

# Quantitative IR Thermography for continuous flow microwave heating

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The work here presented proposes an enhanced procedure to describe bulk temperature profiles in pipe flow subjected to microwaves, by means of on-line infrared temperature readouts. Such procedure is intended to overcome the well-known difficulties connected with temperature readings while microwave heating takes place. In fact, temperatures are usually measured onto few points by means of fiberoptic probes, which are often not reasonable for properly controlling the process at hand, as the electromagnetic distribution inside the cavity yields highly uneven spatial distribution of the temperature field. This study proves that looking inside the microwave oven through a metallic grid can be successfully realized, provided proper preliminary calibration procedures are in place. In order to assess whether the resulting temperature readouts are reasonable, experimental bulk temperature related to two temperature levels and two flow rates, are compared with numerical results obtained by running a 3D FEM model, developed by the authors. Experimental and numerical results were found in quite satisfying agreement.

*Key-Words:* - Fiberoptic probes, Infrared thermography, Infrared image processing, Microwave heating of liquids, Maxwell's equations, Numerical modelling, Experimental validation, FEM, CFD, Comsol multiphysics.

## 1 Introduction

Using microwaves (MW) to thermally treating liquids is an effective, low-cost way of processing. Warming takes place directly inside the exposed mass in contrast with conventional methods where heating is transferred through convection and heat conduction from the surface to the inner side, [1] - [2]. It is widely recognized that attracting advantages of MW application over conventional heating techniques essentially consist in minor periods needed for the processing, higher energy efficiency, absence of contaminants released in the environment. The latter aspect is becoming particularly attractive, as for instance proved by specific investments set under the Horizon 2020 program, aimed to the reduction of pollutant sources in the Mediterranean. While there is a large amount of papers dealing with numerical modelling of MW heating of solid substances [3]-[10], minor attention seems to have been paid to liquids, see for instance [11]-[13], despite a well-established interest into thermally

treating them [14]-[19], e.g., for heating, drying, melting, pasteurizing, sterilizing, transesterification processing etc. First attempts have been made into modelling the MW power transferring by theoretical models, i.e. based on the Lambert's law. Nevertheless, such an assumption is not valid for small samples (that is the case at hand) in which expected EM spatial gradients must be derived by solving Maxwell's equations. Concerning liquids, a numerical approach is needed, especially when such a set of equations is coupled with convective heat transfer and liquid momentum equations. Not to mention, the case in which dielectric properties are assumed to be temperature dependent.

Among the others (MEFiSTo-3D, Microwave studio, QuickWave-3D and so on), there are basically two main commercial models capable of solving thermal treatment by MWs: ANSYS Multiphysics [20] and COMSOL Multiphysics [13], [21], [22], [23]. A comparison between ANSYS and COMSOL models is proposed in [12].

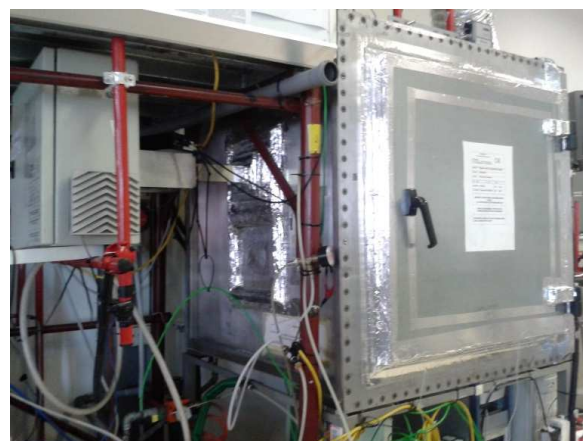
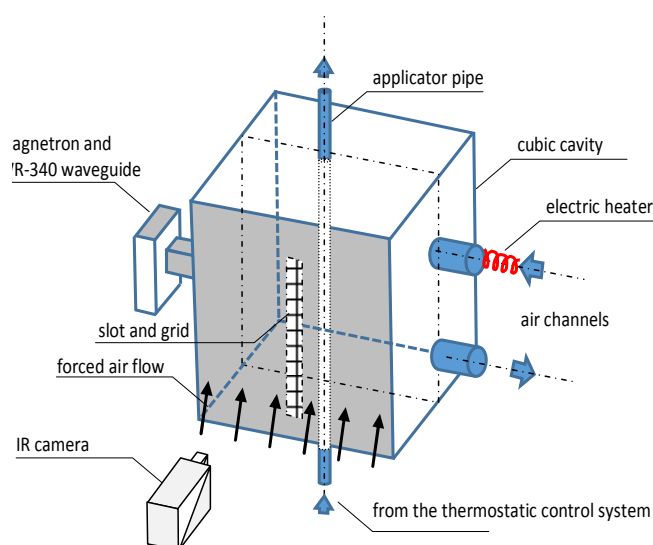


