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Thermal characteristics of a thermosyphon heated enclosure

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Abstract

This work analyzes and compares the thermal characteristics of two enclosures heated using different methods: one is heated using two-phase thermosyphons and the other is heated with a more conventional approach; hot exhaustion gases flowing into the enclosure. The first type has been evaluated for application to baking ovens, while the second type is already commonly used for this application in some countries. The methodology used to compare the two enclosures was developed by the present authors and is presented in the literature. The results show that the enclosure heated using thermosyphons has more uniform temperature and radiative heat transfer coefficient distributions compared to the conventional approach. The conventional enclosure tends to present larger convective heat transfer coefficients than the thermosyphon assisted enclosure because of the exhaustion gases movement. The thermosyphon assisted enclosure present a very uniform temperature distribution, which causes the air to become stagnant and the heat transfer coefficients to be low. The radiation field inside the thermosyphon assisted enclosure is more uniform than the conventional enclosure. A more uniform radiation field is obviously better when applying the enclosure for cooking as it makes the process more even.

Keywords: Baking ovens; Thermosyphon; Heat transfer coefficient measurement

1. Introduction

Two-phase thermosyphons are high efficiency heat transfer devices: they can transport heat at a very high rate with a relatively small temperature drop, i.e., they present a low thermal resistance. There are innumerous applications to two-phase thermosyphons, from large heat exchangers for the petroleum industry to small water solar heaters. Faghri [1], Peterson [2], among others, review the theory and the applications of the two-phase thermosyphon technology. Apart from featuring a low thermal resistance, another interesting characteristic of two-phase thermosyphons is a very uniform temperature distribution in the condenser section when the heat transfer coefficient between the condenser and the external environment is small, such as in baking oven applications. Recently, Mantelli and co-workers [3–5] successfully applied two-phase thermosyphons to isothermalize enclosures. Their work was focused on applications to bakery ovens, especially for bread baking. The thermosyphon condensers were placed inside the enclosure, while the evaporators were confined in a combustion chamber below the enclosure. Heat was supplied by gas (propane-butane) burners inside the combustion chamber. As a result, a very uniform temperature distribution was obtained inside the enclosure, which translated to the bread being baked very evenly.

Another important characteristic of this approach is that the thermosyphons transfer the heat from the combustion chamber into the enclosure without mixing the exhaustion gases with the air inside the enclosure. The approach normally adopted in baking ovens in some countries is to heat the oven enclosure by means of a gas (propane/butane) burner placed bellow the enclosure bottom wall, at the centerline. The combustion gases flow into the enclosure through holes on the bottom wall and exit the enclosure through holes on the back wall. The temperature distribution inside the enclosure resulting from this approach presents considerably large variations [4]. The pres-

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Nomenclature

m	mass	kg
c_p	specific heat at constant pressure	$J \cdot kg^{-1} \cdot K^{-1}$
T	temperature	K
t	time	s
h	heat transfer coefficient	$W \cdot m^{-2} \cdot K^{-1}$
A	surface area	m^2
q	heat transfer rate	W

Greek letters

ε	surface emissivity
σ	Stefan–Boltzman constant, 5.67×10^{-8} $W \cdot m^{-2} \cdot K^{-4}$

Subscripts

rad	radiation
air	enclosure air
w	enclosure wall

ence of exhaustion gases inside the cooking chamber is also undesirable. Furthermore, intense thermal radiation from the bottom wall, which is closer to the gas burner than the other walls, makes the radiative heat flux distribution inside the enclosure very uneven, as it will be seen later. The closer to the bottom wall, the more intense is thermal radiation.

Milanez and Mantelli [6] proposed an experimental methodology to assess how even is the cooking inside the enclosure without effectively cooking any food at all. The basic idea is to measure the spatial distribution of the heat transfer coefficient between the enclosure and small aluminum blocks spread inside the enclosure. The objective of the present work is to apply the methodology proposed by Milanez and Mantelli [6] in the analysis of the thermal performance of two types of enclosure: one heated by thermosyphons and one heated by hot exhaustion gases.

