# THERMAL AND FLUID DYNAMICS ANALYSIS OF DOMESTIC GAS OVENS

**Louise N. Bätzner, louise@labcet.ufsc.br**
**Donato G. Nascimento, donato@labcet.ufsc.br**
**Rafael C. Catapan, catapan@labcet.ufsc.br**
**Helena N. Fomentin, helena@labcet.ufsc.br**
**Amir A. M. Oliveira, amir.oliveira@gmail.com**
Laboratory of Combustion and Thermal Systems Engineering
Department of Mechanical Engineering, Federal University of Santa Catarina, +55 (48) 3721 9390
Campus Universitário João David Ferreira Lima, Florianópolis - SC – Brazil, 88040-900

**Fernando M. Pereira, fernando@mecanica.ufrgs.br**
Department of Mechanical Engineering, Federal University of Rio Grande do Sul, +55 (51) 3308 3114
R. Sarmento Leite 425, Porto Alegre - RS – Brazil, 90050-170

**William A. Ibanez, william_a_ibanez@whirlpool.com**
**Andre A. Lopes, andre_a_lopes@whirlpool.com**
**Pablo F. Rodeiro, pablo_f_rodeiro@whirlpool.com**
Simulation Based Design – Whirlpool Latin America
R. Dona Francisca, 7.200 - Distrito Industrial, Joinville – SC – Brazil, 89219-900

***Abstract.*** *The present work proposes a methodology for the thermal and fluid dynamic analysis of domestic gas ovens for model validation. The methodology is based on the temperature mapping and velocity measurements in the oven cavity. A global one-dimensional model is developed in order to estimate the surface temperatures and thermal resistances of the walls. The proposed experimental methodology is able to provide the required information to the validation of simulation tools for oven design. Conditions that lead to incomplete combustion are identified. The bottom plate of the oven cavity is a candidate for design optimization, since the major part of the heat losses to the surroundings occurs in this region. The one dimensional model is able to predict simple features about the thermal behavior of the oven. Differences as high as 60% between predicted and measured surface temperatures were found. The largest difference in the measured and estimated global resistances is observed in the door. For the other walls, the differences are less than 15%. The main source of error is probably the estimative of the conduction heat flux across the glass.*

***Keywords****: experimental analysis; model validation; thermal resistances; domestic gas oven.*

## 1. INTRODUCTION

The modeling of the thermal behavior of domestic ovens can be used as a tool for design and optimization. Thermal models allow for extensive parametric studies, avoiding the higher costs of prototype construction and testing. However, such models require a validation procedure based on experimental data to assure their accuracy.

Little attention has been addressed in the literature for the development of a systematic parametric study of the domestic or small scale ovens. Navaneethakrishnan et al. [2007] analyzed the flow and temperature distribution in a small scale bakery oven by using a two dimensional steady state model. They show that an optimal configuration for thermal homogeneity and efficiency can be obtained by combining a heat source at the bottom plate of the oven cavity and a guidance for better flow distribution. The issue of the thermal homogeneity was also addressed in Silva and Mantelli [2004] by studying the inclusion of high capacity conducting devices in the oven cavity. Other studies addressed the coupling between radiation and natural convection in enclosure cavities [Wang et al., 2006; Kuznetso and Sheremet, 2009].

The present work proposes a methodology for the thermal and fluid dynamic analysis of domestic gas ovens for model validation. The methodology is based on the temperature mapping and velocity measurements in the oven cavity. A global one-dimensional model is developed in order to estimate the surface temperatures and thermal resistances of the walls.

## 2. EXPERIMENTAL

### 2.1 Experimental apparatus

Figure 1 presents the experimental apparatus used in this work.

Figure 1. Experimental apparatus.

Preliminary tests were carried out in the oven to identify possible leak points in the wall junctions of the cavity. Hot smoke was produced inside the cavity for testing the leak in operating and non operating conditions. All leak points were sealed with appropriated hot temperature resistant tape. The oven was then reassembled with the necessary reinforcement to guarantee the structure, allowing the reproducibility of the data during all period of the tests. New preliminary tests indicated the difference between inlet and outlet volumetric flow rate in the cavity is less that 10%, which is quite acceptable based on the experimental techniques for measurements and on the apparatus built from a commercial product.

