Energy simulation and design of a hot box suitable for dynamic tests of building envelope opaque components

Alessandro Prada – Free University of Bolzano, Bolzano, Italy
Davide S. Gigli – Free University of Bolzano, Bolzano, Italy
Andrea Gasparella – Free University of Bolzano, Bolzano, Italy
Marco Baratieri – Free University of Bolzano, Bolzano, Italy

Abstract

The aim of this work is to set up a procedure for the evaluation of dynamic thermal characteristics starting from experimental data collected by means of a hot box apparatus. The system involved is a calibrated and guarded hot box, available at the Free University of Bolzano, equipped with air cooling and heating systems capable of keeping stable conditions at the boundary of an opaque envelope structure. Since the experimental tests require dynamic boundary conditions (e.g. periodic temperature variations), an upgrade of the apparatus is then necessary. Consequently, a thermo-fluid dynamic model of the hot box apparatus is developed with the purpose of designing an effective improvement in the appliance. In addition, this model is also applied for the simulation of the hot box operation during the dynamic test and the numerical results are validated versus the experimental data. The numerical analysis also enables the correct understanding of the operation of the whole equipment. The paper firstly describes the methodology involved in the implementation of the numerical analysis and in the system design. Following on from this point, the experimental tests carried out on several timber components, subjected to periodic boundary conditions and the model validation, are presented.

1. Introduction

In the last few years the use of dynamic simulation has increased due to the purpose of better modelling the dynamic interaction between building and energy systems. A reliable evaluation of the thermal behaviour of building envelopes requires dynamic simulations to take into account their thermal capacity. An improvement of experimental methods to measure the dynamic thermal characteristics of the components is then needed to test calculations.

In this regard, while it is well known how to measure the steady-state thermal transmission properties - both regarding the standard procedures (EN 1934:1998) and in terms of the performance of different approaches (Asdrubali and Baldinelli, 2011) - there are no standard technical rules for the evaluation of dynamic parameters (e.g., periodic thermal transmittance and time shift) by means of experimental laboratory tests.

In the literature there are few examples of dynamic tests on envelope components carried out in the laboratory.

Ulgen (2002), has investigated the behaviour of opaque wall materials by means of a simulation unit, consisting of two volumes separated by the wall sample. One volume has adiabatic boundaries and no heat generation, while in the other volume a heat generation device generates a sinusoidal temperature signal. The aim of the study is to measure time lag and decrement factor for different wall compositions.

A calibrated guarded hot-box unit has been adjusted to measure the dynamic thermal properties of insulated brick walls (Sala et al., 2008). The forcing temperature follows a triangular signal of 10 °C amplitude in an interval of 2 hours. The results are compared with those obtained from a finite volume simulation. By means of the same apparatus also the response factors of a wall have been obtained (Martìn et al., 2010). The designed procedure does not require the measurement of the material properties.

Furthermore, the analysis of transient heat transfer data has also been applied to compare effective thermal transmittances of both isotropic and anisotropic building materials (Yesilata and Turgut, 2007). In particular, sample measurements with ordinary concrete and rubberized concretes have been performed in an adiabatic hot box apparatus.

The present work can be divided into two sections, the former relevant to the experimental tests carried out through a hot box apparatus equipped with a heat-flow meter and the latter regarding the system modelling validated through the collected data. The goal of the experimental tests is to describe the thermal behaviour of sample envelope components (i.e., section of a timber wall) under dynamic boundary conditions. The amplitude variations of the heat flux on the internal side and the amplitude variations of the surface temperature on the external side have been recorded. According to EN ISO 13786:2007 it has been possible to compute the delay time and decrement factor of the sample. A thermo-fluid dynamic model (based on finite element method, f.e.m.) of the hot box apparatus has been developed for the hot box design and for the simulation of the system operation during the dynamic test. The simulation results have been validated versus the experimental data.

2. Materials and methods

2.1 Hot box apparatus description

According to the EN 1934 standard, the hot box apparatus - designed and constructed for the purpose of the present research - is an insulated box (aluminium plates filled with 10 cm of polyurethane insulation) of a height of 170 cm, a width of 170 cm and a 110 cm length, with one side left open (Figure 1). A black screen separates the inner volume of the hot box from the test wall to avoid heat exchange by radiation between the hot box and the sample.