Fig.1. Sketch and picture of the available MW pilot plant

The correctness of the adopted numerical procedure should be checked by experimental validation of the results. Still, there are few cases of literature where such a requirement is fulfilled (see for instance [24], [25]). Although the experimental approach provides evidence for the matter being investigated, satisfactory experimentation may be unpractical as a great amount of tests and an adequate number of sampling points are usually necessary to obtain reliable results since scale-up analysis is not allowed. Measurement of interesting parameters is not an easy task as the measuring devices may interfere with the electromagnetic field and with the flow motion in continuous processes. To tackle with the latter issue, appropriate procedures for field measurement have been long established in the past. Mamouni and co-workers [26], [27] proposed the correlation microwave thermography technique (CMWT) for the accurate description of thermal distribution on material's surfaces. In [28] kidney-bladder system pathologies are assessed by correlating MW radiometry with temperature measurements taken by fiber optic probes. Cuccurullo et al., proposed an innovative procedure for real time controlling apple drying based on IR thermography, [29].

Here, a first attempt is made for a quantitative infrared thermography temperature readout to describe in real time temperature field inside an illuminated MW cavity. Given the fact that temperature measurements are usually taken onto few points by means of fiber optic probes, the proposed procedure is intended to promote higher resolutions than standard's. Such an approach is needed, as strongly uneven spatial distribution of the temperature field, produced by MW application, are

expected. Considering the theoretical results obtained in [22], a linear distribution of temperature can be assumed for high flow rates, flowing in a circular pipe exposed to MWs. On such assumption, a procedure, based on the processing of infrared images taken inside the oven and able to provide the water bulk temperature field along the pipe, is proposed.

## 2 Materials and methods

### 2.1 Experimental setup

Experiments were performed in a microwave pilot plant, Figure 1, intended for general purposes in order to encompass different loads, i.e. different materials and samples distributions, weight, size. Microwaves were generated by a magnetron rated at 2 kW nominal power output and operating at a frequency of 2.4 GHz. A rectangular WR340 waveguide connects the magnetron to the cavity. Microwaves illuminated an insulated metallic cubic chamber (0.9 m side length) housing the pyrex (MW transparent) glass applicator pipe (8 mm inner diameter, 1.5 mm thick) carrying water continuous flow to be heated. The inner chamber walls were insulated by polystyrene slabs black painted. The pipe was placed inside the chamber in such a way that its longitudinal axis lied down along a symmetry plane due to both geometry and load conditions. Such a choice was realized having in mind to suitably reduce computational efforts, as next explained.

A circulating centrifugal pump drawn out water from a thermostatic bath to continuously feed the applicator-pipe with a fixed inlet temperature. The

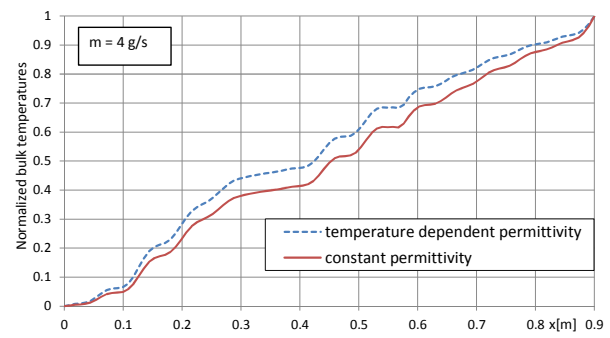
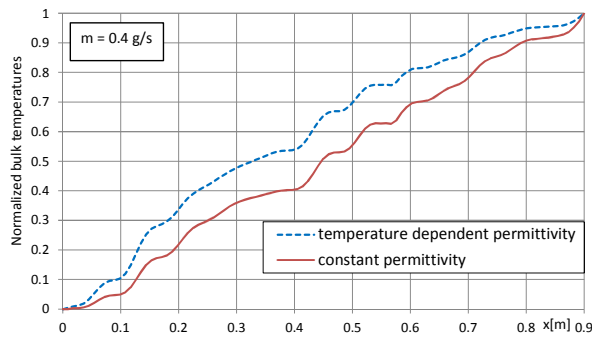


