Application of a Zone Model to the Simulation of Heat Transfer in Fire Resistance Furnaces Piloted with Thermocouples or Plate Thermometers

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ABSTRACT

During the previous years, visible differences have been observed on the thermal response of a given specimen placed in standard furnaces piloted with thermocouples. The definition of a method for harmonising the severity of the heat flux absorbed by a specimen in fire resistance furnaces is necessary in order to reduce the differences observed. For several years, the project of using plate thermometers instead of thermocouples to control the furnace temperature has been examined and discussed. These new sensors can be more appropriate than thermocouples to impose to the specimen a heat impact as they seem to be largely independent of the furnace characteristics. A new zone model has been developed to calculate the heat fluxes exchanged in the furnace and the temperature of the specimen, the furnace being controlled either by thermocouples or by plate thermometers. The model's equations bring more light on the reasons why the heat flux received by a plate thermometer is weakly dependent of construction parameters. The main features of this model are described. In order to present the capabilities and limits of the model, calculated results are compared to recent test results obtained on a reference specimen equipped with "calibration elements", and prediction results are given for some different specimens for which no tests have been executed, showing the advantages of plate thermometer control.

KEYWORDS : heat transfer, fire resistance test, computer model, furnace harmonisation, plate thermometer.

NOMENCLATURE

c	specific heat ($J \cdot kg^{-1} \cdot K^{-1}$)
Dt	time step
d	diameter of thermocouple bead (m)
E	radiative incident heat flux ($W \cdot m^{-2}$)
F	view factor
h	convective heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
J	radiative flux leaving a surface ($W \cdot m^{-2}$)
k	thermal conductivity ($W \cdot m^{-1} \cdot K^{-1}$)
l	characteristic length (m)

Greek letters

ε	emissivity
κ	attenuation coefficient (m^{-1})
σ	Stefan-Boltzmann constant ($W \cdot m^{-2} \cdot K^{-4}$)
τ	transmittance (m^{-1})

subscripts

est	estimated value
conv	convective

m	mass (kg)	g	gas
Nu	Nusselt number	PT	plate thermometer
Pr	Prandlt number	S	surface
Re	Reynolds number	sens	sensor
S	surface area (m^2)	spec	specimen
T	temperature (K)	thc	thermocouple
t	time (s)		

INTRODUCTION

In the fire resistance standard test procedure, the gas temperature in a furnace is recorded by thermocouples placed near the specimen and this recorded value has to follow a given time-temperature evolution, e.g. ISO 834, [9]. As the design and characteristics of existing test furnaces vary considerably (dimensions of the furnace, nature of the fuel, number of burners), various furnaces may expose a given specimen to different heating conditions. Several previous works did show that the fire resistance rating of a structure element tested in different furnaces in accordance with ISO 834 may differ from a furnace to another (e.g. [4]). A possible way for reaching harmonised fire resistance tests could be the design of an unique type of furnace, with the drawback of implying an important investment. The way followed presently in the European Union is to control the existing furnaces with "plate thermometers" instead of thermocouples in order to reduce the differences observed between them [15], [16]. Several previous studies have already discussed the advantages of this possibility [3], [4]. Some models have been developed or used to represent the heat exchanges in a fire resistance furnace : zone models [8], [13], or field models [2], [14]. As field (or CFD) models offer the advantage of a more realistic and detailed description of heat transfer, their use is not easy and they are requiring a pre- and post-processing. Zone models are less exact but can be run on a PC. In a simple furnace zone model, the unique gas temperature is set equal to the temperature of the controlling sensor. As thermocouples re-radiate heat, their equilibrium temperature is lower than the gas temperature and the difference can reach about 100 °C. In the zone model developed in this study, ECHAFO [5], [6], [7], we distinguished these two temperatures and introduced an equation of heat balance on a thermocouple or on a plate thermometer in order to calculate the gas temperature from the given sensor temperature. The only other zone model offering this possibility is the one presented by Harada, Yabuki and Terai, [8], that was developed independently during the period when this study was realised. The model ECHAFO is used here to present new evidence on the reasons why a plate thermometer should permit to control a furnace independently of its characteristics. Some comparisons are given between measured and calculated results. Predictions are then presented for different types of specimens, the furnace being controlled either by thermocouples, or by plate thermometers.