The box is equipped with an evaporator (the cooling unit), an electrical resistance (the heating unit) and a cylindrical horizontal fan, which generates a regular-shaped air flow stream on the surface of the component, also reducing the surface thermal resistance. The evaporator and the electrical resistance work together adjusted by a PID type regulation unit, tuned up in order to keep an internal constant temperature in a range of ± 0.05 °C with respect to the set point value. The tuning of the system allows for stable conditions and a satisfying uniformity of temperature on the sample surface, in the measurement area. Peripheral inhomogeneity is due to viscosity effects of the air flow near the hot box perimeter and to geometrical thermal bridges. According to the standard, these phenomena have to be assessed and if they are kept within the prescribed ranges they have negligible effects on the measured physical quantities.

For a steady state operation, the hot box is used in its standard configuration having two twin chambers kept at a constant temperature (usually 20°C and -10°C) in order to measure the thermal conductance (C_s) of the envelope component. For the transient operation one of the two chambers has been set up to keep steady-state conditions: the set point was at 20°C, with an air flow rate of 3.5 m s^{-1} and relative humidity that ranges between 45% and 65%.

An electrical heater has been used to impose the forcing temperature signal on one side of the test wall considered as the external side. The electrical heater is a 120 x 120 cm copper coil crossed by an electric current of 6.5 A, which can generate a nominal thermal power of 1500 W. At the present time the control system only allows an on/off behaviour of the heater, as a first step to simulate a dynamic forcing temperature on the external side of the component.

The hot box has been designed with reduced dimensions with respect to traditional hot box systems, in order to have a reduced mass and, thus, reducing the thermal inertia of the system. This allows for a higher precision in controlling the operation conditions and, consequently, more precise measurements of the dynamic response of a component due to imposed thermal variation.

The heat flow variation on the internal side of the component has been measured by the use of a heat-flow meter, namely a thermopile made of 250 type T thermocouples, which measures the heat flux on a component surface by means of the variation of electric potential through the two sides of the flow meter. The thermometric quantity, i.e., the electric potential in mV, is converted into Wm^{-2} through calibration values obtained by means of guarded hotplate tests, suitable for conductivity measurements.

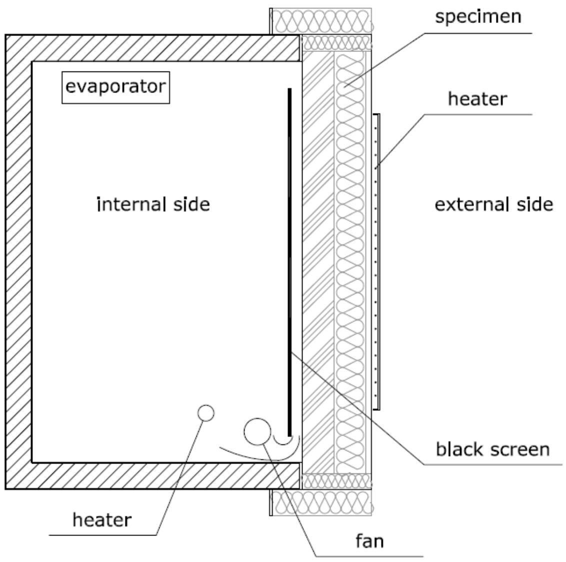

Fig. 1 – Hot box apparatus

The heat-flow meter has been surrounded by a guard ring having thermal properties similar to the heat flow meter. On the internal side of the component, 8 type T thermocouples measure the temperature on the layer between the inner side of the sample and the guard ring, a further 4 thermocouples measure the temperature in the layer between the inner side of the sample and the heat-flow meter. On the external side, there are 12 thermocouples in the same position of the opposite side, which measure the external surface temperatures. The thermocouples are placed 1 mm under the external surface of the component sample, to avoid the influence of direct radiation due to the electrical heater.

2.2 Experimental tests

The test envelope components are characterized by timber frame structures. Two different samples (i.e., sample n. 1 and 2) have been chosen. Table 1 and 2 show the main characteristics and physical properties of the samples. The sample geometry and structural characteristics have been reported in Figure 2. The internal side temperature (with respect to the test component, see Figure 2) is held at a constant value of 20°C by means of the hot box apparatus. The power of the inner electrical resistance is usually limited to 45%, to reduce the thermal variations inside the hot box, allowing for a quick convergence of the PID controller and consequently a finer tuning of the setup point. The air stream flow rate is set to 3.5 m s^{-1}, while the relative humidity is not controlled. After the time necessary to obtain the steady state conditions in the inner side (i.e., usually 24 hours) - also dependent on the balance with the temperature of the lab also set up to 20°C - the external side is then subjected to a 1500 W thermal radiation for 2 hours, followed by 22 hours of rest. The 24 period on/off thermal power signal is controlled by a programmable switch. The electrical heater is placed very close to the external surface, to minimize the influence of air buoyancy effects, which could cause temperature gradients between the lower and the higher part of the sample.