Fig.2. Normalized bulk temperatures for an arbitrarily fixed power absorption

flow rate was accurately tuned by acting on an inverter controlling the pump speed. The liquid leaving the cavity was cooled by a heat exchanger before being re-heated by the thermostatic control system in order to obtain the previous inlet temperature, thus realizing a closed loop.

A centrifugal fan facilitated the air removal by forcing external air into the cavity; the renewal air flow was kept constant throughout the experiments in order to stabilize the heat transfer between the pipe and the environment. The channel feeding the external air flow was equipped with an electric heater controlled by the feedback from a thermocouple in order to realize a fixed temperature level for the air inside the illuminated chamber, that is 23°C.

A longwave IR radiometer, thermaCAM by Flir mod. P65, looked at the target pipe through a rectangular slot 30 mm x 700 mm properly shielded with a metallic grid trespassing by infrared radiation arising from the detected scene (less than 15 μm wavelength for what of interest) but being sealed for high-length EM radiation produced by the magnetron (12 cm wavelength). Finally, a further air flow was forced externally, parallel to slot holding the grid in order to establish its temperature to 19 ± 2.5°C.

## 2.2 The numerical procedure

A 3D finite element (FEM) model was developed in COMSOL v4.4 environment [30]. The software allows coupling electromagnetic, flow and thermal problems to predict temperature distribution in the moving fluid. The need of considering coupled physics and thus a general solution, arises by noting that, due to the geometry at hand, no simplified heating distributions can be sought (i.e. the ones based on Lambert Law's) [31].

The electric field distribution  $\underline{E}$  in the microwave cavity, both for the air and for the applicator pipe carrying the fluid under process, is determined by imposing

$$\nabla \times \left( \frac{1}{\mu_r} \nabla \times \underline{E} \right) - k_0^2 \left( \varepsilon_r - \frac{i\sigma}{\omega \varepsilon_0} \right) \underline{E} = \underline{0} \quad (1)$$

where  $\varepsilon_r$  and  $\mu_r$  are the relative permittivity and permeability respectively,  $\omega$  is the angular wave frequency,  $k_0$  is the wavenumber in vacuum, and  $\sigma$  is the electric conductivity. EM properties are assumed independent of temperature, thus allowing to uncouple the thermal and the EM sub-problems: obviously, this move allows to dramatically reduce computational efforts. The effects of such a choice are evaluated later on.

Assuming negligible resistive material losses, inner walls can be assumed as perfect electric conductors, that is,  $\underline{n} \times \underline{E} = \underline{0}$ , for the waveguide and the cavity surfaces, being  $\underline{n}$  the local normal vector.

Continuity boundary conditions were set by default for all the interfaces between the confining domains, that is, the pipe, the cavity, and the waveguide. Such condition may be expressed as

$$\underline{n} \times (\underline{E}_i - \underline{E}_j) = \underline{0} \quad (2)$$

being  $i$  and  $j$  the neighbouring discrete facets, sharing the interface at hand.

Due to the symmetry of the problem, and load conditions around the plane crossing vertically the oven, the waveguide, and the pipe, the model is reduced to one-half of the device, allowing a more accuracy in the calculation. The condition of perfect magnetic conductor was applied for the surfaces yielding on the symmetry plane:

$$\underline{n} \times \underline{H} = \underline{0} \quad (3)$$

$\underline{H}$  being the magnetic field, which has to be therefore parallel to the local normal vector  $\underline{n}$  on the symmetry plane.