2 THE MODEL

ECHAFO belongs to the zone models category. In ECHAFO, the temperature of the sensor used to control the furnace is given as a function of time, the unknowns are the zone temperatures. The furnace gas volume is the unique volume zone ; its characteristics are : gas temperature T_g (variable to be calculated), gas transmittance τ_g (supposed constant) or absorption coefficient κ, and dimensions of the occupied volume (constant). Other zones are real or virtual surfaces :

- the exposed surfaces of the five furnace walls,
- the exposed surface of the specimen,
- the exposed surfaces of the calibration elements used with the reference specimen (see § 3),
- the exposed surfaces of the piloting sensors (thermocouples or plate thermometers),

– The unexposed surfaces corresponding to the exposed surfaces listed above. For a thermocouple, there is no unexposed surface. We assume that temperature is uniform in the bead (same value on the surface and in the bulk).

The writing of all the thermal exchange expressions (convective, radiative, and conductive) leads to a system of equations in which T_g is an unknown as all the surface temperatures. The temperature of the sensor used to pilot the furnace, T_{thc} or T_{PT} , is of course introduced into the equations, and is given as input data. It is supposed to be fixed by the ISO 834 law (T_{ISO} = T_0 + 345 Log_{10} (8 t+1)), where T_0 is the initial temperature and t is in min.).

The gas emissivity ε_g is calculated through $\varepsilon_g = 1 - \exp(-\kappa\, l_f)$, where κ is a given constant, and $l_f = 3.6 \times$ (Volume of the furnace/Total area of the five walls plus specimen). In this model, κ is "tuned" once for each furnace.

We consider in the model one calibration element (see § 3), and one sensor, both of them supposed to be placed on an axis perpendicular to the specimen, near the centre of the specimen. For the comparisons made between calculated temperatures of a zone e.g. (surface of calibration element or of specimen) to measured temperatures of corresponding objects, we compared the unique calculated temperature to the arithmetic mean value of the values recorded during the test at different locations. The calculation of radiative heat fluxes is based on view factors between the whole zone surfaces and not between parts of them : this simplification leads to average values of fluxes on these surfaces.

The convection coefficients h (for the walls, the specimen, thermocouples and plate thermometers) are expressed through simple engineering formulae.

The equation of heat diffusion (in walls, specimen and plate thermometer) is solved with the help of a finite differences scheme where the thermal conductivities of solids may vary with temperature.

2.1 Heat impact on exposed objects

2.1.1 Heat exchanges on a thermocouple bead

– **Convective heat flux**

We use the following relations, relative to a gas flow impinging to a cylinder :

$h_{thc} = Nu\dfrac{k_{air}}{d}$ is the coefficient of convective heat transfer, where k_{air} is given the value for air (function of gas temperature).

We assume that the Nusselt number is given by : $Nu = C\,Re^m Pr^{1/3}$

where Re is the Reynolds number, $Re = v\dfrac{d}{n_{air}}$,

and Pr is the Prandtl number, supposed to be constant, Pr =0.73.

with the following empirical coefficients in the expression of Nu [1] :

$0.4 < Re \le 4$	$C = 0{,}989,\ m = 0.330$	$40 < Re \le 4000$	$C = 0{,}683,\ m = 0.466$
$4 < Re \le 40$	$C = 0{,}911,\ m = 0.385$	$4\ 10^3 < Re \le 4\ 10^4$	$C = 0{,}193,\ m = 0.618$

v at about 10 cm from the specimen, was given a single constant typical value, 3 m · sec^{-1}, that in real world may vary from a location to another, and from a furnace to another. This value was estimated as a typical average figure obtained from some computations with the field model SOFIE.

These expressions lead to high values of h_{thc}, up to 100 W · m^{-2} · K^{-1} or more.

The convective heat flux entering the thermocouple bead is given by : $h_{thc}(T_g - T_{thc})$ (W · m^{-2})

where T_g is the gas temperature, and T_{thc} the temperature of the thermocouple.