Type K and type E thermocouples (250 μm diameter lead wires - Omega) were used to measure the temperatures in 60 points at the multi-layer oven walls as well as in the inlet and outlet orifices of the cavity and the combustion chamber. The temperature data were recorded by a data acquisition system (34970A - Agilent) interfaced to a personal computer. Although radiation effects cannot be totally neglected in the measurements of the flow temperature (at inlet and outlet orifices) preliminary tests have indicated that such effects have a small contribution to the temperatures measured in the experimental conditions applied here. We could not distinguish such effect from the experimental error that we found.

Surface temperatures were measured by an infrared camera (SC500 - FLIR). Preliminary tests were made to define the surface total emissivity, taking into account environment reflection. This was carried out by placing thermocouples on the surface and correcting the emissivity until the temperatures predicted by the infrared camera fit the values predicted by the thermocouples. This procedure produces a total emissivity that is a sum of the actual surface emissivity and reflection on the surface.

The flow velocity was measured with a vane anemometer (445 - 0635.9540 - Testo) at the inlet and outlet orifices of the cavity. The fuel used was butane (95% pure) and was controlled by a needle valve and measured by an electronic mass flow meter (1820 - Omega). The fuel pressure was kept constant at 280 kPa, as recommended by the manufacturer, making sure that the data were measured in operating actual conditions. The total power used in the tests (see Table 1 below) were chosen in order to allow a significant variation in the experimental conditions, but at the same time to protect the vane anemometer against contact with high temperature flow which can produce untrue data.

### 2.2 Experimental procedure and uncertainties

Table 1 presents the experimental conditions tested in this work. The parameters chosen for the analysis were the total power and the number of the orifices opened in the cavity floor. Each condition is a combination of these two parameters, allowing the validation of the model in different situations. To allow experimental reproducibility, the measurements always took place in steady state conditions, monitored by thermocouples placed at the cavity's center and at the glass door center, at the inner part of the cavity.

Table 1. Experimental conditions (see Fig. 2).

| Condition | Opened orifices | Total Power, kW |
|---|---|---|
| BP+ GF+ | 1L, 2L, 4L, 1R, 2R, 4R | 3.5 |
| BP+ GF- | 1L, 2L, 4L, 1R, 2R, 4R | 2.6 |
| BP- GF+ | 3L, 5L, 3R, 5R | 3.5 |
| BP- GF- | 3L, 5L, 3R, 5R | 2.6 |

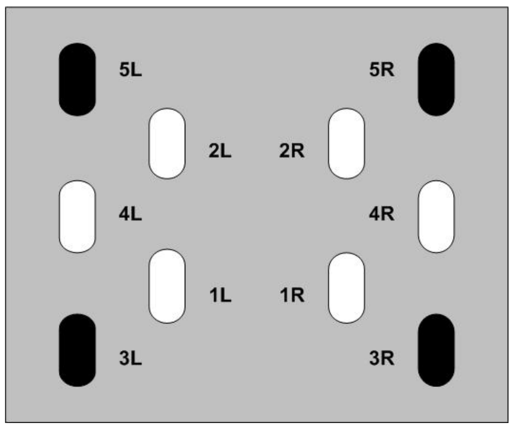



Figure 2. Bottom plate scheme with orifices 1, 2 and 4 opened (represented by the condition BP+).

All measurements were carried out in steady state conditions and each experimental point represents an average over three measurements. The tests were carried out during a period of 6 months, without any modification in the apparatus structure, which allow the reproducibility of the data as well as produce a representative experimental uncertainty of the wide range of data. Table 2 shows the experimental uncertainties calculated for each measured parameter.

Table 2. Experimental uncertainties.

| Flow temperature [ºC] | Surface temperature [ºC] | Bottom plate surface temperature [ºC] | Inlet velocity (at bottom plate orifices) [m/s] | Outlet velocity (at chimney) [m/s] |
|---|---|---|---|---|
| ±18 | ±20 | ±60 | ±0.30 | ±0.16 |

## 3. MODELING

### 3.1 One dimensional model

Initially, thermal resistances were estimated for each oven wall based on the experimental results. Theses resistances thermally connect the internal wall temperature Ti with the ambient temperature T∞ and can be used in numerical simulations as boundary conditions for the determination of the heat losses to the surroundings through the cavity walls.

