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A NEW METHODOLOGY FOR MEASURING HEAT TRANSFER COEFFICIENTS-APPLICATION TO THERMOSYPHON HEATED ENCLOSURES

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ABSTRACT

A new experimental method is developed in this work in order to measure heat transfer coefficients. It consists basically of measuring the variation of the temperature of small aluminum blocks with time. Both convective and radiative heat transfer coefficients can be obtained by using polished and black blocks, respectively. The method is successfully applied here to obtain heat transfer coefficient distributions inside enclosures. Two enclosures were tested: one was heated using two-phase thermosyphons and the other was heated by hot exhaustion gases. The results show that the enclosure heated using thermosyphons presents more uniform heat transfer coefficient distributions.

KEY WORDS: Heat transfer coefficient measurement, Two-phase thermosyphon application.

1. INTRODUCTION

Two-phase thermosyphons are high efficiency heat transfer devices. Apart from featuring a very low thermal resistance, another important characteristic of two-phase thermosyphons is a very uniform temperature distribution in the condenser section when the external heat transfer coefficient is small. Recently, Mantelli and co-workers (Mantelli et al. 1999, 2003 and da Silva & Mantelli, 2004), successfully applied two-phase thermosyphons to isothermalize enclosures, such as bakery ovens. The enclosure is heated by thermosyphons attached to the side walls. The thermosyphon condensers are inside the enclosure, while the evaporators are confined in a combustion chamber placed below the enclosure. Heat is supplied by gas (propane-butane) burners inside the combustion chambers. The thermosyphons transfer the heat from the combustion chamber into the enclosure without mixing the exhaustion gases with the air inside the enclosure.

The usual approach to heat bakery ovens is to use 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. The presence of exhaustion gases inside the cavity may also be undesirable when using this approach for baking, for example. Furthermore, intense thermal radiation from the bottom wall, which is close to the gas burner and can reach temperatures beyond 400°C, makes the radiative heat flux distribution inside the cavity very uneven. The closer to the bottom wall, the more intense is the thermal radiation. This is also an inconvenience when applying the enclosure for baking.

A new experimental test method was developed in order to compare the thermal aspects of the two types of enclosures mentioned above, i. e., the enclosure heated by thermosyphons and the conventional enclosure heated by hot exhaustion gases. This methodology is presented here, and consists basically of measuring the heat transfer coefficient distributions inside the enclosures. The application of the method is not restricted to enclosures, but can also be used in any other applications when one needs to measure heat transfer coefficients.

2. PROBLEM GEOMETRY

The geometry of the enclosure under study is presented in Fig. 1. It is composed of two mild steel

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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 x 0.48 x 0.61 m. Eight thermosyphons are attached internally to 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). An 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 glass wool 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%. A gas burner is placed bellow each row of evaporators. The evaporators and the burner are confined in a combustion chamber, completely separated from the cavity.

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 "L" shaped plate. A steel wire cramp is used to squeeze the fin against the thermosyphon. The function of the "L" shaped plate is to distribute the contact pressure more evenly over the interface, which avoids the appearance of 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. Under compression, the aluminum tape deforms easily, helping to fill the gaps between the thermosyphon and the fin, also contributing to decrease the thermal contact resistance.

3. EXPERIMENTAL PROGRAM

The experimental study consists basically of measuring temperature and heat transfer coefficient distributions inside the enclosure. The temperature distributions are also measured in order to help the data analysis. Two different enclosures were tested: one heated using the conventional approach, i.e., hot exhaustion gases flowing into the cavity, and the other using thermosyphons, as described in the last section.

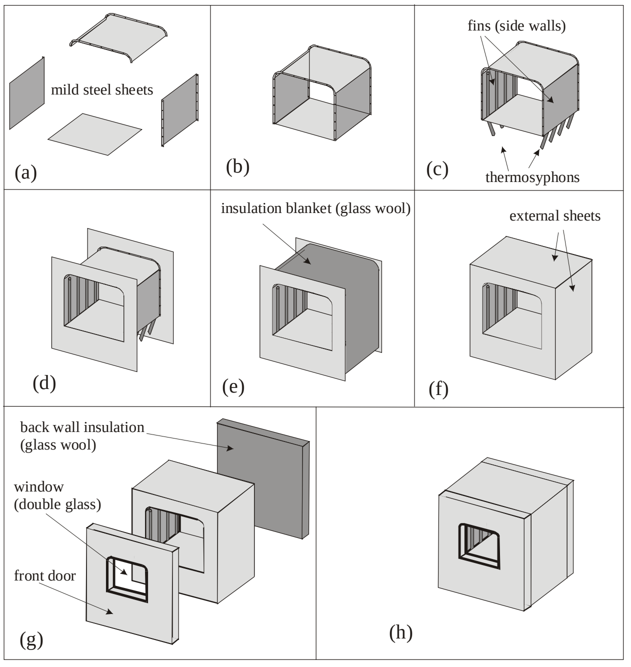


Figure 1. Enclosure geometry.