- **Radiative heat flux**

The radiative incident flux is here given by : $E_{thc} = \varepsilon_g \sigma T_g^4 \quad + \quad \tau_g \frac{\sum_{j=1}^{j=6} S_j J_j}{\sum_{j=1}^{j=6} S_j}$

where ε_g is the gas emissivity, σ is the Stefan-Boltzmann constant, τ_g is the transmittance of the gas volume (τ_g = 1 - ε_g), S_j is the area of wall j, and J_j is the radiative flux leaving wall j (emission plus diffusion). The terms J_j are calculated by solving an implicit system of equations relating incident heat fluxes to emitted and diffused fluxes. In this expression of E_{thc}, we assume that the thermocouple bead is receiving radiated heat from all the walls, that is not quite true as the thermocouple bead cannot see the wall behind it. The advantage of this simple assumption, applied to an "average" thermocouple", with no precision required on its position, is that it does not imply to calculate view factors. We observed that the error generated by this simple approach is not influencing the temperature of the thermocouple bead on a visible way because the convective heat flux on the thermocouple is dominating the radiative flux.

- **Heat balance on the thermocouple bead**

$S_{thc}\left[h_{thc}\left(T_g - T_{thc}\right) + \varepsilon_{thc} E_{thc} - \varepsilon_{thc}\sigma T_{thc}^4\right]$ is the power (W) entering the thermocouple mass, then:

$$h_{thc}\left(T_g - T_{thc}\right) + \varepsilon_{thc} E_{thc} - \varepsilon_{thc}\sigma T_{thc}^4 = \frac{m_{thc}}{S_{thc}} c_{thc} \frac{dT_{thc}}{dt}$$

where S_{thc} is the exposed area, m_{thc} the mass, and c_{thc} the heat capacity of the thermocouple per mass unit.

We assume a uniform temperature T_{thc} in the sensor et we neglect other heat loss terms. After a few seconds, the right hand side is quite negligible.

2.1.2 Heat exchanges on the exposed face of a plate thermometer

- **Convective heat flux on the exposed face**

$\dot{q}''_{conv,PT} = h_{PT}(T_g - T_{PT})$ is the convective flux entering this face, where h_{PT} is given by :

$$h_{PT} = Nu \frac{k_{air}}{l_{PT}}$$

and l_{PT} is a characteristic length for this thermal exchange, here the height of the surface, 10 cm.

The Nusselt number is expressed according to the empirical relation [1] : $Nu = 0.664\, Re^{1/2}$

The Reynolds number is expressed as above.

We find that h_{PT} is of the order of 20 W · m^{-2} · K^{-1}.

- **Radiative heat flux on the exposed face SPT**

$$E_{PT} = \varepsilon_g \sigma T_g^4 \quad + \quad \tau_g \sum_{j=1}^{j=6} F_{SPT,j} J_j$$

We use here the view factors $F_{SPT,j}$ to express the radiative contributions of the walls. Of course, the view factor of the specimen (behind the plate thermometer) is zero.

- **Heat balance of the plate thermometer**

$S_{PT}\left[h_{PT}\left(T_g - T_{PT}\right)+\varepsilon_{PT}E_{PT} - \varepsilon_{PT}\sigma T_{PT}^4\right]$ is the power (W) entering the plate thermometer on its exposed face, where S_{PT} is the area. Then : $\left[h_{PT}\left(T_g - T_{PT}\right)+\varepsilon_{PT}E_{PT} - \varepsilon_{PT}\sigma T_{PT}^4\right] = -k_{PT}\left(\frac{\partial T_{PT}}{\partial x}\right)_S$

where $k_{PT}\left(\frac{\partial T_{PT}}{\partial x}\right)_S$ is the conducted heat flux through the exposed face S, k_{PT} being the heat conductivity of the exposed plate. This latter term is expressed in writing the equation of heat diffusion under a discretised scheme, with boundary conditions at the unexposed face. As for thermocouples, this heat balance introduces different values for T_{PT} and T_g.

After a few minutes, $k_{PT}\left(\frac{\partial T_{PT}}{\partial x}\right)_S$ is very small if compared to the radiation terms in the left hand side. When T_g reaches a few hundreds of °C, the term $h_{PT}\left(T_g - T_{PT}\right)$ becomes negligible compared to e.g. $\varepsilon_{PT}E_{PT}$. Since the furnace is conducted so that T_{PT} be very close to T_{ISO}, *we obtain the very simple approximate result :* $E_{PT} \cong \sigma T_{ISO}^4$, where there is no reference to any characteristic of the furnace.