The heat flux across the wall is estimated using the temperature measurements of the inner and outer surfaces of the insulation layer. The conduction heat flux across the insulation layer, $\dot{q}_{k,ins}$, is calculated as $\dot{q}_{k,ins} = k_{ins}\Delta T_{ins} / L_{ins}$, were $k_{ins}$ is the insulation thermal conductivity, $\Delta T_{ins}$ is the temperature difference between the inner and outer surfaces of the insulation and $L_{ins}$ is the insulation thickness. The total thermal resistance across the wall, $R_e$, is then estimated based on the temperature difference between the inner cavity surface temperature, $T_i$, and the ambient temperature, $T_{amb}$, as $R_e = (T_i - T_{amb}) / \dot{q}_{k,ins}$.

The same procedure was adapted to evaluate the total thermal resistance of the cavity door, that is composed by two glasses spaced by an air layer. On the other hand, the procedure could not be employed to evaluate the back wall, since it is composed by two metal layers without insulation, which makes difficult the evaluation of the heat flux.

Considering $k_{ins}$ = 0.031 W/mK and $k_{glass}$ = 0.8 W/mK and the temperature measurements for the oven cavity at a total power of 4.05 kW, the resulting total resistances are listed in Table 3.

Table 3. Total resistances of the cavity walls.

| Cavity Wall | R(W/K) |
|---|---|
| Top | 2.51 |
| Left | 2.31 |
| Right | 1.87 |
| Door | 0.02 |

A preliminary one-dimensional model of the cavity heat transfer is developed based on thermal resistances, taking into account one average temperature for each wall layer. Mixed convection (forced and natural) and radiation heat transfer are considered at the inner wall surfaces while only natural convection is imposed at the outer wall surfaces. Conduction and natural convection are modeled at the inner layers of the walls.

Figure 3 shows a schematic representation of a typical resistance model for the wall. The model considers the radiant heat transfer between the oven internal surfaces employing the radiosity method. The heat losses to the surroundings are modeled by the global heat transfer resistances determined experimentally ($R_4$ in Figure 3). The gas inside the cavity is modeled by a mean temperature, $T_m$. The thermal resistance $R_{ku,i,4}$ shown in the figure is a combined forced an natural convection resistance.

The cavity floor temperature is imposed based on an area averaged temperature obtained by infrared imaging. At the inlet orifices of the cavity floor, the gas mass flux is imposed with a prescribed temperature, both based on the experimental measurements. The total power of the oven is the sum of the heat flux at the cavity floor, the convective flux at the inlet orifices and the downward radiant heat losses of the cavity floor.