Material	Code	Density (kg m^{-3})	Conductivity (W m^{-1} K^{-1})
Gypsum-Fibreboard	CG	900	0.21
Low density wood fibre	F1	40	0.038
Low density wood fibre	F2	50	0.038
High density wood fibre	F3	160	0.040
Timber structure (beam)	W	450	0.13
Timber structure (pillar)	LW	450	0.13
Timber multilayer panel LVL	LV	530	0.20
Plaster	PL	1000	1.0

Table 1 – Wall samples materials library

The test was usually carried out over two weeks, in order to minimize the influence of the initial conditions due to thermal inertia of the specimen and thus ensuring the achievement of periodic steady state. The transient effects were observed to be negligible usually after 3 or 4 days starting from the first thermal forcing signal.Once reached the periodic steady state – i.e., when the amplitude of either the thermal flux or the temperature variations have constant amplitude between two periods - the heat flux on the internal side and the external surface temperatures were recorded and used for the further analysis.

Table 2 – Wall samples characteristics

code	thickness (cm)	n. sample
CG	2.5	1 - 2
F1	4	1 - 2
LW	16	1 - 2
F2	10	1
F1	5	2
W	10	1
LV	1.25	1 - 2
F1	4	1 - 2
W	4	1 - 2
LV	1.25	1 - 2
F3	8	1 - 2
PL	0.8	1 - 2

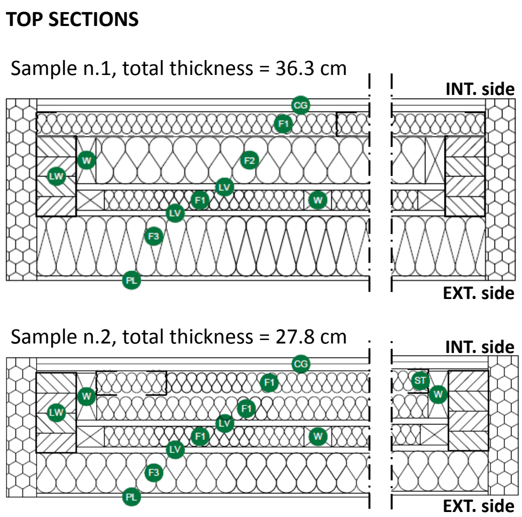

Fig. 2 – Sample geometry and structural characteristics

2.3 Dynamic parameters: experimental evaluation

A set of 4 days in a periodic steady state was chosen to record the external surface temperature (i.e., forcing temperature signal) and the internal heat flux. The Fast Fourier Transform (FFT) algorithm (Press et al., 2007) was applied, computing the first 720 harmonics of the experimental temperature and heat flux. The fundamental signals (first term of the Fourier approximation) were then used to compute the decrement factor and the time shift in agreement with the EN ISO 13786 standard. It is worth pointing out that in the calculation the surface thermal resistances have been not taken into account, thus considering thermal conductance instead of transmittance. This assumption is acceptable, as far as the internal surface temperature is approximately constant. In particular the periodic thermal conductance (1) has been computed as the ratio between the amplitude of the first harmonic of the internal heat flux ($\phi_{i,1}$) and the amplitude of the first harmonic of the external surface temperature ($\theta_{e,1}$). The decrement factor (f) has been then computed (2) dividing the obtained periodic thermal conductance by the thermal conductance relevant to the steady state (C_s), measured by means of the hot box in agreement with EN 1934.

$$C_{ie} = \frac{\phi_{i,1}}{\theta_{e,1}} \tag{1}$$

$$f = \frac{C_{ie}}{C_s} \tag{2}$$

The time shift (5) has also been calculated using the phase displacements (φ and ψ) of the fundamental signals.

$$\theta_e \approx \bar{\theta}_e + \theta_{e,1} cos(\omega t + \varphi) \tag{3}$$

$$\phi_i \approx \bar{\phi}_i + \phi_{i,1} cos(\omega t + \psi) \tag{4}$$

$$\Delta t_f = \frac{\psi - \varphi}{\omega} = \frac{\psi - \varphi}{2\pi} \times 24[h] \tag{5}$$

Both in the case of the decrement factor and of the time shift, the values of each day of the chosen set (4 days) and the average values have been computed.