2.1.3 Incident heat flux on the exposed face of : walls, element of calibration, and specimen

The relations that are applied are formally the same as for the exposed face of a plate thermometer.

The net heat flux entering the specimen can be estimated in the simple following way :

$h_{conv}\left(T_{PT} - T_{spec}\right)+\varepsilon_{PT}\sigma\left(T_{PT}^4 - T_{spec}^4\right)$

assuming that $h_{conv} \cong h_{spec} \cong h_{PT}$. With our notation, this flux is :

$h_{conv}\left(T_g - T_{spec}\right)+\varepsilon_{spec}E_{spec} - \varepsilon_{spec}\sigma T_{spec}^4$ or,

with $E_{spec} = E_{PT} = \sigma T_{ISO}^4$: $h_{conv}\left(T_g - T_{spec}\right)+\varepsilon_{spec}\sigma\left(T_{ISO}^4 - T_{spec}^4\right)$

using $h_{PT}\left(T_g - T_{PT}\right) \approx 0$, we obtain the same equation as Wickström [16].

2.1.4 Incident heat flux on the unexposed face of : walls, element of calibration, and specimen

Similar relations are written with heat exchange coefficients adapted to the external conditions : T_{air} = 20 °C, natural convection on surfaces.

2.2 Algorithms and numerics

From T_{ISO} (t) given for the temperature of the sensor, all the other temperatures are calculated with the global system of equations of energy balance.

At time t= 0, all the temperatures are given the same value (20 °C). We use a time step D$t \approx 1$ sec.

At time t_i, = t_{i-1}+ Dt :

- The surfaces temperatures have been calculated at time t_{i-1}.
- T_g is given an estimated temperature, T_{est}, higher than the one of the time t_{i-1}. With this value T_{est}, the global system of equations is solved (with a 6th order Runge-Kutta solver). The

surfaces temperatures are then derived at $t = t_i$, as all the corresponding heat fluxes depending on T_{est}. From these values, the temperature of the sensor, T_{sens}, is then calculated (solving the balance on the sensor) and compared to T_{ISO}. According to the difference observed between T_{sens} and T_{ISO}, a new value is given to T_{est}, higher or lower than the previous one, and T_{sens} is calculated again. This iterative process is continued until T_{sens} is very close to T_{ISO} (± 0.1 °C). When the agreement is acceptable, the last T_{est} is kept as the actual temperature of the gas volume T_g and the values of all surfaces temperatures are stored.

3 EXPERIMENTAL CONDITIONS

3.1 The reference specimen [11]

Temperature is measured on five elements (called calibration elements) fixed on a concrete slab of 3 m 3m. Each element consists of two steel plates with ceramic fibre board insulation in-between (figure 2). Its dimensions are 290 mm x 290 mm. The exposed steel plate is 5 mm thick. The unexposed plate has a thickness of 2 mm. Two ceramic fibre boards, with a total thickness of 40 mm are placed between the plates. The thickness 40 mm is chosen for the purpose of giving temperature rise in the order of 140-180 °C at the unexposed side after 60 minutes of standard ISO 834 test.

The exposed and unexposed steel plates received each 2 thermocouples, 1 mm diameter, welded near the centre of hidden faces. In this report only the exposed face thermocouple data are given.

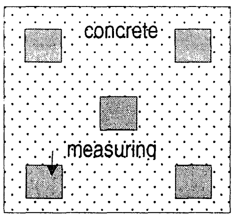

figure 3 : specimen assembly

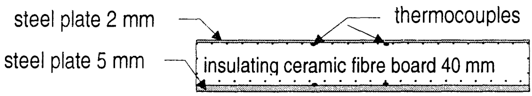

figure 2 : calibration element

Description of specimen assembly : Five measuring elements are placed on the vertical concrete wall (figure 3). Assembly wall was protected by 10 mm mineral insulating board.

Description of temperature and heat flux sensors :

4 plate thermometers were used for controlling the furnace for the « plate control » tests. In other case, they were used for measuring the furnace combustion gas temperature.

6 thermocouples of 1 mm diameter were used for controlling furnace for the « thermocouple control » tests. For the other tests, they were used for measuring the furnace gas temperature.