2. Enclosure geometry

The geometry of the enclosure assisted by thermosyphons is presented in Fig. 1. It is basically composed of two mild steel sheets, which constitute the upper and the bottom walls and two aluminum sheets (side walls) attached to each other by means of riveted joints (a). The sheets are assembled in the form of a rectangular enclosure (b) with dimensions $0.38 \times 0.48 \times 0.61$ m. Eight stainless steel-water thermosyphons are attached internally to the side walls of the enclosure (c), so the side walls act as fins, helping to remove the heat from the thermosyphon condensers. The thermosyphon evaporators are tilted at 45° and are located inside a combustion chamber below the enclosure. Two metal sheets are riveted at the front and at the back of the enclosure (d). A 50 mm thick insulation blanket made of glass wool is wrapped around the enclosure walls and thermosyphons (e). Mild steel sheets are placed externally to protect the insulation blanket (f). A glass wool blanket is used to insulate the enclosure back wall (g). The front door, made of a glass wool blanket sandwiched by metal sheets, completes the enclosure (g). At the center of the front door there is a double glass window for inspection.

Eight 12.7 mm outer diameter and 10.2 mm inner diameter stainless steel-water thermosyphons are used. The condenser section of the thermosyphons is 270 mm long, there is no adiabatic zone and the evaporator is 90 mm long. The nominal filling ratio is 100% of the evaporator volume. A gas burner is

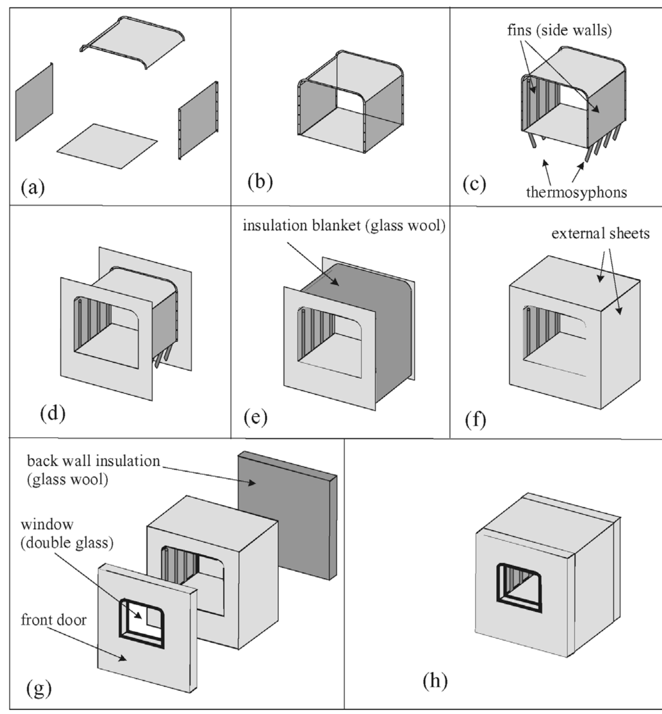

Fig. 1. Enclosure geometry.

placed bellow each row of evaporators. The evaporators and the burner are confined in a combustion chamber, completely separated from the enclosure.

The details of the thermosyphon/fin attachment is shown in Fig. 2. The fin was deformed in order to accommodate $1/3$ of the area of the condenser. The fin is sandwiched between the thermosyphon and a steel angle. A steel wire clamp is used to squeeze the fin against the thermosyphon. The function of the steel angle is to distribute the contact pressure more evenly over the interface, avoiding gaps where there is no effective contact, which would increase the thermal contact resistance between the thermosyphon and the fin. Between thermosyphon and the fin there is also an aluminum tape. The aluminum tape deforms under compression, helping to fill the gaps between the thermosyphon and the fin. Given the high thermal conductivity of aluminum, the tape also contributes to the decrease of the thermal resistance at the contact between the thermosyphons and the fins.

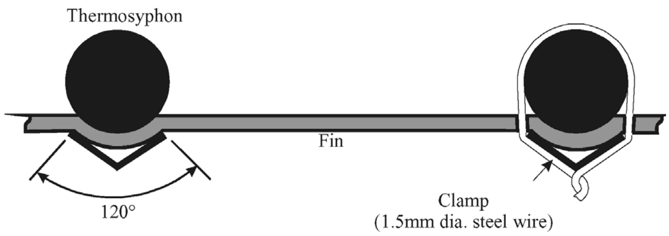

Fig. 2. Details of the thermosyphons/fin attachment.