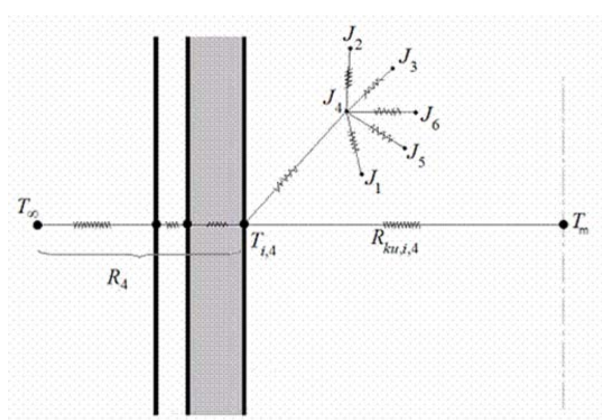


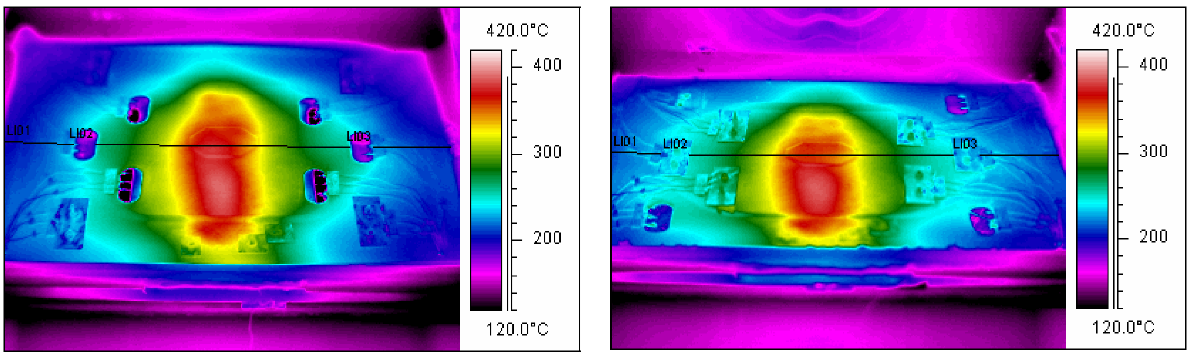

Figure 3. Wall resistance model.

# 4. RESULTS

## 4.1 Experimental

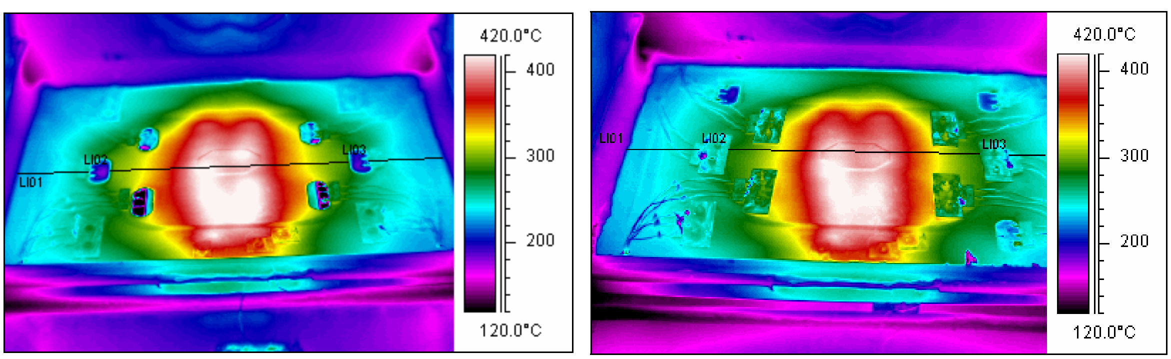
Figures 4 and 5 show infrared images of the bottom plate at BP+GF- (Fig. 4a), BP-GF- (Fig. 4b), BP+GF+ (Fig. 5a) and BP-GF+ (Fig. 5b). Experimental conditions are shown in the Table 1. Both images in Fig. 4 show a maximum temperature close to 390ºC at the center of the bottom plate while the maximum temperature exceeds 420ºC in both images in the Fig. 5. On the other hand, the temperature distribution along the bottom plate is qualitatively quite similar between images in the same total power, e.g. Fig. 4a and 4b.

Figure 6 shows the temperature distribution at the centerline of the bottom plate, as shown in Fig. 4 and 5, for the experimental conditions analyzed here. The lines blue and gray were taken for low total power (GF-) and the lines yellow and red for high total power (GF+). An increase in the total power makes the temperature peak at the position 0.0 broad, even generating two peaks of temperature. This is a result of the higher velocity jet at orifice of the burner in a higher total power, making the contact area for hot gases higher.




(a) (b)

Figure 4. Infrared image of the bottom plate at (a) BP+GF- and (b) BP-GF-. Experimental conditions are shown in the Table 1.




(a) (b)

Figure 5. Infrared image of the bottom plate at (a) BP+GF+ and (b) BP-GF+. Experimental conditions are shown in the Table 1.

