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A dynamic model for the thermal-hygrometric simulation of buildings

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Abstract: Dynamic models for the energy simulation of building-plant systems are becoming useful tools in the process of building design by defining operating conditions and finding appropriate control strategies. Therefore simple softwares able to correctly predict the thermal behaviour of rooms, and thus allowing to get comfort conditions and loads, are needed. In this paper a dynamic simulation model (THESIS, "THermal SImulation Software") is presented. The building structures equations are described by means of a LTI (Linear Time Invariant) state space model. The heat conduction equations for the walls are solved through an explicit finite difference technique. The model is implemented in the MATLAB/SIMULINK environment.

1. INTRODUCTION

Computer modelling is becoming a common practice to investigate different solutions in Heating, Ventilation, and Air Conditioning (HVAC) systems design and to optimize sizes, performances, and controls of systems and components. Different models and software are available today, with different degrees of complexity and sensitivity to the involved physical parameters. The suitability of a model depends, first of all, on the complexity of the required calculation. For sizing winter heating systems, a steady-state model can be sufficient, whereas for sizing cooling systems, a dynamic model is required. A dynamic model is needed also for evaluating seasonal energy consumption both in heating and cooling conditions. Also, some buildings (such as open-space environments or technological building as web-hotels) may have the simultaneous presence of cooling and heating loads (Bettella *et. al* [2001]). In this case, a dynamic model to study the overall building-plant system is needed to optimize energy consumption.

A well-known example of dynamic model of buildings is provided by the ASHRAE method (ASHRAE [1997], Mitalas [1972]) which is a non-geometrical model based on the notion of transfer function. Other known models are the geometrical NBSLD (Kusuda [1976]) and its successive developments (Brunello *et al.* [2001]) based on the solution of the thermal balance problem and on the transfer function technique for thermal conduction through walls (Stephenson and Mitalas [1971]). Such models aim at determining the patterns of the thermal flux to be supplied to a room to maintain a given set-point temperature or estimating the room temperature, for a given flux. No standard method exists, yet, and discussion on this point is still open inside Standardization Committees, such as the European Committee for Standardization, CEN (Technical Committe 89, Working Group 6).

In this work a MATLAB model for thermal simulation of building is presented, which is based on a discrete-time, LTI (Linear Time Invariant) state space model. The advantage of such solution over other existing techniques is that the explicit, recursive nature of the model allows to substantially increase the efficiency of the simulation. The equations of the thermal room balance and conduction through walls are solved through the finite difference technique.

2. THEORY

The thermal-hygrometric simulation of a building can be carried out at different levels of complexity. A sufficient level of detail can often be obtained by using models where the room air temperature and humidity are considered uniform. In this case, the dynamics of each room is described in terms of the air energy and mass equation and the walls energy equation. As far as the energy equations are concerned, many contributions have to be taken into account, such as he convective, conductive and internal heat fluxes, the mutual radiation between surfaces, solar radiation, the presence of the heating/cooling system, the infiltration and ventilation air flow rates. In some building typology, with relevant room dimensions or large amount of glazing surfaces, a different approach can be considered.

2.1 Energy and mass balance of the room air

The model is based on the assumption of uniform room air (no stratification). Therefore, for each room there is one instantaneous value of the internal air temperature

a	thermal diffusivity, m^2/s
c_p	specific heat, J/kgK
c_s	shading coefficient
F	view factor
h	heat transfer coefficient, $\mathrm{W/m^2K}$
I	solar incident and diffuse radiation, $\mathrm{W/m^2}$
k	thermal conductivity, W/mK
$\dot{m}$	mass flow, kg/s
N_{wl}	number of walls
N_{wn}	number of windows
n	nondimensionalized time
Q	heat release rate, W
q	heat flux, $\mathrm{W/m^2}$
t	time, s
t_c	sampling time, s
S	surface, $\mathrm{m^2}$
$SHGF$	solar heat gain factor, $\mathrm{W/m^2}$
T	temperature, K
T_m	mean temperature, K
U	global transmittance, $\mathrm{W/m^2}$
V	volume, $\mathrm{m^3}$
x	air specific humidity, g/kg
α	solar absorption
Δs	layer thickness, m
ϵ	solar emissivity
ρ	density, $\mathrm{kg/m^3}$
ψ	coefficient, $\rho_{ia} c_{ia} V/t_c$,$\rho_w c_w S \Delta s/t_c$, W/K
σ_n	Stefan-Boltzmann constant,$5.67 \cdot 10^{-8}$, $\mathrm{W/m^2K^4}$
θ	coefficient, $\rho_{ia} V/t_c$, kg/s
Subscripts	
c	convection
ea	external air
ia	internal air
$ichg$	internal convective heat gain
inf	infiltration air
$irhg$	internal radiant heat gain
ivg	internal vapor gain
op	operative
p	heat production system
r	infrared radiation
s	solar radiation
va	ventilation air
w	wall

Table 1. Symbols used throughout the paper.