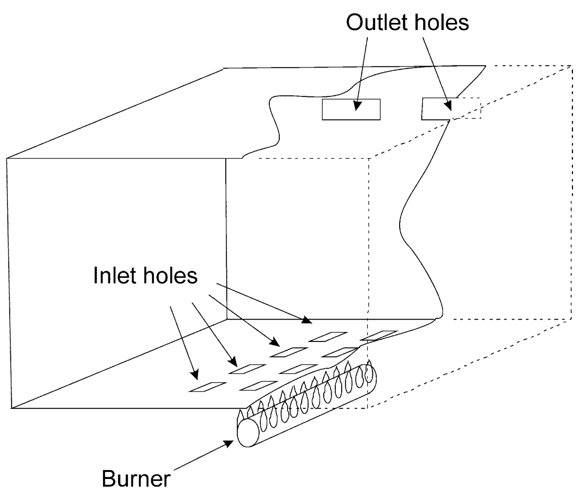

Fig. 3. Conventional approach.

The geometry of the conventional enclosure, i.e., the one heated by exhaustion gases, is similar to the enclosure described above. The basic difference is that there are no thermosyphons and no combustion chambers in the conventional enclosure. The side walls are made of mild steel sheets, like the other enclosure walls. The gas burner is horizontally placed underneath the bottom wall of the enclosure and in the longitudinal direction. The burner has a cylindrical shape, with a diameter of 40 mm and a length of 400 mm. The hot exhaustion gases flow into the enclosure through 8 holes (25×35 mm each) on the bottom wall and right above the burner and exit the enclosure through 2 holes (30×80 mm) on the upper end of the back wall. Fig. 3 shows the heating approach of the conventional enclosure in more details.

3. Experimental program

The experimental study consisted basically of measuring the temperature distribution inside the enclosures as well as the heat transfer coefficient between the enclosure and small aluminum blocks spread inside it. Two different enclosures were tested: one heated using the conventional approach, i.e., hot exhaustion gases flowing into the enclosure, and the other using thermosyphons, as described in the last section.

Two types of tests were conducted: transient and steady state. The transient tests consisted of turning the gas burner on from thermal equilibrium at room temperature (start-up). The steady state tests consisted of pre-heating the enclosure to a temperature level of approximately 220 °C before making the measurements.

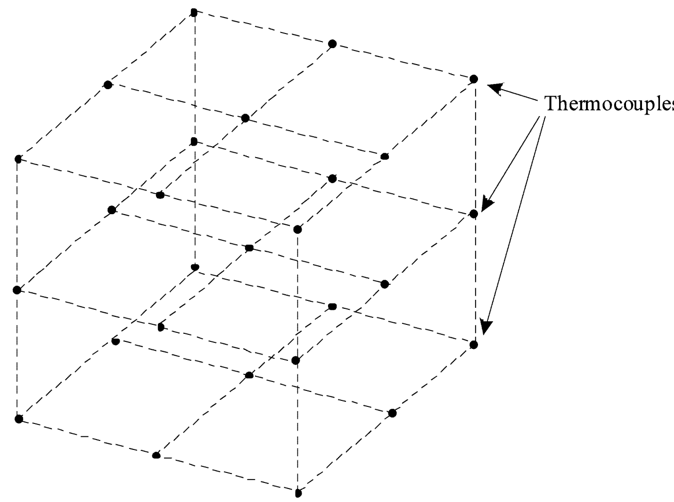

Fig. 4. Thermocouple locations.

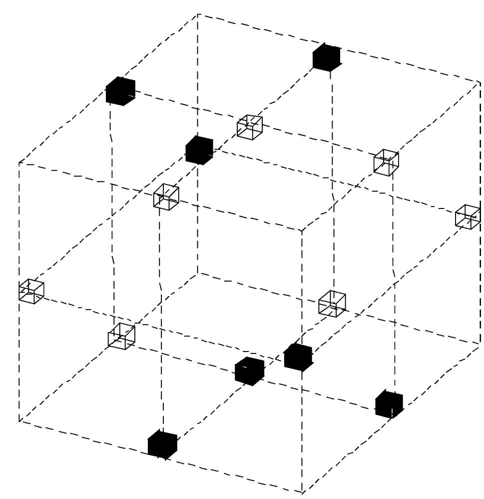

Fig. 5. Position of the blocks.

3.1. *Temperature distribution measurements*

The temperature distribution tests consisted of measuring the temperature in several points of a control volume in the form of a parallelepiped with dimensions $305 \times 240 \times 170$ mm as well as at the geometric centers of the enclosure walls. The temperatures inside the control volume were measured with the thermocouples fixed in a 3 mm diameter steel wire rig (see Fig. 4). The thermocouples were placed in 3 sections located at the bottom, middle and top of the control volume. Each section had 9 thermocouples, according to Fig. 4.