2.4 Simulation model

With the purpose of modelling the behaviour of the hot-box apparatus when a dynamic test is carried out, an unsteady thermo-fluid dynamic model has been developed. This numerical analysis investigates the applicability of the hot-box upgrade appliances for the experimental estimation of the wall dynamic characteristics according to EN ISO 13786. In particular, the numerical model has been herein applied to a well-insulated timber wall (sample 1) when a sinusoidal variation of the wall surface temperature is imposed. The domain is discretized by means of an unstructured triangular mesh. Globally, 20268 triangular elements are used to discretize the whole domain with a greater thickening in the zone of the convective heat exchange between the air flow and the wall surface.

In order to model the behaviour of the turbulent system with a low computational cost, the Reynolds-averaged Navier Stokes (RANS) equations are used. Adopting the weakly compressible hypothesis (i.e. compressibility is taken into account only in continuity equation) and the Newtonian behaviour of the fluid, the system of partial differential equations describing the problem becomes:

<u>Conservation of mass for fluid</u>

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho u) = 0 \tag{6}$$

<u>Momentum equation with the eddie closure relation</u>

$$\rho\left(\frac{\partial u}{\partial t} + u \cdot \nabla u\right) = \nabla \cdot \begin{bmatrix} -pI + (\eta + \eta_T) \\ \left(\nabla u + (\nabla u)^T - \frac{2}{3}(\nabla \cdot u)I\right) - \frac{2}{3}\rho k I \end{bmatrix} + \mathrm{F} \tag{7}$$

<u>Heat equation for solid domain</u>

$$\rho c_p \frac{\partial T}{\partial t} + \nabla \cdot (-\lambda \nabla T) = 0 \tag{8}$$

<u>Heat equation for fluid domain</u>

$$\rho c_p \frac{\partial T}{\partial t} + \nabla \cdot (-(\lambda + \lambda_T)\nabla T) = -\rho c_p u \nabla \mathrm{T} \tag{9}$$

<u>Transport equation of the turbulent kinetic energy</u>

$$\rho\left(\frac{\partial k}{\partial t} + u \cdot \nabla k\right) = \nabla \cdot [(\eta + \eta_T)\nabla k] + $$

$$+\frac{1}{2}\eta_T(\nabla u + (\nabla u)^T)^2 - \rho\varepsilon \tag{10}$$

<u>Transport equation of the turbulent energy dissipation rate</u>

$$\rho\left(\frac{\partial \varepsilon}{\partial t} + u \cdot \nabla \varepsilon\right) = -\rho C_{\varepsilon 2}\frac{\varepsilon^2}{k}\nabla \cdot \left[\left(\eta + \frac{\eta_T}{1.3}\right)\nabla \varepsilon\right] + $$

$$+\frac{1}{2}C_{\varepsilon 1}\frac{\varepsilon}{k}\eta_T(\nabla u + (\nabla u)^T)^2 \tag{11}$$

where η_T and λ_T are respectively the eddy viscosity and the eddy conductivity computed as:

$$\eta_T = \rho C_\eta \frac{\varepsilon^2}{k} \tag{12}$$

$$\lambda_T = \eta_T c_p Pr_t \tag{13}$$

where Pr_t is the turbulent Prandtl number computed with the Kays-Crawford relation. In particular, the values of the constant adopted in the $k - \varepsilon$ model are reported in Table 3.

C_η	$C_{\varepsilon 1}$	$C_{\varepsilon 2}$
0.09	1.44	1.92

Table 3 – Constant adopted in the k-ε model

Since the momentum equations are nonlinear, the solution of the coupled heat and fluid dynamic problem become unstable if the Galerkin finite element method is adopted for the spatial discretization of the domain (Hauke, 2002). Consequently, in all the turbulent simulations herein reported an artificial diffusion is introduced. The diffusion is required in order to ensure the numerical convergence of the problem solution. However, the artificial diffusion parameter is kept to as low a value as possible while still getting a converged simulation. The key aspect in the numerical model of the hot box behaviour is the definition of the initial and boundary conditions. While the steady state solution at the time t=0 is imposed as initial conditions, several boundary conditions are defined both for heat transfer and RANS equations. In particular, while for the solid domain the Newton-Robin equation with the convective heat coefficients defined by EN ISO 6946 are used, Table 4 shows the BC applied for the ventilation channel.

