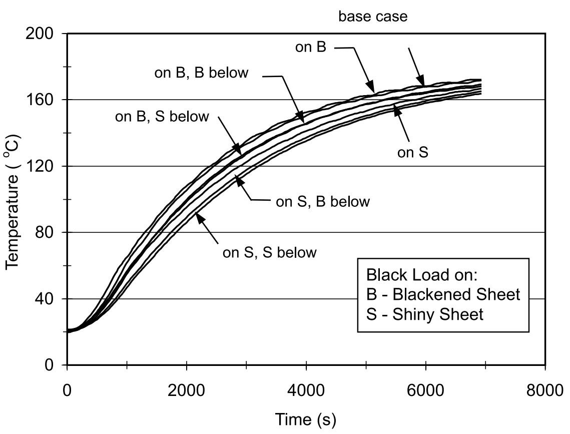

Fig. 7. Thermal response of the blackened thermal load to the presence of a supporting sheet with and without other blockages situated beneath the load.

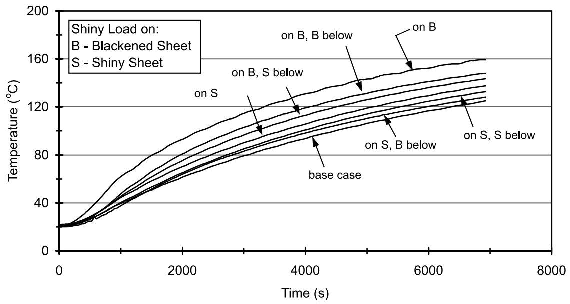

Fig. 8. Thermal response of the foil-covered thermal load to the presence of a supporting sheet with and without other blockages situated beneath the load.

It is also interesting to compare the effect of the presence of a supporting sheet with the base case in which the thermal load is free standing (resting upon an oven rack) and where there are no blockages. For the case of a blackened load, the base case provides the highest heating rate, i.e. the presence of the supporting sheet and the blockages reduces the heating rate. In contrast, for the case of the foil-covered load, the base case provides the slowest rate of heating, and the presence of a supporting sheet and of blockages increases the heating rate.

3.4. Thermal load positioned in various locations in the oven

The final focus of this study is the issue of the location of the load in the oven and its influence on its thermal response. All told, the number of investigated positions was 15, encompassing three elevations and five locations within each elevation. To achieve a concise presentation of the data, results will be presented only for the lowest and highest elevation, respectively, 10 and 36 cm (4 and 14 in.) above the floor of the oven. The locations at which the load was positioned at a given elevation are illustrated in Fig. 1(h). The data for the blackened load are presented in Fig. 9, while those for the foil-covered load are shown in Fig. 10. The experiments on the position-related effects were performed using a thermal load whose mass was one half that of the thermal load used for the experiments set forth in the preceding sections of the paper. Therefore, any comparisons of between the heating rates of Figs. 9 and 10 with those set forth in previous figures would not be meaningful.

Both figures are laid out in a common format. There are two ordinate scales to enable the separate presentations for the two elevations to be made on a single graph

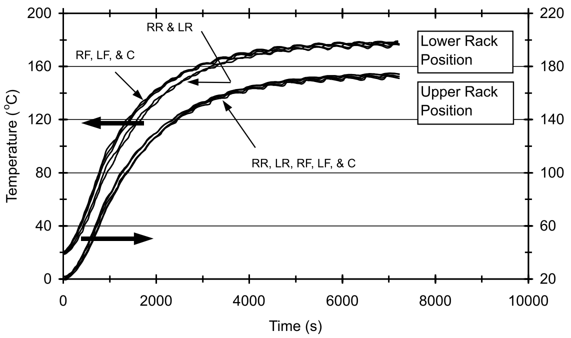

Fig. 9. Thermal response of the blackened thermal load to the position of the load in the oven.