3.2. *Heat transfer coefficient measurements*

As described in Milanez and Mantelli [6], the heat transfer distribution inside the enclosure is measured by means of a steel wire rig containing $15 \times 15 \times 15$ mm aluminum blocks spread inside the enclosure, according to Fig. 5. Some of the blocks were painted in black while others were polished. By measuring the temperature of the blocks as a function of time, one can obtain the heat transfer coefficient h [W·m^{-2}·K^{-1}] between the enclosure and the blocks by means of the following expression:

$$mc_p \frac{\Delta T}{\Delta t} = hA(T_{\text{air}} - T) \tag{1}$$

where m [kg] is the block mass, c_p [J·kg^{-1}·°C^{-1}] is the specific heat at constant pressure, T_{air} [°C] is the air temperature, measured with thermocouples positioned near the blocks, T [°C] is the block temperature and A [m^2] is the block surface area. The Biot number for the block is less than 10^{-3}, which means it can be assumed to be at a uniform temperature.

The thermocouples used are #40 and type K. The time interval between two temperature readings is $\Delta t = 5$ s and ΔT [°C] is the increase of the block temperature between two consecutive readings. Every 5 seconds the data acquisition system reads the temperatures of the blocks and stores the data in a personal computer. Afterwards, the heat transfer coefficients at each time interval were obtained by solving Eq. (1) for h.

The procedure adopted for the transient tests consisted of turning on the gas burner with the enclosure at room temperature and with the rig containing the aluminum blocks placed inside the enclosure. The procedure adopted for the steady state tests consisted of pre-heating the enclosure to 220 °C and, with the temperature stabilized, the rig with the aluminum blocks was inserted into the enclosure. In both the transient and the steady state measurements, the tests were finished when the block temperature stopped varying with time, because according to Eq. (1), the value of h can only be calculated when there is a variation of the block temperature with time.

The objective of testing polished blocks and black blocks is to measure the radiative and the convective contributions of the heat transfer coefficients. Due to the low absorptivity, the polished aluminum blocks are practically subjected only to convective heat transfer. On the other hand, the blocks painted in black absorb heat both by radiation and by convection. Assuming that the effects of convection and radiation are additive, the radiation heat transfer coefficient can be obtained by subtracting the h value of the polished blocks from the value of black blocks. Therefore, in this work the radiation heat transfer rate is defined as:

$$q_{\text{rad}} = h_{\text{rad}} A(T_{\text{air}} - T) \tag{2}$$

The rate of heat transfer by radiation is not actually proportional to the difference between the temperatures of the air and of the block, as expressed by Eq. (2). The rate of radiation heat transfer is proportional to the difference between the fourth powers of the absolute temperatures of the walls and the blocks. Since the dimensions of the aluminum blocks are much smaller than the dimensions of the cooking chamber, and assuming also that the surface is diffuse and gray, the following relation can be used to estimate the radiation heat transfer rate between the walls at temperature T_w and the block at temperature T:

$$q_{\text{rad}} = \varepsilon\sigma A\left(T_w^4 - T^4\right) \tag{3}$$

where ε is the block surface emissivity and $\sigma = 5.67 \times 10^{-8}$ W·m^{-2}·K^{-4} is the Stefan–Boltzmann constant. From Eqs. (2) and (3) one gets:

$$h_{\text{rad}} = \frac{\varepsilon\sigma(T_w^4 - T^4)}{(T_{\text{air}} - T)} \tag{4}$$

Therefore, the radiation heat transfer coefficient obtained using this procedure is not constant with time because the temperature of the block varies from room temperature (beginning of the test) to the steady state temperature (approximately 220 °C). This result will be helpful in the analysis of the experimental data that follows later.