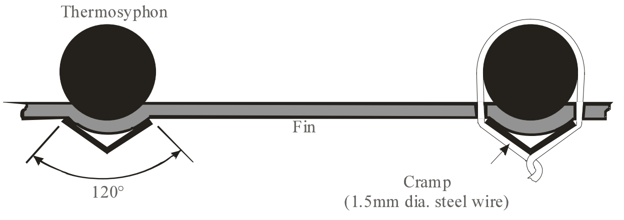


Figure 2. Details of the thermosyphons/fin attachment.

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

3.1. Temperature Distribution Measurements

The temperature distribution tests consisted of measuring the temperature in several points in a control volume in the form of a parallelepiped of dimensions 305 x 240 x 170 mm located in the center of the enclosure. The temperatures were measured with 27 type T thermocouples fixed in a 3 mm diameter steel wire rig. 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. 3.

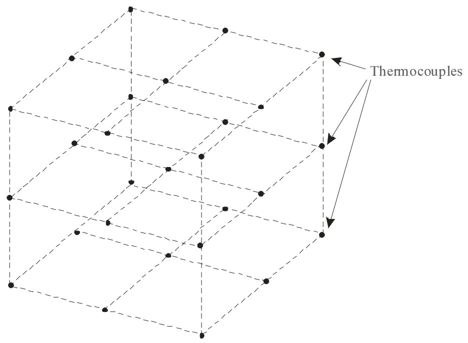


Figure 3. Thermocouple locations

3.2. Heat Transfer Coefficient Measurements

A special apparatus was designed and built in order to measure the heat flux distribution inside the cavity. It consists of a rig with $15 \times 15 \times 15\,\text{mm}$ aluminum blocks spread inside the cavity (see Fig. 4). Some of the blocks were painted in black while others were left polished. By measuring the temperature of the blocks with time, one can obtain the heat transfer coefficient h [W/m²K] between the enclosure and the blocks by means of the following expression:

$$mc_p \frac{\Delta T}{\Delta t} = h\,A\left(T_{air} - T\right) \qquad (1)$$

where m [kg] is the block mass, c_p [J/kg°C] is the specific heat at constant pressure, T_{air} [°C] is the air temperature inside the enclosure, T [°C] is the block temperature and A [m²] is the block surface area. The time interval between two temperature readings is $\Delta t = 5\text{s}$ 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 a stores the data in a personal computer file. Afterwards, the heat transfer coefficients at each time interval are obtained by solving Eq. (1) for h.

The objective of testing polished blocks and black blocks is to measure the radiation and convective heat transfer coefficients. Given the low absortivity, the polished aluminum blocks are practically subjected to convective heat transfer only. On the other hand, the blocks painted in black absorb heat 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. The radiation heat transfer coefficient is defined as:

$$q_{rad} \equiv h_{rad}\,A\left(T_{air} - T\right) \qquad (2)$$

It is interesting to bring into attention some limitations of this test, specially regarding to the radiation heat transfer coefficient. First, the rate of heat transfer by radiation is not proportional to the difference between the temperatures of the air and of the plate, as in 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 exchange between the walls at temperature T_w and the plate at temperature T:

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

where ε is the surface emissivity and σ=5.67x10^{-8} W/m²K^4 is the Stefan-Boltzmann constant. From Eqs. (2) and (3) one gets:

$$h_{rad} = \frac{\varepsilon\,\sigma\left(T_w^4 - T^4\right)}{\left(T_{air} - T\right)} \qquad (4)$$

Therefore, the radiation heat transfer coefficient obtained using this procedure is not constant, even assuming the temperature difference between the air and the walls as constant. This result will be helpful in the analysis of the experimental data that follow later.

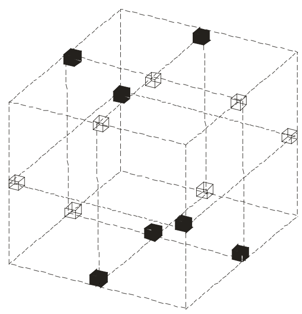


Figure 4. Position of the flux-meters

4. RESULTS AND DISCUSSION

4.1. Temperature Distribution Results

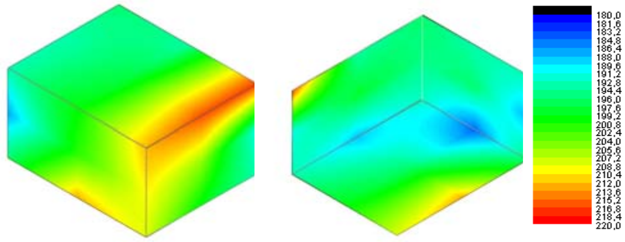
Figure 5 shows the three-dimensional temperature distribution measured from the enclosure heated using the conventional approach, i.e., hot exhaustion gases going into the enclosure. The temperature map presented corresponds to the time instant when the geometric center of the enclosure reaches 200°C during start-up from room temperature. Linear interpolation was used to calculate the temperatures between two consecutive thermocouples of the temperature rig shown in Fig. 3. Figure 5 part (a) is the top-front-left view of the temperature map and part (b) is the bottom-back-right view. The maximum temperature variation inside the control volume is 40°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.