INLET	OUTLET	WALL
$u = u_0$	$p = p_0$ no normal stress	Logarithmic wall function
$k = \tilde{k}(t)$	-	-
$\varepsilon = \tilde{\varepsilon}(t)$	-	-
$T = \tilde{T}(t)$	$\hat{n} \cdot (\lambda \nabla T) = 0$	Robin –Newton with radiative exchanges

Table 4 – BC of the ventilation cavity

3. Results and discussion

3.1 Experimental tests

The first experimental runs have been carried out in standard steady state conditions with the complete double hot box, in order to measure the thermal conductance (i.e., EN ISO 1934) of the two envelope components (Table 5).

The two samples have then been tested under dynamic conditions. An example of the measurements performed on sample n.1 is reported in figure 3, where the external forcing temperature and the internal heat flux have been plotted versus time. In the same figure also the reconstruction obtained through the FFT procedure (i.e., using 720 harmonics) has been pointed out and the first term of the approximation (i.e., the fundamental signal). The decrement factor and time shift have then been computed using the amplitudes and phase displacements of the fundamentals signals of temperature and heat flux. Table 6 shows the values of the dynamic parameters obtained for each day of the chosen set and the relevant average value.

Sample	Thermal conductance (W m^{-2} K^{-1})
n.1	0.172
n.2	0.240

Table 5 – Thermal conductance of the samples, measured in agreement with EN ISO 1934

	C_{ie} (W m^{-1} K^{-1})	f (-)	Δt_f (hours)
Sample n.1			
Day 1	0.0326	0.1894	8.82
Day 2	0.0269	0.1566	9.20
Day 3	0.0311	0.1806	9.40
Day 4	0.0346	0.2012	9.32
Average	*0.0312*	*0.1819*	*9.18*
Sample n.2			
Day 1	0.1029	0.4288	6.65
Day 2	0.1004	0.4183	6.75
Day 3	0.1018	0.4242	6.67
Day 4	0.0997	0.4213	6.87
Average	*0.1011*	*0.4215*	*6.73*

Table 6 – Dynamic parameters obtained from the experimental measurements (4 days set)

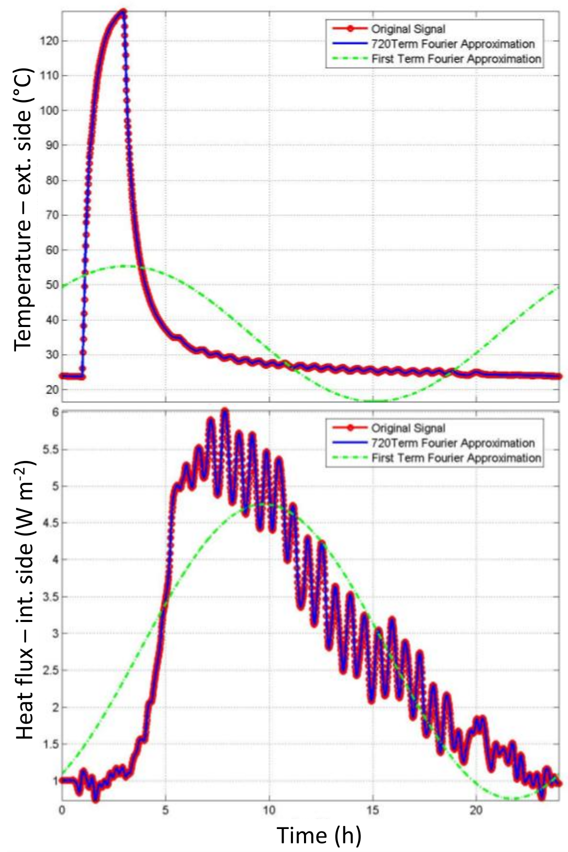

Fig. 3 – Surface temperature (ext. side) and heat flux (int. side) recorded on the first day of periodic steady state - sample 1

3.2 Model test

The finite element model has been applied to the sample no.1 case. The simulation was carried out during the same period of the measured test obtaining a comparison on the same time basis (day 1 to 4).