4. Results and discussion

4.1. Temperature distribution results

Figs. 6 and 7 show the measured three-dimensional temperature distributions during start up (transient test). Fig. 6 is the enclosure heated using the conventional approach, i.e., hot exhaustion gases flowing into the enclosure and Fig. 7 corresponds to the thermosyphon assisted enclosure. The tempera-

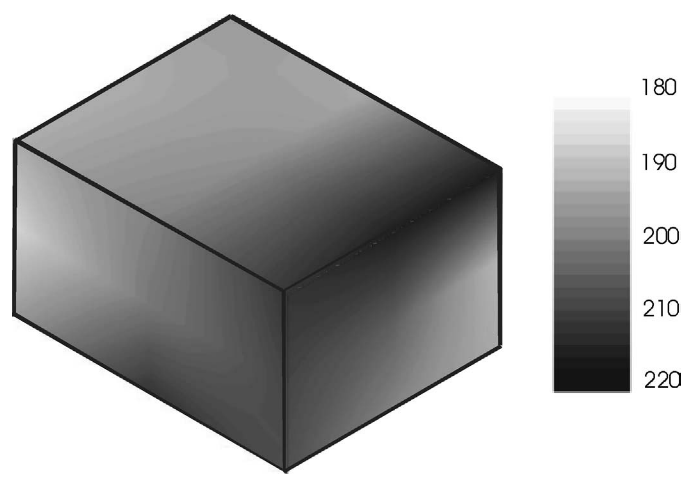

Fig. 6. Temperature map inside the enclosure heated using the conventional approach.

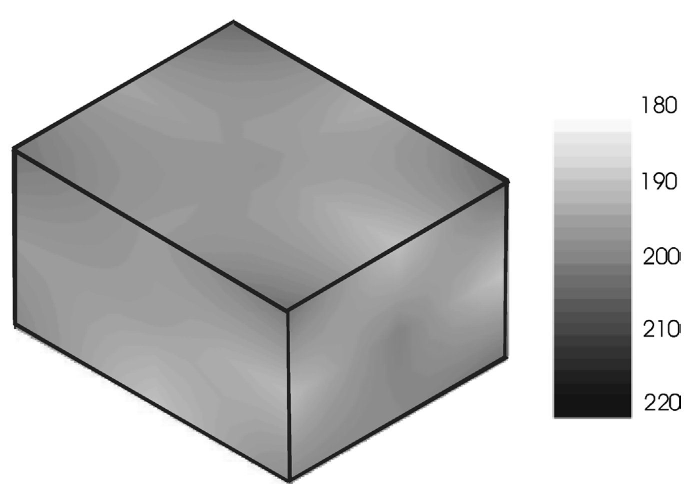

Fig. 7. Temperature map inside the thermosyphon assisted enclosure.

ture maps correspond to the time instant when the geometric center of the enclosure reaches 200 °C during start-up from room temperature. Linear interpolation was used to calculate the temperatures between two consecutive thermocouples of the rig shown in Fig. 4. The maximum temperature variation inside the control volume is 40 °C for the conventional enclosure, while for the enclosure heated by thermosyphon, the maximum temperature difference is only 8 °C. Two hot regions can be clearly observed in the conventional approach: the center of the front-lower edge and the right-upper edge of the control volume. The lower-front edge corresponds to the exhaustion gases exiting from the burner.

Milanez and Mantelli [6] present the temperature distribution measurement results in more details, including the tests under steady state, which resulted in a maximum temperature difference of 27 °C for the conventional enclosure and 7 °C for the thermosyphon assisted enclosure.

Figs. 8 and 9 show the temperature of the geometric centers of the enclosure walls as a function of time during start-up for the two enclosures tested. The temperatures of the air at some locations inside the enclosure are also presented in these figures for comparison purposes. As one can see on Fig. 8, which corresponds to the conventional enclosure, the temperature of the bottom wall reach very large values (beyond 400 °C) while the

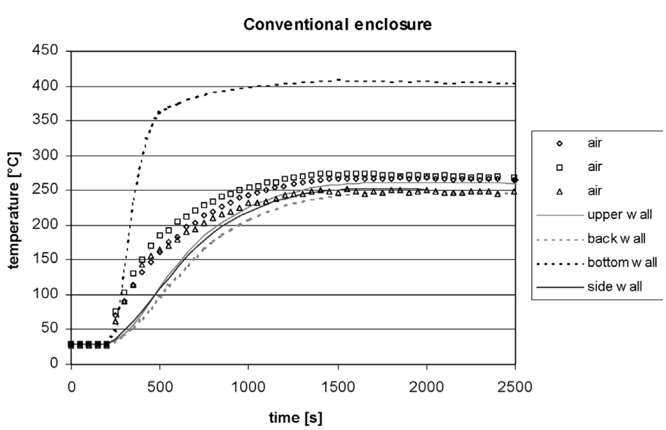

Fig. 8. Measured temperature variation with time during start-up for the conventional enclosure.