and humidity, described by one energy and one mass balance equation, respectively. The energy equation for the internal air is as follows

$$\rho_{ia}(t)\, c_{ia}(t)\, V\, \frac{dT_{ia}}{dt} = Q_c(t) + Q_{va}(t) + Q_s(t) + Q_{ichg}(t) + Q_{inf}(t)\ , \quad (1)$$

where the convective heat flux is expressed by:

$$Q_c(t) = \sum_{i=1}^{N_{wl}} h_{ia,i}\, S_i\, [T_{w,i,1}(t) - T_{ia}(t)]\ . \quad (2)$$

The infiltration and ventilation heat fluxes are instead:

$$Q_{inf}(t) = \dot{m}_{inf}(t)\, [c_{ea}(t)\, T_{ea}(t) - c_{ia}(t)\, T_{ia}(t)] \quad (3)$$

$$Q_{va}(t) = \dot{m}_{va}(t)\, [c_{va}(t)\, T_{va}(t) - c_{ia}(t)\, T_{ia}(t)]\ . \quad (4)$$

The mass equation for the internal humidity is given by

$$\rho_{ia}(t)\, V\, \frac{dx_{ia}}{dt} = \dot{m}_{va}\, (x_{va}(t) - x_{ia}(t)) + \dot{m}_{inf}\, (x_{ea}(t) - x_{ia}(t)) + \dot{m}_{ivg}(t)\ . \quad (5)$$

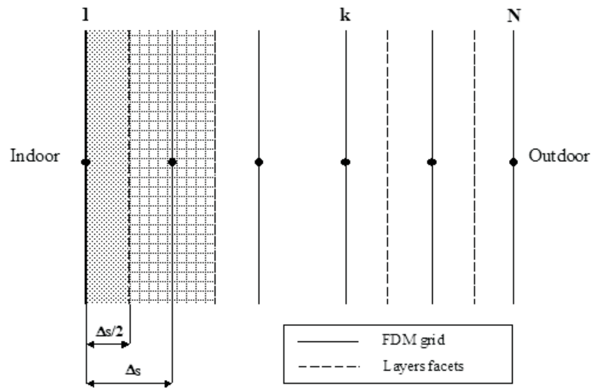

Fig. 1. Finite difference scheme for a single material wall.

2.2 Energy balance of the wall

In order to correctly solve the one-dimensional energy equation for the walls and to take into account thermal conduction and capacity phenomena, the finite difference scheme has been applied. In Figure 1, the mesh applied to a wall made of a single material is shown. The mesh splits the wall into a certain number of grid points. The same number of wall layers can be obtained if they are defined with facets located midway between the grid points. In the 1-D scheme, this approach leads to grid points located into the geometric center of each layer and allows to associate with each grid point the equivalent mass of its surrounding layer. Boundary grid points makes exception because they only have "half" layer and thus half mass capacity. In the case of walls made of more than one material, the same scheme applies. The grid actually passes through the interfaces between materials, thus layers composed of two different materials are present. In this case grid points may not appear in the center of these "two-material" layers, if a different mesh is applied to the materials.

The equation for the k-th point associated to a single material layer of the i-th wall is the following:

$$\rho_{w,i}(t)\, c_{w,i}(t)\, S_i\, \Delta s_i\, \frac{dT_{w,i,k}}{dt} = S_i\, \frac{k_{w,i,k}}{\Delta s_i}\, [T_{w,i,k+1}(t) - T_{w,i,k}(t)] + S_i\, \frac{k_{w,i,k}}{\Delta s_i}\, [T_{w,i,k-1}(t) - T_{w,i,k}(t)]\ . \quad (6)$$

For boundary grid points in contact with the internal air (1-st grid point), the equation (6) becomes:

$$\rho_{w,i}(t)\, c_{w,i}(t)\, S_i\, \frac{\Delta s_i}{2}\, \frac{dT_{w,i,1}}{dt} = h_{ia,i}\, S_i\, [T_{ia}(t) - T_{w,i,1}(t)] + S_i\, \frac{k_{w,i,1}}{\Delta s_i}\, [T_{w,i,2}(t) - T_{w,i,1}(t)] + Q_{r,i}(t) + Q_{s,i}(t) + Q_{irhg,i}(t)\ , \quad (7)$$

where the term associated with the heat exchange between internal surfaces via infrared radiation, in the hypothesis of grey bodies with emissivity close to one, is computed as:

$$Q_{r,i}(t) = S_i \sum_{j=1}^{N_{wl}-1} F_{ij}\, 4\, \sigma_n \epsilon_i\, T_m^3 \left[T_{w,j,1}(t) - T_{w,i,1}(t)\right] \quad (8)$$

and the solar radiation heat flux contribute is computed for each window using the ASHRAE method (ASHRAE [1997]):

$$Q_{s,i}(t) = \frac{S_i}{\sum_{i=1}^{N_{wl}} S_i} \sum_{k=1}^{N_{wn}} S_k\, c_{s,k}(t)\, SHGF_{s,k}(t)\,. \quad (9)$$