Temperature distribution is determined for fully developed Newtonian fluid in laminar motion,

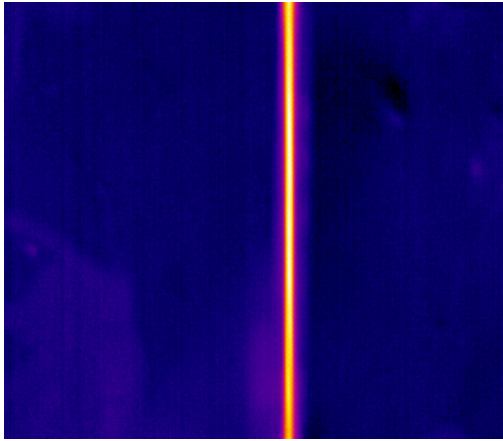


Fig. 3. Net apparent applicator pipe temperatures

considering constant flow properties; in such hypotheses, the energy balance reduces to

$$\rho c_p U \frac{\partial T}{\partial X} = k \nabla^2 T + U_{\text{gen}} \quad (4)$$

where  $T$  is the temperature,  $\rho$  is the fluid density,  $c_p$  is the specific heat,  $k$  is the thermal conductivity,  $X$  is the axial coordinate,  $U(R) = 2U_{\text{av}}(1 - 4R^2/D_i^2)$  is the axial Poiseuille velocity profile,  $D_i$  being the internal pipe diameter and  $R$  the radial coordinate,  $U_{\text{av}}$  is the average velocity on the cross section being considered;  $U_{\text{gen}}$  is the specific heat generation, i.e. the “electromagnetic power loss density” ( $\text{W}/\text{m}^3$ ) resulting from the numerical solution of the EM problem. The power-generation term realizes the coupling of the EM field with the energy balance equation. It represents the “heat source” term:

$$U_{\text{gen}}(X, Y, Z) = \frac{1}{2} \omega \epsilon_0 \epsilon'' |E(X, Y, Z)|^2 \quad (5)$$

being  $\epsilon_0$  the free-space permittivity and  $\epsilon''$  the relative dielectric loss of the material.

In order to exploit the bulk temperature sensitivity to the temperature dependent permittivity, the plots shown in Figures 2 are realized. The constant and variable permittivity cases are considered by using the approach developed by the same Authors and reported in [21]: it can be obviously inferred that for increasing mass flow rates the difference between the curves are vanishing due to reduced temperature increases for fixed heating power. As a consequence, in what follows, restricting experimental tests to high flow rates and having in mind to verify if experimental temperature readouts are reasonable, constant permittivity values are considered.

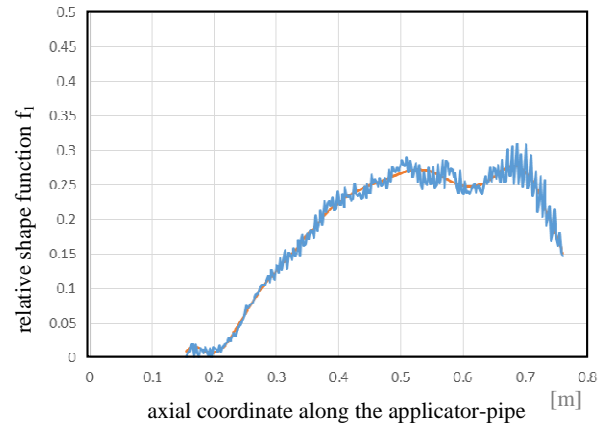


Fig. 4. Measured and interpolated relative shape-function  $f_1$

### 2.3. Temperature readout procedure

The presence of the grid is a major obstacle wishing to perform temperature-readout when looking inside the illuminated cavity. The focus is set on the applicator pipe, while the instantaneous field of view (IFOV) of the radiometer in use may well find the hot spots corresponding to the pipe below the grid. Nevertheless, the radiometer does not accurately measure pipe temperatures due to the slit response function (SRF) effect. Because of the SRF, the object's temperature drops as the distance from the radiometer increases. The latter was set in order to encompass in the IR image the maximum pipe extension compliant with the available slot-window carrying the grid. On the other hand, there is the need of getting as close to the target as possible, in the respect of the minimum focal distance.

A preliminary calibration and a suitable procedure have been then adopted. First, aiming to reduce reflections, the glass-pipe, the grid and the cavity walls have been coated with a high emissivity black paint whose value was measured to be  $\epsilon_{\perp} = 0.95$  along the normal (perpendicular line drawn to the surface). In principle, this value is directional and as such it is affected by the relative position of the target with respect to the IR camera.