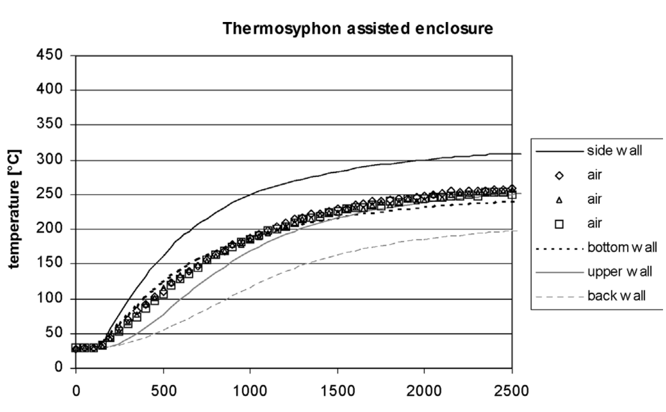

Fig. 9. Temperature variation with time during start-up for the thermosyphon assisted enclosure.

temperatures of the air and of the other walls reach steady state at approximately the same level (250 °C). The bottom wall temperature is much hotter than the rest of the enclosure because of the gas burner, which is placed underneath. As for the enclosure assisted by thermosyphons, shown in Fig. 9, the hotter elements are the side walls, which are in contact with the thermosyphon condensers. One can also perceive that the maximum temperature difference between the walls of the conventional enclosure is around 150 °C, while for the thermosyphon assisted enclosure the difference is around 100 °C.

4.2. Heat transfer coefficient results

Table 1 presents the average of the measured values of the heat transfer coefficients for all cases tested. As one can see, the average heat transfer coefficient between the polished bocks and the enclosure, which is controlled primarily by convection, for the conventional approach is larger than for the enclosure with thermosyphons. This is because the air temperature distribution of the enclosure assisted by thermosyphons is more uniform than the conventional enclosure, as presented in the last section. The more uniform is the temperature distribution of the air inside the enclosure, the less intense is the natural convection induced air flows inside the enclosure and, as a consequence, the smaller is the convective heat transfer coefficient. The convection heat transfer coefficients are larger during transient than under steady state due to the same reason. During transient, the air temperature gradients are larger than during steady state, as mentioned previously. The air flow due to natural convection is more intense and consequently the convection heat transfer coefficients during transient are larger than under steady state.

Yovanovich's correlation for external natural convection around a cube, with the dimensions of the aluminum blocks used in this study, in an environment with stagnant air [7] yield a value of 13.5 $W\cdot m^{-2}\cdot K^{-1}$, which agrees quite well with the average of the measured value for the thermosyphon assisted enclosure under steady state condition (14.6 $W\cdot m^{-2}\cdot K^{-1}$). It can be concluded that the air inside the thermosyphon assisted enclosure under steady state is predominantly stagnant. As for the conventional enclosure, air movement induced by the flow of the exhaustion gases lead to larger values of convection heat transfer coefficients (17.1 $W\cdot m^{-2}\cdot K^{-1}$) than predicted by Yovanovich's correlation (stagnant air). Both enclosures presented similar convection heat transfer coefficients during start-up (18.9–19.0 $W\cdot m^{-2}\cdot K^{-1}$) and are larger than the stagnant air value obtained by Yovanovich's correlation. This result

Table 1
Average values of the measured heat transfer coefficients [$W\cdot m^{-2}\cdot K^{-1}$]

Test		Polished	Black	Black-polished
Thermosyphon assisted	steady state	14.6	28.2	13.6
	transient	18.9	27.2	8.3
Conventional approach	steady state	17.1	36.2	19.1
	transient	19.0	29.4	10.4

once more shows that the natural convection is more intense during start-up.

Regarding to the radiation heat transfer coefficients, which are given by the difference between the values of the black and the polished surfaces, the conventional enclosure presents larger mean values, as it can be verified in the last column of Table 1. This happens because in the conventional approach, the floor of the enclosure reaches very high temperatures (above 400 °C) due to the vicinity of the flames of the gas burner placed underneath. This fact can be clearly observed in Fig. 10, which presents the heat transfer coefficient distribution between the blocks and the enclosures during steady state. As one can see, the heat transfer coefficient measured from the black blocks placed close to the floor of the conventional enclosure are much larger than the rest. As for the enclosure with thermosyphons, the heat transfer coefficients of the black blocks present a more uniform distribution. In the thermosyphon assisted enclosure, the flames of the combustion chamber are not in contact with the walls of the enclosure. The heat generated in the combustion chambers is spread over the lateral walls of the enclosure through the thermosyphons. As a result, the radiation field inside the enclosure assisted by thermosyphons presents a more uniform distribution than the enclosure heated with the conventional approach.