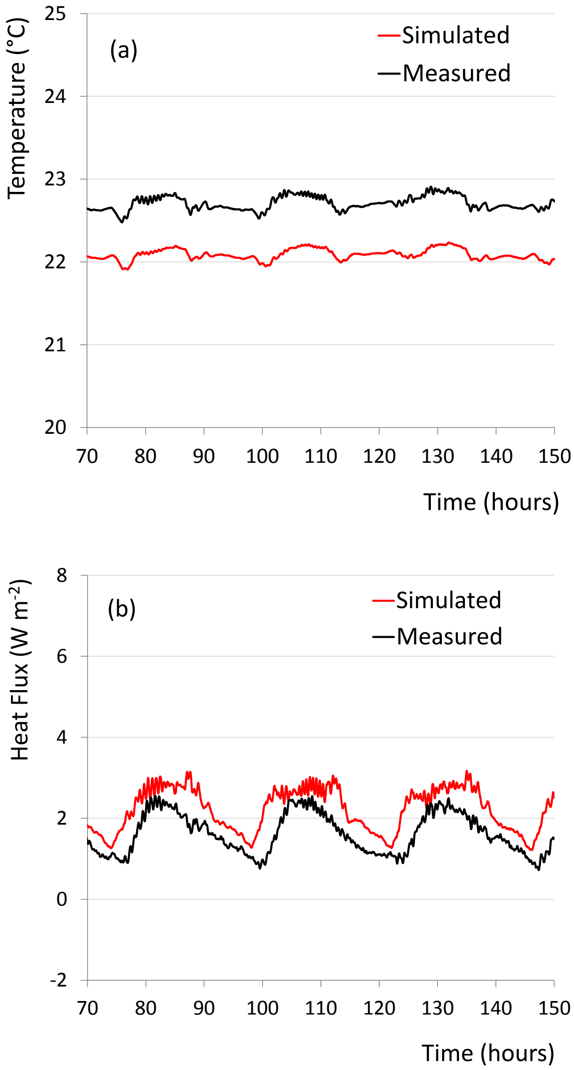

Fig. 4 – Surface temperature (a) and heat flux (b) on the wall internal side: comparison between simulated and measured values (sample n.1)

Figure 4 shows the comparison between the simulated and measured surface temperature and heat flux on the internal side of the test wall (the signals have been subjected to a proper filtering). Although the shape and the trend of the curves show a satisfying agreement, a constant deviation has been observed between the measured and simulated signals. The same behaviour is consequently obtained, and further clarified, by computing the first harmonic of the simulated heat flux when compared with the measured one (figure 5). The detected constant deviation (i.e., measured Vs simulated signal) can be explained on a purely steady state basis, since the periodic signals show a very similar phase during the tests. In particular, the average difference of temperature is approx. 0.6 K, while the average deviation of the heat flux has been assessed at 0.5 W m^{-2}.

The values of the periodic thermal conductance, decrement factor and time shift have been computed using the first harmonic of the simulated heat flux and of the measured forcing temperature. As for the measured values, the calculation has been carried out on a 4-day basis also obtaining the average values (table 7). The comparison between the average values is shown in figure 6.

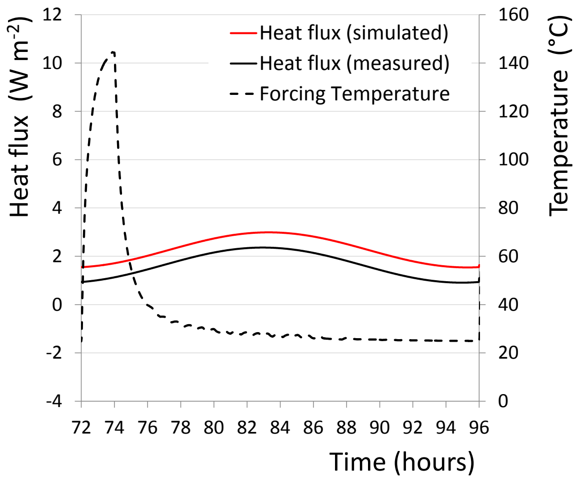

Fig. 5 – First harmonic signals of heat flux on the wall internal side: comparison between simulated and measured values (sample n.1)