Then the following two configurations have been considered:

- the “test configuration”, i.e. the applicator-pipe carrying the fluid @ fixed inlet temperature.
- the “reference configuration”, i.e. a polystyrene slab placed inside the cavity in order to blind the pipe to the camera view. The slab was black painted to realize a normal emissivity of 0.95 and its temperature,  $T_{\text{slab}}$ , was measured by four fiberoptic probes.

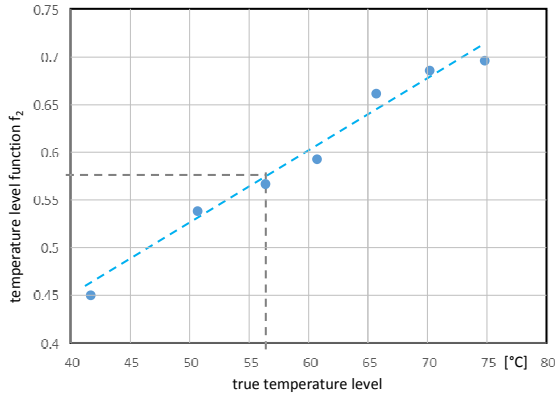


Fig 5. Temperature level function  $f_2$  obtained with a linear regression

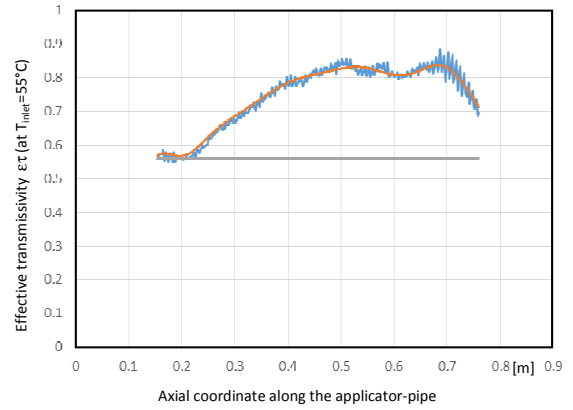


Fig. 6. The reconstructed and measured true temperature profiles @  $T_{inlet} = 55^\circ\text{C}$

For both (a) and (b) configurations, neglecting the atmosphere contribution, the fundamental equation of IR thermography relates the spectral radiant power incident on the radiometer to the radiance leaving the surface under consideration. For the case at hand, the attenuation due to the grid must be taken into account. The radiance coming from the inner walls is attenuated by a factor  $\tau$ , which can be defined as “grid transmittance”, which accounts for the SRF grid effect. The latter parameter depends on both the geometry and the temperature level involved. Additionally, the radiometer receives both the radiance reflected from the external surroundings ambient to the grid and the emission by the grid itself. The inner and outer surrounding environments are considered as a blackbodies @ uniform temperatures  $T_i$  and  $T_o$ , respectively. Finally, the radiometric signal weighted over the sensitivity band by the spectral response of the detection system, including the detector sensitivity, the transmissivity of the optical device and amplification by the electronics, is proportional to the target radiance as follows:

$$I(T_{slab}^{app}) = \tau [\epsilon I(T_{slab}) + (1-\epsilon) I(T_i)] + (1-\tau) [\epsilon I(T_{grid}) + (1-\epsilon) I(T_o)] \quad (6)$$

$$I(T_{pipe}^{app}) = \tau [\epsilon I(T_{pipe}) + (1-\epsilon) I(T_i)] + (1-\tau) [\epsilon I(T_{grid}) + (1-\epsilon) I(T_o)] \quad (7)$$

where:

- (1)  $I$  is the blackbody Planck’s function, evaluated at a suitable wavelength, within the radiometer spectral sensitivity window,  $\lambda^+$ . With reference to the radiometer in use, this latter parameter turns out to be  $\lambda^+ = 11.5 \mu\text{m}$  as obtained by instrument calibration data.

- (2) the apparent temperatures are the surface ones given by an equivalent blackbody as seen by the IR equipment, both for the pipe,  $T_{pipe}^{app}$ , and the polystyrene slab,  $T_{slab}^{app}$ .