The temperature maps of the thermosyphon assisted enclosure are shown in Fig. 6. The temperature fields inside the control volume are very uniform, with a maximum temperature variation of only 8°C.

Figures 7 and 8 show the steady state temperature maps inside the enclosure heated using the conventional approach and the enclosure heated by thermosyphons, respectively. The maximum temperature difference in the conventional approach is 27°C. For the thermosyphon assisted enclosure, the maximum temperature difference is 7°C. Once more, the enclosure with thermosyphons presents a much more uniform temperature distribution. It can also be observed that the temperature differences during transient (Figs. 5 and 6) are larger than during steady state (Figs. 7 and 8).

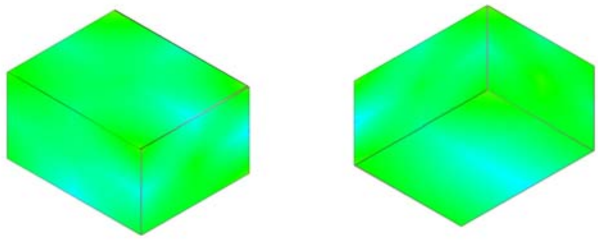
4.2. Heat Transfer Coefficient Results

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 of the polished bocks, which is primarily convection controlled, inside the enclosure heated using the conventional approach is larger than in the enclosure with thermosyphons. This is because the temperature distribution of the enclosure assisted by thermosyphons is more uniform, as presented in the last section. The more uniform is the temperature distribution, 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.



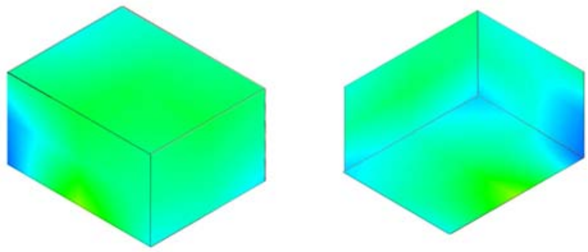
(a) (b)

Figure 5. Temperature map inside the enclosure heated using the conventional approach during start-up.



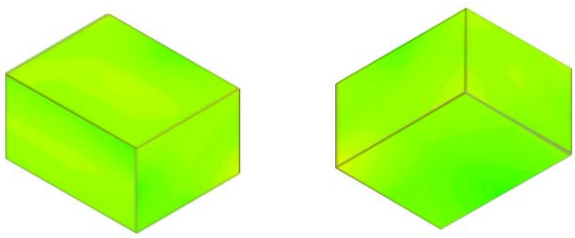
(a) (b)

Figure 6. Temperature map inside the thermosyphon assisted enclosure during start-up.



(a) (b)

Figure 7. Temperature map inside the enclosure heated using the conventional approach during steady state.



(a) (b)

Figure 8. Temperature map inside the thermosyphon assisted enclosure during steady state.

The convection heat transfer coefficients are larger during transient than steady state. Again, during transient the air temperature gradients are larger than during steady state, as already noticed in the temperature maps (Figs. 5 to 8). The natural convection flows inside the enclosure increase the air speed and consequently increase the convection heat transfer coefficients during transient.

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 (Bejan, 1995) yield a value of 13.5 W/m²K, which agrees very well with the average of the measured value (14.6 W/m²K). It can be concluded that the dimensions of the enclosure are large enough for one to consider that the heat transfer between the air and the blocks is not affected by the limits of the enclosure and that the air inside the cavity 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 than stagnant air (thermosyphons), as it can be observed in Table 1.

Table 1. Average values of the measured heat transfer coefficients [W/m²K]

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

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 Tab. 1. This happens because in the conventional approach, the floor of the enclosure reach very high temperatures due to the vicinity of the flames below it (above 400°C). This fact can be clearly observed in Fig. 9, which presents the heat transfer coefficient distribution inside both the enclosures tested at steady state. As one can see, the heat transfer coefficient obtained from 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. This is because the flames of the combustion chamber are not in contact with the walls of the enclosure. The heat generated in the combustion chamber 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 using the conventional approach.