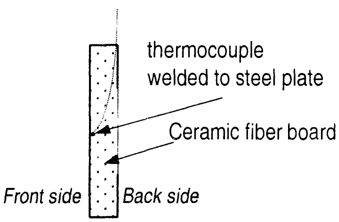

figure 4 : plate thermometer

Description of plate thermometer : The plate thermometer is made of one piece of a 0.7 mm thick steel plate (figure 4). The front size of the plate thermometer has a 100 mm by 100 mm area. It is insulated on the back side to prevent it from radiate impact from the specimen. It is placed in the furnace near the specimen with the front side facing the interior of the furnace.

4 CALCULATION RESULTS AND COMPARISON WITH EXPERIMENTAL DATA

4.1 CSTB vertical furnace piloted with thermocouples

The following variables were measured and calculated : the gas temperature measured with an aspiration thermometer (figure 5), the temperature of exposed side of calibration element (figure 6) and the radiative incident heat flux on the centre of the specimen (figure 7).

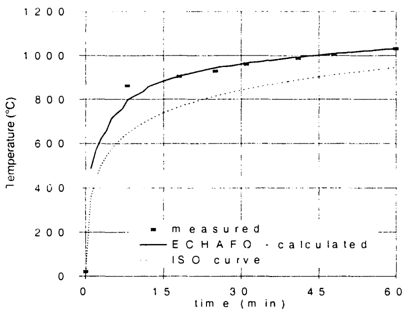

figure 5 : gas temperature and ISO curve

For the comparison, the experimental values of the gas temperature and of the temperature of the elements of calibration were averaged on the different locations of measurement.
For the calculation, the value of the gas velocity was supposed to be 3 m $\cdot$ s^{-1} and the absorption coefficient κ was set equal to 0.1 m^{-1} (low value corresponding to gas burners for this furnace).

The agreement is pretty good, except for the first ten minutes for which the measured values are higher than the calculated values.

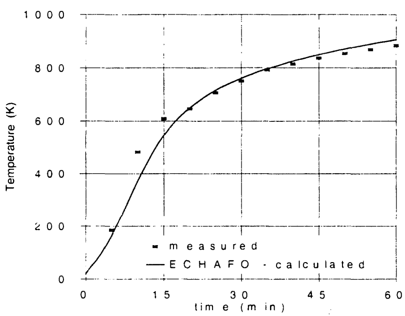

figure 6 : Temperature of the exposed side calibration element

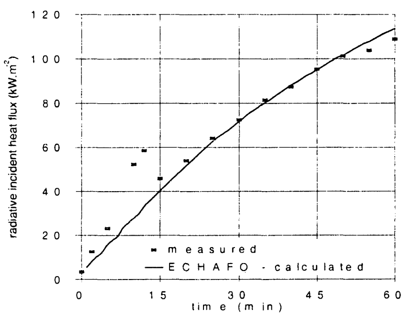

figure 7 : radiative incident heat flux on the specimen

4.2 Other three vertical furnaces

In [4], three vertical furnaces leading to visible differences in the thermal response of the reference specimen when piloted with 1mm thermocouples were used for a repetition of the tests with plate thermometers. When plate thermometers were the piloting sensors, a very good agreement was observed on the temperature of the exposed face of the elements of calibration.

Figure 8 shows the measured and calculated values for the two ways of piloting the furnaces. The "measured value" given on the figures correspond to an averaging on the recorded temperatures on the five elements.