In the above equations, the grid transmittance must be considered strongly dependent on the position since the grid contribution to emission assumes a different weight from point to point. In fact, in spite of the tangential air flow running over the grid, an irregular temperature distribution is still observed, probably arising as an after effect of eddy currents occurrence.

The calibration curve related to the specific radiometer allows the retrieval of apparent temperatures, see next paragraph. The exogen contributions due to both the reflections arising from inner and outer walls clearly appear. Only the former contribution is attenuated by the presence of the grid. The last contribution represents the reinforcement due to the grid emission.

Subtracting the two previous equations (6) and (7), the contributions due to both the grid and the inner and outer environments cancel, thus yielding “the net equivalent emissivity for cold surroundings”,  $\epsilon\tau$  - function:

$$\epsilon\tau = \frac{I(T_{pipe}^{app}) - I(T_{slab}^{app})}{I(T_{pipe}) - I(T_{slab})} \quad (8)$$

Most commonly in thermography, the subtraction (Figure 3) of images related to configurations (a) and (b) produces the further benefit of cancelling the grid established uneven temperature distribution.

In order to explicit the dependence of the radiance on the temperature, the Plank’s law can be rearranged, considering that

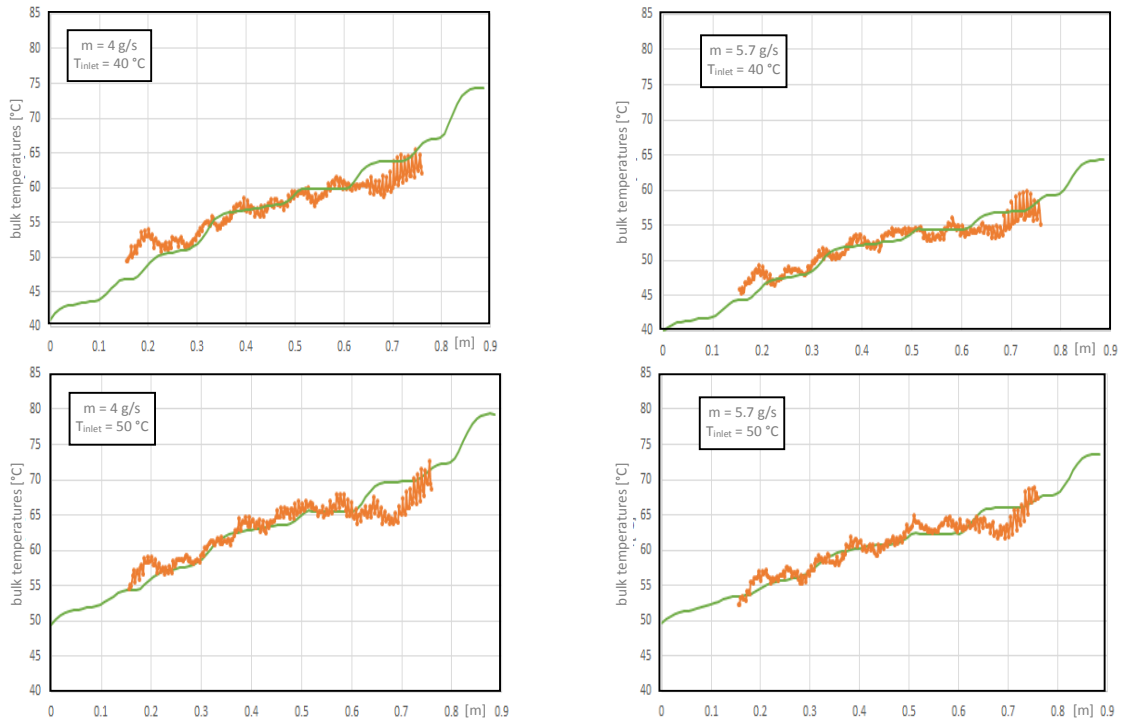


Fig.7. Numerical and experimental bulk temperatures for inlet temperatures  $T_{inlet}= 40$  and  $50\text{ }^{\circ}\text{C}$ , and two flow rates  $m = 4$  and  $5.7\text{ g/s}$ .