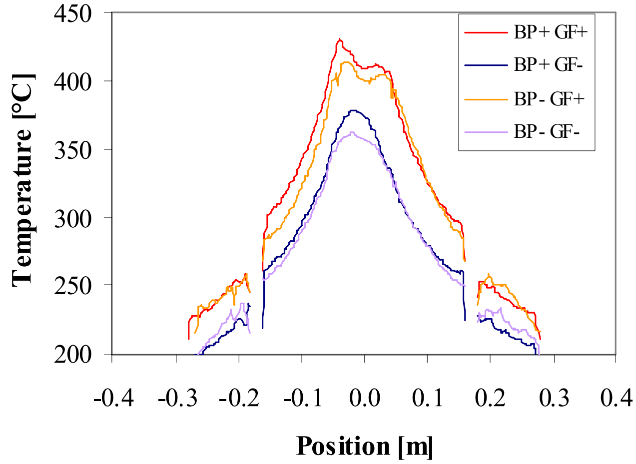



Figure 6. Temperature at the bottom plate centerline at different total burner power. Experimental conditions are shown in the Table 1.

### 4.2 Comparison between model and experiment

Table 4 shows a comparison between the results obtained with the model and the experiments. The global resistance of the back wall was estimated to be 0.5 W/K. Differences as high as 60% in the predicted and measured temperatures are seen in the Table 4. The total power obtained by the numerical model is 2,9 kW, while the experimental power is 4,05 kW. These differences show that the model oversimplifies the problem.

Table 4. Comparison between modeling and experimental results.

| Wall | Temperatures [◦C] | | |
|---|---|---|---|
| | **Model** | **Experimental** | **Difference** |
| 1. Bottom | 338 | 338 | 0% |
| 2. Top | 323 | 272 | 19% |
| 3. Back | 318 | 217 | 46% |
| 4. Right | 325 | 256 | 27% |
| 5. Door | 317 | 197 | 60% |
| 6. Left | 325 | 262 | 24% |

Better results can be obtained if the global wall resistances are adjusted to result in less than 5% of error for the predicted wall temperatures. In this case the total power predicted by the model increases to 3,9 kW, which is 4% lower than the experimental value. Table 5 shows the new global resistances obtained. The largest difference is observed in the door global resistance that is one order of magnitude larger than the previous value. For the other walls, the differences are less than 15%. These results suggest that the experimental procedure to evaluate the global thermal resistance of the door was incorrect. The main source of error is probably the estimative of the conduction heat flux across the glass.

Table 5. Global resistances: experimental and modeling results.

| Cavity Wall | R experimental (W/K) | R adjusted (W/K) |
|---|---|---|
| Top | 2.51 | 2.69 |
| Left | 2.31 | 2.03 |
| Right | 1.87 | 2.03 |
| Door | 0.02 | 0.53 |

## 5. CONCLUSIONS

The present work proposes a methodology for the thermal and fluid dynamic analysis of domestic gas ovens for model validation. The methodology is based on the temperature mapping and velocity measurements in the oven cavity. A global one-dimensional model is developed in order to estimate the surface temperatures and thermal resistances of the walls. The proposed experimental methodology is able to provide the required information to the validation of simulation tools for oven design. The one dimensional model is able to predict simple features of the thermal behavior of the oven. The largest difference in the measured and estimated global resistances is observed in the door. For the other walls, the differences are less than 15%. The main source of error is probably the estimative of the conduction heat flux across the glass.

## 6. REFERENCES

Dewitt, David P., Incropera, Frank P., 2003, “Fundamentals of Heat and Mass Transfer”, 5th edition.

Kuznetsov, G.V., Sheremet, M.A., 2009, “Conjugate natural convection with radiation in an enclosure”, International Journal of Heat and Mass Transfer, Vol. 52, pp. 2215 – 2223.

Navaneethakrishnan, P., Srinivasan, P. S. S., Dhandapani, S., 2007, “Heat transfer and heating rate of food stuffs in commercial shop ovens”, Sādhanā, Vol. 32, part 5, pp. 535 – 544.

Silva, A. K., Mantelli, M. B. H., 2004, “Thermal applicability of two phase thermosyphons in cooking chambers”, Apllied Thermal Engineering, Vol. 24, pp. 717 – 733.

Wang, H., Xin, S., Le Queré, P., 2006, “Étude numérique du couplage de la convection naturelle avec le rayonnment des surfaces en cavité carrée remplie d’air”, C. R. Mecanique, Vol. 334, pp. 48 – 57.

## 7. ACKNOWLEDGEMENTS

This work was developed within the framework of project “Thermal Management” sponsored by Whirlpool Latin America.

## 8. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.