As already mentioned, the convection heat transfer coefficients present larger values during transient than during steady state because the air temperature distribution is more uniform during steady state than during transient, and larger temperature gradients lead to more intense natural convection. On the other hand, the average radiation heat transfer coefficient, which is calculated from the difference between the average of the black blocks and the polished blocks, present larger values during steady state in both enclosures tested, as it can be observed from the data of the last column of Table 1. This behavior can easily be explained under the light of Eq. (4), which shows that the radiation coefficient is strongly affected by the temperature of the enclosure walls. In steady state, the temperatures of the walls are approximately at the same level of the air (see Figs. 8 and 9) and remain constant with time. As for the transient test (start-up), the walls are initially at room temperature. Initially, the walls are cold and therefore the radiation coefficient is small. As the wall temperatures increase with time, the radiation heat transfer coefficient also increases. Fig. 11 (a) presents a graph of the heat transfer coefficient, calculated form Eq. (1),

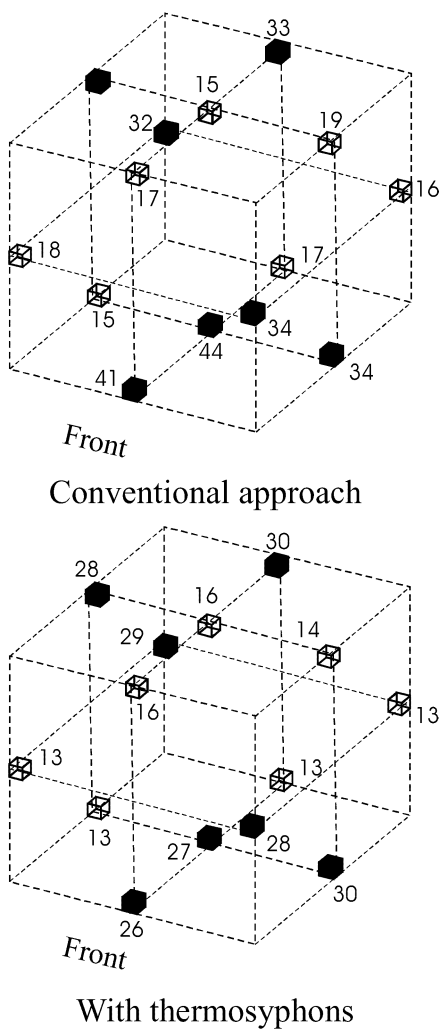

Fig. 10. Distribution of heat transfer coefficients inside the enclosure under steady state [W·m^{-2}·K^{-1}].