$$\frac{dI_{b,\lambda} / I_{b,\lambda}}{dT / T} = \frac{C_2}{\lambda T} \frac{e^{\frac{C_2}{\lambda T}}}{e^{\frac{C_2}{\lambda T}} - 1} \approx \frac{C_2}{\lambda T} \quad (9)$$

$C_2 = 14390\text{ }\mu\text{m K}$  being the 2nd radiation constant. The approximation due to Wien, which is recalled in the last passage, leads to an error less than 1% if the wavelengths are not greater than  $2.5\lambda_{max}$ ,  $\lambda_{max}$  being the wavelength for which the maximum emission is attained at the actual temperature. These conditions are fully recovered for the case at hand. In view of the previous expression, if one assumes  $I \approx T^n$ , it is readily recovered that  $n \approx 5\lambda_{max} / \lambda^+$ . Finally, the last expression can be set as:

$$\varepsilon\tau = \frac{(T_{pipe}^{app})^{n(T_{pipe}^{app})} - (T_{slab}^{app})^{n(T_{slab}^{app})}}{(T_{pipe})^{n(T_{pipe})} - (T_{slab})^{n(T_{slab})}} \quad (10)$$

The notation resembles that the index  $n$  depends on the temperature itself. In view of this equation, provided the apparent temperatures are known via IR thermography and the slab temperature via the fiberoptic probes, the function  $\varepsilon\tau$  is required in order to infer the pipe temperature. The calibration is then

intended to determine the  $\varepsilon\tau$  as function of both geometry and temperature levels. The calibration is carried on as follows. First, the water flow discharge was kept as high as a linear temperature increase could be experienced for the bulk temperature of the flowing fluid [22]. Thus, measuring the fluid inlet and outlet temperatures by thermocouples placed externally to the illuminated cavity, allowed to know the true bulk water temperature along the longitudinal  $x$  axis of the pipe,  $T_{pipe}^{bulk}(x)$ . On the other hand, the need of realizing high flow rates determined temperature increases contained within  $\Delta T_{bulk} = 2.6\text{ }^{\circ}\text{C}$  and thus allowed to fix the corresponding temperature level for the test at hand. Seven inlet temperatures, namely  $T_{inlet} = 40, 50, 55, 60, 65, 70$  and  $75\text{ }^{\circ}\text{C}$ , were considered which covered the temperature range involved in the successive experimental tests.

Assuming that the surface pipe temperature closely approaches the fluid bulk temperature  $T_{pipe}(x) \cong T_{pipe}^{bulk}(x)$ , knowing the apparent temperatures via IR thermography and measuring the slab temperature, eq. (10) allows to evaluate  $\varepsilon\tau$  as function of the axial coordinate along the tube longitudinal axis ( $x$ ) in correspondence of the seven preset true temperature levels,  $T = (T_{inlet} + T_{outlet})/2$ .

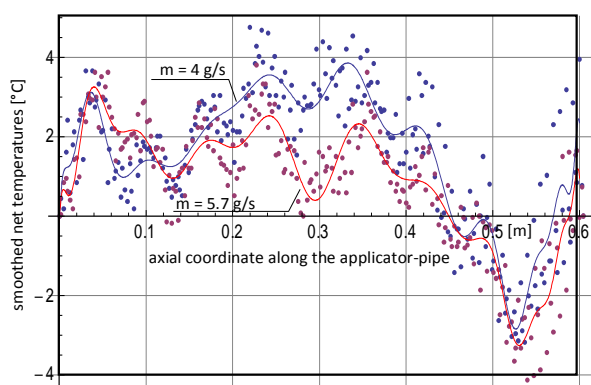


Fig.8. Polynomial fitting of net temperatures data for two flow rates, taking  $T_{inlet} = 50\text{ }^{\circ}\text{C}$

In particular, since the relative-shape of the  $\varepsilon\tau$ -function along the axial coordinate appeared to be the same but its level was increasing with temperature level, the  $\varepsilon\tau$ -function was sought as  $\varepsilon\tau(x, T) = f_1(x) + f_2(T)$ .