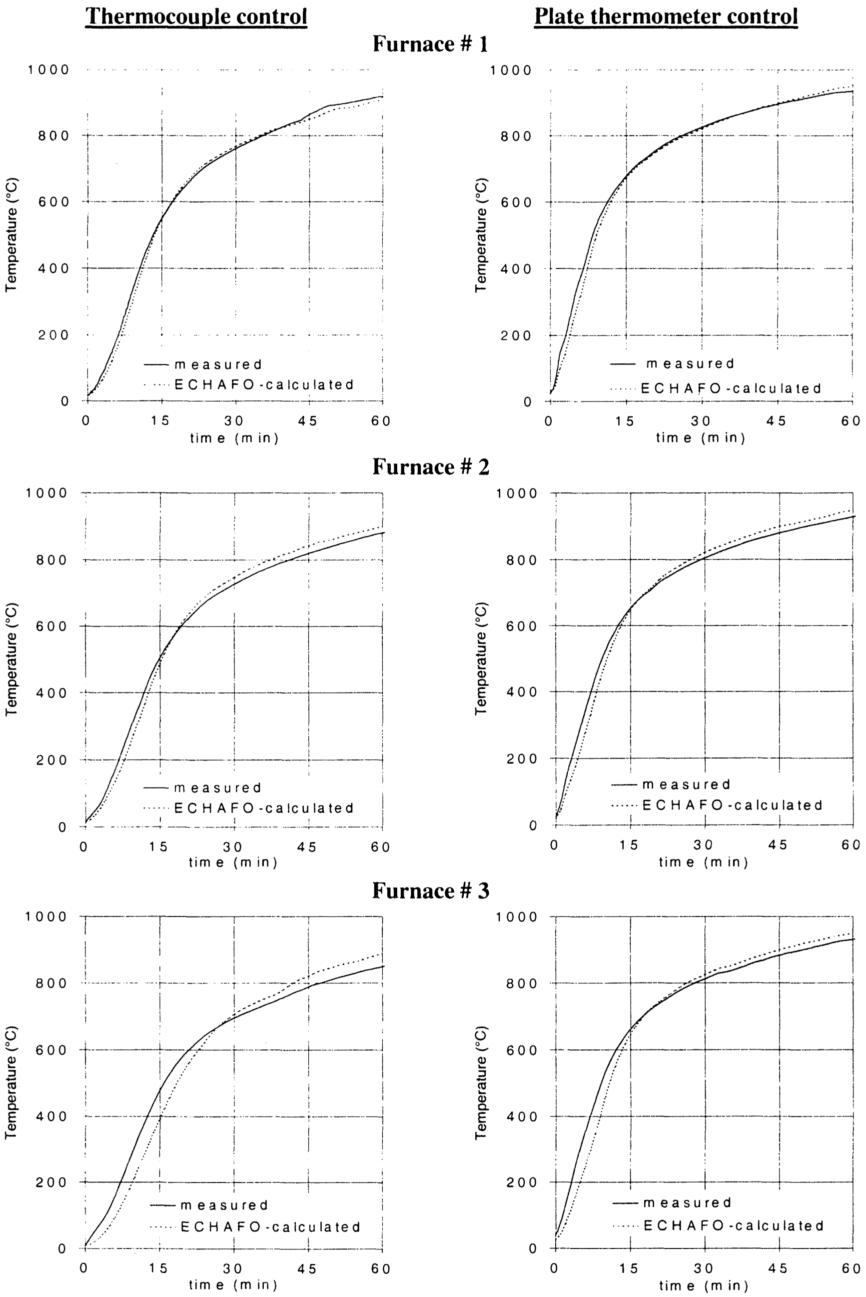

figure 8 : Measured and calculated temperature of exposed side of calibration element