	C_{ie} (W m^{-1} K^{-1})	f (-)	Δt_f (hours)
Sample n.1			
Day 1	0.0299	0.1738	8.68
Day 2	0.0303	0.1763	8.91
Day 3	0.0219	0.1275	10.35
Day 4	0.0188	0.1092	10.75
Average	*0.0252*	*0.1467*	*9.67*

Table 7 – Dynamic parameters obtained by means of the FEM model (4 days set, sample n.1)

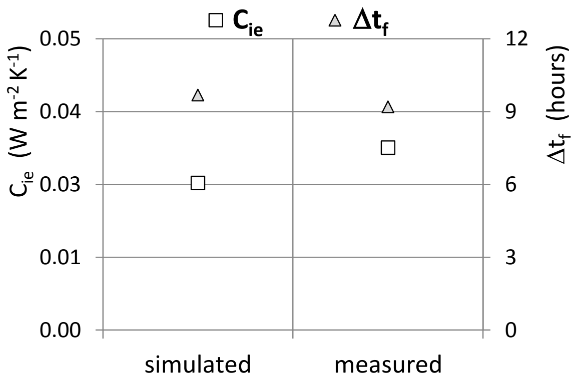

Fig. 6 – Periodic thermal conductance and time shift: comparison between simulated and measured data (4 days set, sample n.1)

3.3 Remarks

The calculation of the decrement factor and time shift using the experimental data lies on the assumption that it is possible to consider a periodic thermal conductance instead of a transmittance. This has been considered an acceptable assumption, since the definition of the periodic thermal conductance implies a constant internal surface temperature, which it has been measured to be in the range of 0.5°C (i.e., approx. 1% of the first harmonic amplitude relevant to the external forcing temperature). Further analysis will foresee a set of two runs per sample - using forcing temperature signals of different amplitude and shape - in order to be able to estimate the decrement factor and the periodic thermal transmittance starting from the heat transfer matrix as defined in EN ISO 13786.

The deviations resulting between the measured and the simulated surface temperature and heat flux (on the wall internal side) deviations can be attributed to steady state effects. In particular a calibration of the model has to be carried out as a further development, for example, tuning the thermo-physical properties of the wall materials or using a 2D-time dependent air velocity profile as boundary condition in the hot box cavity.

4. Conclusion

A hot box laboratory apparatus has been upgraded and used for the dynamic testing of timber opaque components. Two types of timber components have been tested using a time dependent temperature signal. The decrement factor and the time shift have been computed using the measured heat fluxes. The experimental data have then been compared with the results of a simulation carried out by means of a thermo-fluid dynamic model. The resulting comparison is satisfying, even if a further calibration of the model is needed to achieve a better agreement.

The developed model seems to be a suitable tool for the further planning of the research activity. In particular, by applying a sinusoidal forcing temperature it will be possible to validate the results obtained using only the first harmonic and also to point out a range of applicability of this procedure with varying wall characteristics.

An assessment of the periodic thermal transmittance and of the time shift will also be possible only through the experimental data, by performing a series of tests (i.e., two at least) on the same wall sample, using different forcing signal (e.g., using different amplitudes) and solving the equation system with the transfer matrix as reported in Annex B of EN 13786.

5. Nomenclature

Symbols

C_{ie}	Periodic thermal conductance of the component[W m^{-2} K^{-1}]
C_s	Thermal conductance [W m^{-2} K^{-1}]
F	Volume force vector [N m^{-3}]
I	Identity tensor [-]
T	Temperature [K]
c_p	Specific heat [J kg^{-1} K^{-1}]
F	Decrement factor [-]
k	Turbulent kinetic energy
$\hat{n}$	Surface normal versor [-]
p	Pressure [Pa]
t	time [s]
u	Vector of velocity [m s^{-1}]

Greek

ϕ	Specific heat flux [W m^{-2}]
ε	Turbulent kinetic energy dissipation rate [-]
φ	Phase difference [rad]

η	Dynamic viscosity [Pa s]
λ	Thermal conductivity [W m^{-1} K^{-1}]
θ	Temperature [°C]
ρ	Specific mass [kg m^{-3}]
ω	Angular frequency [rad s^{-1}]
ψ	Phase difference [rad]

Subscripts/Superscripts

0	initial condition
1	the first harmonic of the signal
i	internal side of the sample
e	external side of the sample
T	referred to turbulent quantity

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