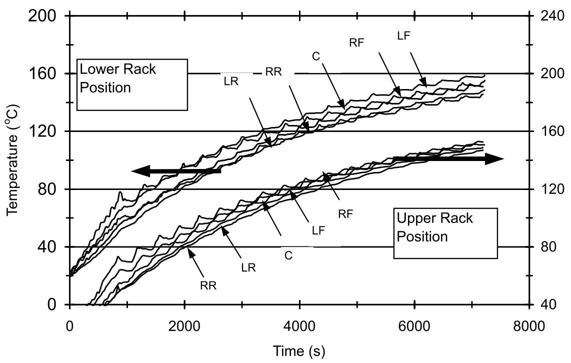

Fig. 10. Thermal response of the foil-covered thermal load to the position of the load in the oven.

without data overlap. To this end, the left-hand ordinate corresponds to loads positioned at the lowest elevation and the right-hand ordinate corresponds to the loads at the upper position. Note that the use of different ordinate scales gives rise to the downward shifting of the results at the upper elevation. To compare the results for the two elevations, it is necessary to imagine the translation of the lower set of curves upward by one horizontal gridline.

Attention will first be turned to the results of the blackened load of Fig. 9. The general impression conveyed by this figure is that the heating of the thermal load is only slightly affected by the horizontal position of the load within a given elevation. A more detailed look reveals that at the lowest elevation, the results for the various positions separate into two groups such that the higher heating rates occur at the front and center positions, whereas the lower heating rates occur at the rear positions. This finding is believed due to the shape of the Calrod heating element that favors the heating of the center and the front. At the highest elevation, it appears that there is no distinction between the heating rates at the various horizontal positions. Next, a comparison can be made between the results at the two elevations. Careful inspection of the curves suggests that, at most, the differences in the load temperature at any given time are about 6 °C (11 °F), with the higher temperatures corresponding to the lowest elevation. This finding is plausible inasmuch as the lower-elevation positions are much closer to the Calrod heater than are the upper-elevation positions.

Fig. 10 will next be discussed. The data conveyed by this figure indicate a greater sensitivity of the heating of the load to the horizontal position of the load. In particular, the sensitivity is accentuated at the lowest elevation. This accentuation is in accord with the reality that the heat transfer to a foil-covered load is primarily by natural convection and that the buoyant plumes which constitute the natural convection flow are stronger in the lower part of the oven than in the upper part of the oven. The position-related spread at the lowest elevation is in the range of 11–17 °C (20–30 °F), while the corresponding spread at the highest elevation is 4–17 °C (7–30 °F).

In common with the results already noted for the blackened thermal load, the highest temperatures at any given elevation occurred at the front and center, whereas the lowest temperatures at the same elevation occurred in the rear of the oven. Finally, a comparison between the temperatures at the two elevations reveals differences of the order of 3 °C (5 °F).

4. Concluding remarks

This investigation is dedicated to enlarging the knowledge base for the thermal design of ovens. It is the expectation of the authors that the information conveyed in this paper will be directly useful in improving the practice of oven design. Of immediate utility is the tabulation of effective heat transfer coefficients. These heat transfer coefficients were measured for approximately 100 different cases encompassing various positions of the thermal load; the presence and absence of blockages; radiative surface finishes of the load, the oven walls, and of the blockages; the presence and absence of water contained in the blockages; and loads that were either free standing or supported on surfaces such as cookie sheets. The heat transfer coefficients were grouped into coherent categories to facilitate their convenient use. These data revealed the relative importance of radiative heat transfer versus convective heat transfer. It was found that the presence of the strongest radiation influence increased the effective heat transfer coefficient by a factor of four compared to that for pure natural convection. A comparison of the pure convection heat transfer coefficients measured here showed good agreement with those evaluated from a literature correlation for natural convection about a heated solid situated in a large, otherwise quiescent space.

When blockages were placed either above or beneath the thermal load, the mere presence of the blockages did not cause significant changes in the heating characteristics of the load. On the other hand, when the blockages contained water, major effects were observed. In particular, when a water-filled blockage was positioned below the load, a lower heating rate resulted; the opposite effect occurred when such a blockage was positioned above the load. These behaviors were made plausible by examining the modes of heat transfer to the load and also by describing how the oven control temperature sensor is affected by the blockages and by thermal plumes that rise from water boiling in these blockages situated beneath the sensor.

Another major facet of the work involved the interaction between the thermal load with surfaces on which the load rested. It was found that when the load surface finish was blackened, the presence of a supporting surface always diminished the heating rate of the load compared with that of a free-standing load. In contrast, when the load was wrapped with a shiny foil, its heating rate was always enhanced when it rested on a supporting surface.

Finally, the effect of the location of the load in the oven was investigated. All told, fifteen locations were selected for this study. In general, it was found that location of a blackened load was not a major factor in determining its heating rate. For a foil-covered load, the effect of location was somewhat larger but never exceeded 17 °C (30 °F).

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