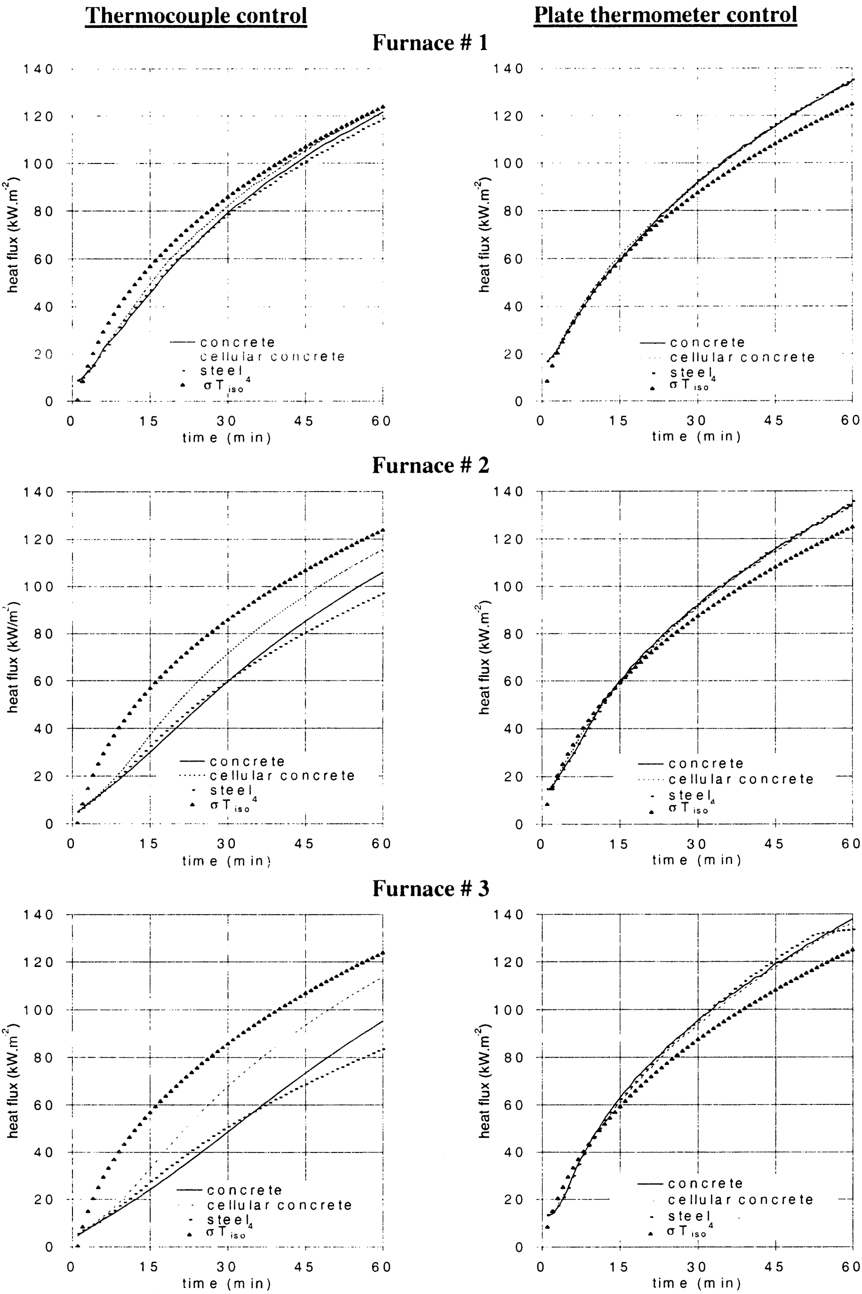

figure 9 : Calculated total incident heat flux on the exposed face of the specimen