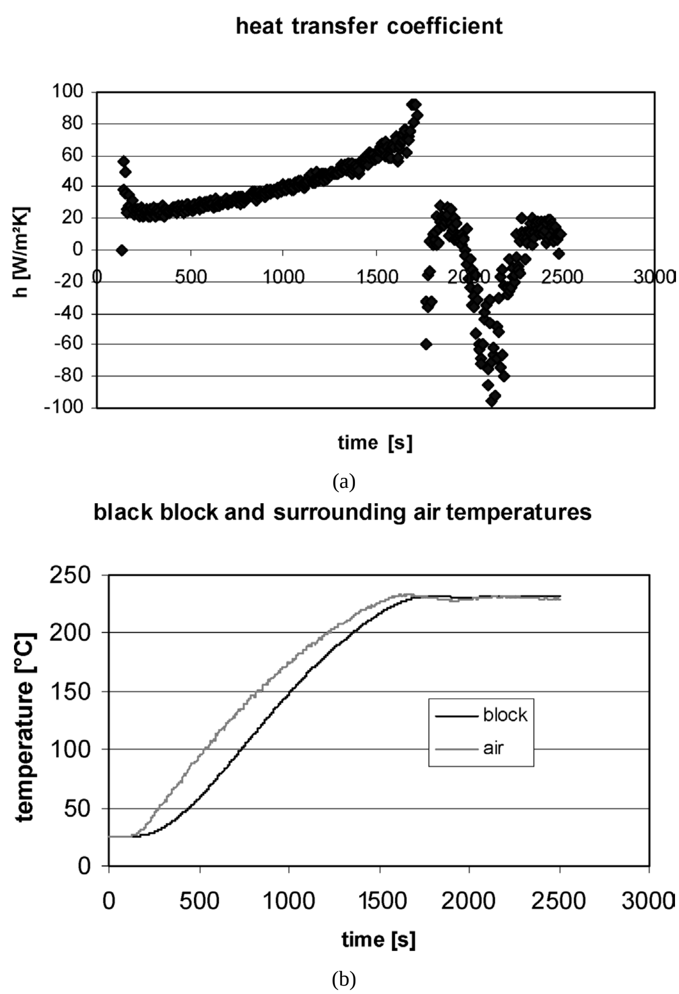

Fig. 11. Measured heat transfer coefficient versus time (a). Measured temperatures of a typical black block and surrounding air during transient (b).

of a typical black block as a function of time, which illustrates this effect. Fig. 11(b) shows the temperatures of the block and the surrounding air as a function of time during transient. It can be observed that, initially, the temperatures of the cube and of the air are close to each other. The oscillations of temperature readings due to the uncertainty of the thermocouples make the first few h data points of Fig. 11(a) to present a large variation. These first few points should be ignored as they bear a large experimental error. A few seconds later, the values stabilize around 25 $\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$ and then start to go up smoothly with time as the temperatures of the walls increase. The values presented for the black blocks on Table 1 correspond to the beginning of the test, when the walls are still cold. By the time $t = 1600$ seconds, approximately, the temperatures of the block and of the surrounding air get close to each other again, which leads to a large scattering of h values once again. Even negative values are calculated because the black block eventually reaches temperatures higher than the air due to intense radiation absorption. These final points should also be ignored because they do not correspond to the real physics of the problem.

The heat transfer coefficient between the enclosure and the polished aluminum blocks, on the other hand, which is controlled by convection, is almost constant with time. From the classical theory of convection heat transfer, one should expect a slight variation of the heat transfer coefficient with time, as the air thermal properties and the Grashoff number, which is proportional to the block temperature, vary with time. However, this variation is small under the conditions tested, so that the convection heat transfer coefficient between the air the blocks is practically constant with time.

5. Summary and conclusions

This work analyzes and compares the thermal characteristics of two enclosures with the same dimensions and similar constructive forms, but that are heated using distinct methods: one is heated using two-phase thermosyphons and the other is heated with a more conventional approach, which uses hot exhaustion gases flowing into the enclosure. The first type has been evaluated for application to baking ovens, while the second type is already commonly used for this application in some countries. Both transient and steady state conditions were tested. Temperature distributions inside the enclosures are also measured in order to help the analysis of the results.

The results show that the enclosure heated using thermosyphons have more uniform temperature and radiative heat transfer coefficient distributions in both steady state and transient conditions. The convective heat transfer coefficients are uniform in all cases tested, but are larger during transient than during steady state because of the larger temperature gradients obtained, which makes natural convection more intense. The conventional enclosure presents larger convective heat transfer coefficients than the thermosyphon assisted enclosure because of the exhaustion gases movement. The thermosyphon assisted enclosure present a very uniform temperature distribution, which makes the air stagnant and the heat transfer coefficients small. This aspect could pose a problem when applying the thermosyphon assisted enclosure to baking ovens, because small heat transfer coefficient means larger cooking time and higher fuel consumption. However, a small fan placed inside the enclosure can solve this problem by artificially inducing the necessary air movement.

The radiation field inside the thermosyphon assisted enclosure is more uniform than the conventional enclosure. The conventional enclosure has the bottom wall very hot because of the gas burner, which is placed underneath. As a consequence, the thermal radiation is much more intense from the bottom than from the remaining walls. The thermosyphons help spreading the heat over a larger area of the side walls. A more uniform radiation field is obviously better when applying the enclosure for cooking as it makes the process more even.

The methodology used here has proven to be very effective in assessing the thermal performance of heated enclosures. It can be used to predict and compare the cooking characteristics of prototype ovens in laboratory before submitting the final prototype to tests with actual food. In the near future, the methodology is gong to be employed in the development of ovens for pizza and cookies.

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