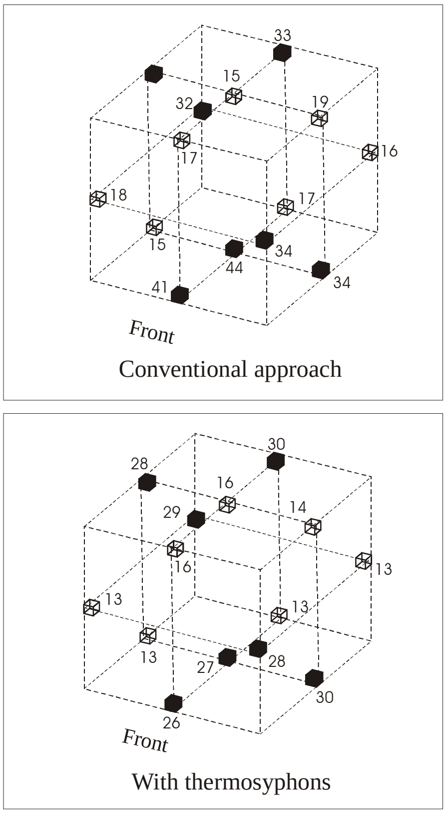


Figure 9. Distribution of heat transfer coefficients inside the enclosure at steady state [W/m²K]

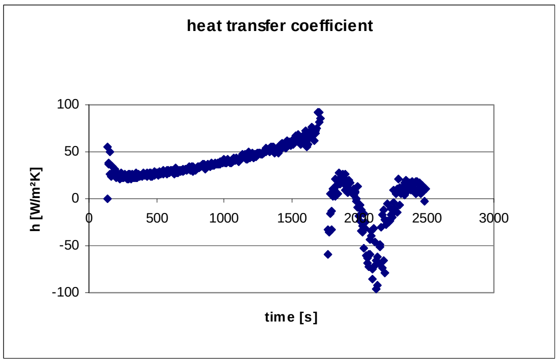
As already mentioned, the convection heat transfer coefficients present larger values during transient than during steady state due to the fact that the air temperature distribution is more uniform during steady state than during transient and larger temperature gradients originate more effective natural convection. On the other hand, the radiation heat transfer coefficients present larger values during steady state, as it can be observed from the data of Table 1. This behavior can easily be explained in the light of Eq. (4), which shows that the radiation coefficient is strongly affected by the temperature of the cooking chamber walls. In steady state, the walls temperature are approximately at the same level of the air, i. e., 220°C. As for the transient test (start-up), the walls temperatures are initially at room temperature.

The cooking chamber walls are cold and therefore the radiation coefficient is small. As the wall temperatures increase, the radiation heat transfer coefficient increases too. Figure 10 (a) presents a graph of the heat transfer coefficient of a typical black block as a function of time, which illustrates this effect. The experimental values of *h* shown in this graph are calculated according to Eq. (1). Figure 10 (b) shows the temperatures of the block (cube) 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. 10 (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 and start to go up smoothly with time as the temperatures of the walls increase. At 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 eventually the black cube 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.

5. SUMMARY AND CONCLUSIONS

A novel experimental method was developed here in order to obtain heat transfer coefficients. The method was successfully employed to measure the heat transfer coefficient distributions inside two enclosures: one heated using two-phase thermosyphons and one employing a more conventional heating approach, which uses hot exhaustion gases flowing into the enclosure. Both transient and steady state conditions were tested. Temperature distributions inside the enclosures are also measured in order help the analysis of results. The results show that the enclosure heated using thermosyphons have a more uniform temperature and radiative heat transfer coefficient distributions. The convective heat transfer coefficient are uniform in the two cases tested. The convective heat transfer coefficient is larger during transient than during steady state because of the larger temperature gradients induce more effective natural convection air flows. The conventional enclosure tends to present larger convective heat transfer than the thermosyphon assisted enclosure because of the exhaustion gases movement inside the conventional enclosure.

(a)



(b)

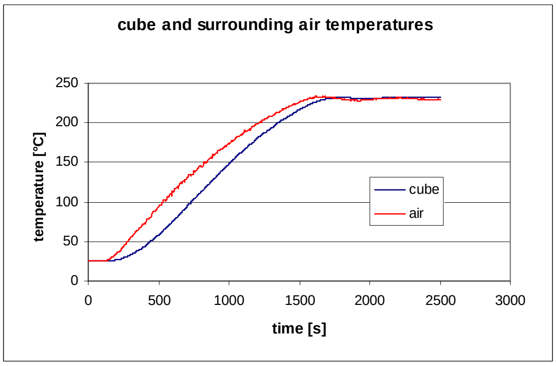


Figure 10. Heat transfer coefficient versus time. Measurement of a typical black block during start-up.

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