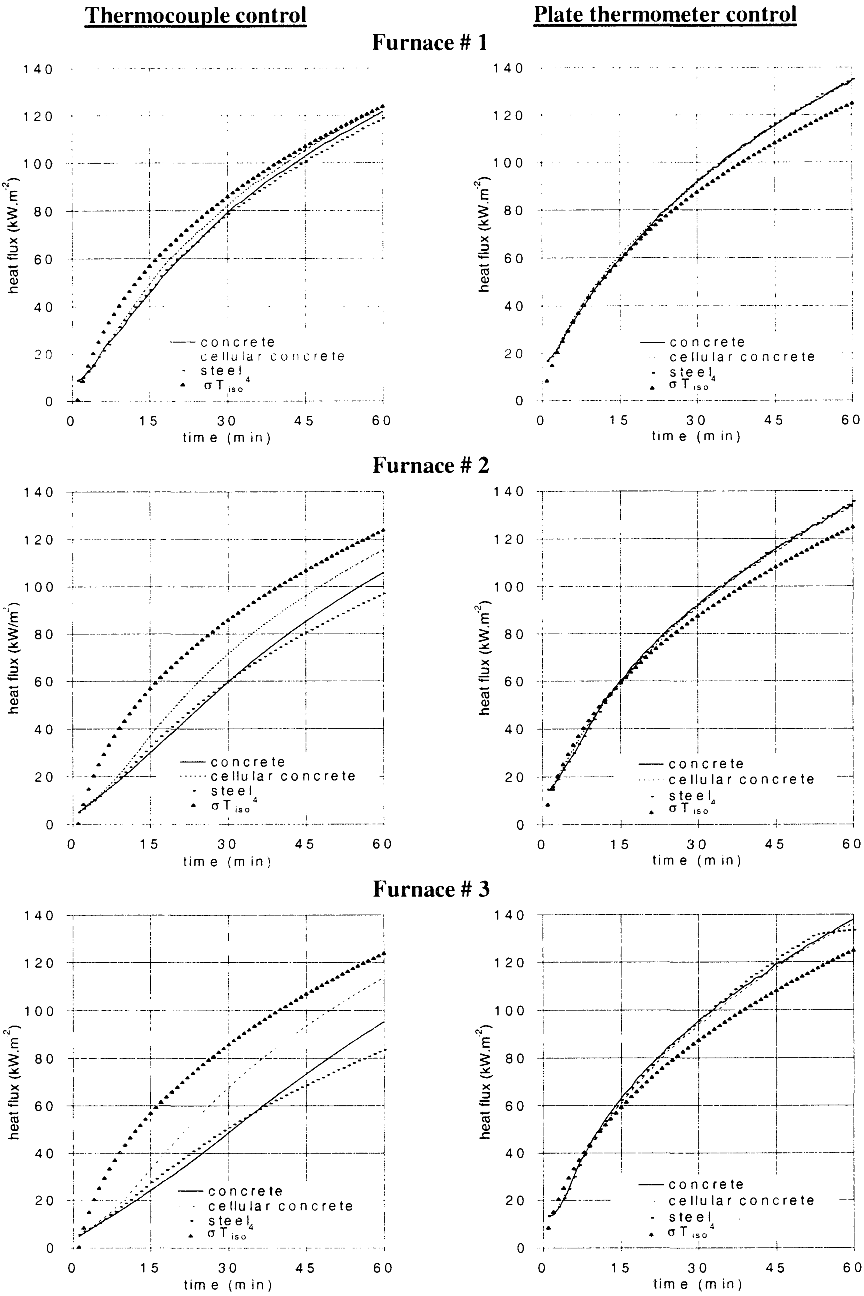

figure 9 : Calculated total incident heat flux on the exposed face of the specimen

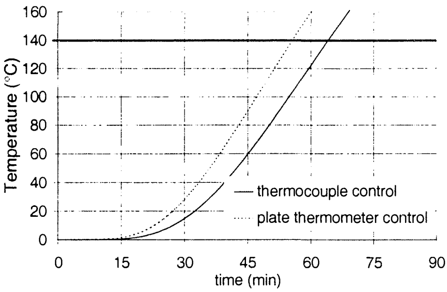

figure 11 : furnace # 2
unexposed side temperature of specimen

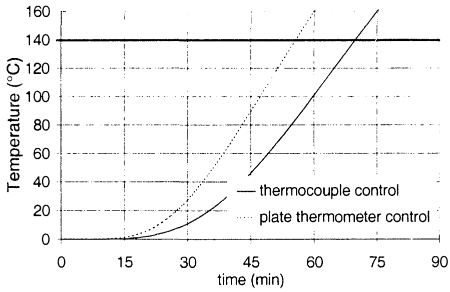

figure 12 : furnace # 3
unexposed side temperature of specimen

7 FINAL COMMENTS AND CONCLUSION

– The calculations made with ECHAFO, as some measurements of T_g with an aspiration thermometer [5], show that T_g is higher than T_{ISO}.
This observation is in full agreement with the calculation results presented in [8] and [14]. This point does confirm that T_g and the temperature of the sensor used for piloting the furnace have to be distinguished in a model.

– Other calculations with ECHAFO have shown [6] the influence of the furnace depth and the absorption coefficient κ on the heat flux received by the specimen and lead to the same comments as in [8] on the importance of these parameters.

– The results obtained with this zone model give unique zone values for e.g. the temperature of the elements of calibration. In the tests, five elements of calibration are used that are not receiving exactly the same heat flux. This point had been addressed by Cooke, [3], saying that (for a set of tests) the advantage of plate thermometers is more visible for the central element of calibration than for all of them. The tests results obtained in [4] are more in favour of the use of plate thermometer control. Nevertheless, this question is connected to the limits of zone models compared to CFD models.

– If tests are not conducted with the temperature of the piloting sensors following closely the ISO curve, the advantages of plate thermometer control can be strongly reduced, [10].

In conclusion, a simple zone model as ECHAFO can help to explain the advantages of plate thermometer control, is able to provide results close to test measurements, and may be used for approximate predictions for a given furnace, e.g. before executing a set of tests.

Acknowledgements

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