The function  $f_1$  was recovered by averaging the seven relative shapes and turned out to be well approximated ( $R^2 = 0.996$ ) by a 10-th order polynomial, Figure 4, while  $f_2$  resulted to be almost linear with temperature, Figure 5, the maximum relative error being contained within 3%, i.e.  $R^2 = 0.974$ . As an example, the measured  $\varepsilon\tau$ -function @  $55^{\circ}\text{C}$  is compared with the corresponding one based on the above interpolating structure in Figure 6: the satisfying agreement is witnessed by an average relative error contained within 1.68%.

Applying the same procedure to the remaining selected temperature levels similarly shows that measured and reconstructed curves fairly agree, thus showing that the procedure at hand fits well the experimental data. The maximum among the average relative errors corresponding to the selected temperature levels is 2.3 %.

The knowledge of the  $\varepsilon\tau$ -function enables the IR image processing described in the following section.

## 2.4 Image processing

In view of the discussion previously presented, tests related to configuration (a) were performed. Provided steady conditions were attained, an image sequence at the rate of 10 images per second was taken for two seconds. Since frames in time sequence differ only in the distribution of random noise, twenty frames were averaged to form a single image with noise variance reduced proportionally to the sequence length. Performing a horizontal scan of the formed image, the maximum apparent temperature for each row was

mass flow rate [g/s]	$T_{inlet}$ [ $^{\circ}\text{C}$ ]	RMSE [ $^{\circ}\text{C}$ ]
4	40	5.5
5.7	40	4.1
4	50	4.6
5.7	50	3.6

Table 1. RMSE of bulk temperatures for different mass flow rates and temperature levels

extracted, identifying in such way the proper pipe longitudinal apparent temperature.

In a similar fashion, processing an image sequence related to configuration (b), the apparent slab temperatures were extracted in correspondence of the pixels selected by processing configuration (a). Finally, eq. (10) was solved for each point  $x$  along the pipe axis in terms of the unknown true pipe temperature  $T_{pipe}$ ,  $T_{slab}$  being measured by the fiberoptic probes.

## 3 Results and discussion

Experimental curves and the corresponding numerical ones resulting from data reduction are reported in Figure 7. Two temperature values of inlet temperature, namely 40 and  $50\text{ }^{\circ}\text{C}$ , and two flow rates, that is 4 and 5.7 g/s, were considered. Thermal and dielectric properties used during the numerical simulations were evaluated in correspondence of the average temperature of the flowing liquid resulting from the experiments. The input power feeding the numerical model was chosen such that the total power loss was equal to the one resulting from the experiments by calorimetric computations.

A quite satisfying agreement between experimental and numerical data exists, at least in terms of qualitative behaviour: the uneven temperature distribution which is due, in turn, to the EM field patterns, seems to be recovered. The corresponding RMSE values, as shown in Table 1, confirm that increasing the flow rates increases the accuracy of the prediction. This occurrence is expected since the uneven temperature distribution inside pipe section is reduced with increasing the fluid average speed [22], both because of the increasing frequency effect of the forcing EM field and of the increased velocity which compensates the augmented energy concentration in

the region nearby the longitudinal axis of the pipe. Therefore, bulk temperature values are closely approached by the wall temperature in each section, as supposed in advance.

A further proof of the reliability of the experimental tests may be deduced by generating a symbolic representation (see Figure 8), giving the best fit of experimental temperature profiles evaluated in excess with respect to the local bulk temperatures corresponding to the uniform heat generation solution provided by the same Authors [21]. In fact, this “move” allows evidencing the expected increasing attenuation and delay of the net temperature profiles because of the increased flow rates.

#### 4 Conclusion

Bulk temperature distribution in water flowing in a glass-pipe, subjected to microwaves, were estimated by a new experimental procedure developed by the Authors. High-resolution thermal maps in real time were obtained by processing infrared thermographic images. The presence of the grid between the pipe and the radiometer required a suitable calibration in order to evaluate the  $\epsilon\tau$ -function along the pipe, before performing the experimental tests.

Two temperature levels, namely 40 and 50 °C, and two flow rates, that is 4 and 5.7 g/s, were considered. The experimental values of bulk temperature were compared with results obtained by solving a FEM model, which accounts for constant dielectric and thermal properties. A quite satisfying agreement between experimental and numerical data was found, with a better prediction for higher flow rates.

A wider flow range has been explored and the effect of temperature dependent fluid properties will be implemented in the numerical model in order to improve the theoretical prediction of the experimental results.

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