Finally, for the N-th grid point in contact with the external air, the general equation (6) becomes:

$$\begin{aligned} &\rho_{w,i}(t)\, c_{w,i}(t)\, S_i\, \frac{\Delta s_i}{2} \frac{dT_{w,i,N}}{dt} \\ &= h_{ea,i}\, S_i \left[T_{ea}(t) - T_{w,i,N}(t)\right] \\ &+ S_i\, \frac{k_{w,i,N}}{\Delta x_i} \left[T_{w,i,N-1}(t) - T_{w,i,N}(t)\right] + S_i\, \alpha_{w,i}\, I_i(t)\,. \quad (10) \end{aligned}$$

3. STATE SPACE MODEL OF THE ROOM

The physical quantities to be modeled are naturally described by continuous-time signals, however, the approach adopted in Section 2 to formulate the heat transfer equations allows to easily cast the problem in terms of discrete-time signals. In fact, the heat transfer equations derived by means of the finite difference method can be naturally thought as *difference* equations with time step $t_c = \Delta t$, whereas standard discretization schemes can be used to deal with the differential equations for the internal air energy and mass equation.

In this Section we show how to recast the system equations presented in Section 2 into a discrete-time, linear, time-invariant (LTI) *state space* model. Such a choice gives computational advantages, and allows for an efficient implementation in the Matlab-Simulink environment.

To build the discrete-time, LTi state space model of the room, we choose the state vector as follows. As the first two states we take the air temperature T_a and the operative temperature, T_{op}, defined as the average between the air and mean radiant temperature of the internal surfaces. Then, assuming that the room consists of N_{wl} walls and that the i-th wall is discretized into N_i layers, we can define a set of N_i temperature levels (see 1). A state variable is associated with each of such temperatures. The last state variable is associated with the internal air specific humidity x_a. Therefore, $\mathbf{x}(t)$ is as follows:

$$\mathbf{x}(n) = \Big[T_a(n) \quad T_{op}(n) \quad \overbrace{T_{11}(n) \quad T_{12}(n) \quad \dots \quad T_{1N_1}(n)}^{\text{1-st wall}} \quad \dots \quad \overbrace{T_{n1}(n) \quad T_{n2}(n) \quad \dots \quad T_{nN_t}(n)}^{n\text{-th wall}} \quad x_a(n) \Big]^\top .$$

The total number of state variables, that gives the system dimension ns, is given by

$$ns = 3 + \sum_{i=1}^{N_{wl}} N_i\,. \quad (11)$$

By taking as input vector

$$\begin{aligned} \mathbf{u}(t) = [&T_{ea}, x_{ea}, T_{va}, x_{va}, \dot{m}_{ivg}\,, \\ &Q_{ichg}, Q_{irhg}, Q_{ichg}, Q_s, I, SHGF]^T \end{aligned} \quad (12)$$

it is immediate to show that the equations described in the previous subsection can be written as

$$\begin{cases} \mathbf{x}(t+1) = A\mathbf{x}(t) + B\mathbf{u}(t) \\ \mathbf{y}(t) = C\mathbf{x}(t) + D\mathbf{u}(t) \end{cases}, \quad (13)$$

where the output vector $\mathbf{y}(t)$ is formed by appropriate subsets of state and input variables. Therefore the system is linear and time-invariant, and it is complitely specified by the quadruple of matrices (A, B, C, D), where, if ns, b, and c are respectively the number of state, input, and output variables,

$$A \in \mathbb{R}^{ns \times ns} \; B \in \mathbb{R}^{ns \times b} \; C \in \mathbb{R}^{c \times ns} \; D \in \mathbb{R}^{c \times b}\,. \quad (14)$$

If we take as output variables all the components of the state vector and all the input variables, which is a reasonable choice for off-line processing of the simulation data, the C and D matrices in (13) are

$$C = \begin{bmatrix} \mathcal{I}_{ns} \\ 0_b \end{bmatrix}, \qquad D = \begin{bmatrix} 0_{ns} \\ \mathcal{I}_b \end{bmatrix} \quad (15)$$

where $\mathcal{I}$ denotes the identity matrix.

4. A SYSTEM THEORETIC PROOF OF THE FOURIER CONDITION

As is well known (Patankar [1980]), convergence of an explicite FDM scheme, applied to the Fourier equation of the heat conduction, requires that the Fourier condition

$$\text{Fo} = \frac{k\, t_c}{c_p \rho \Delta s^2} \le \frac{1}{2} \quad (16)$$

be satisfied. Consequently, the sampling time t_c, for a given value of the spatial discretization interval Δs in a material with thermal diffusivity a, has to satisfy the following condition

$$t_c \le \frac{1}{2} \frac{c_p \rho \Delta s^2}{k}\,. \quad (17)$$

It is interesting to observe that the Fourier condition (16) corresponds to a condition on the asymptotic stability of the state-space system representing the discretized equation for the temperatures of the inner layers of a homogeneous wall (see (6)). In fact, the dynamics of the temperatures $T_{ij}(t)$, $j = 2, \dots, N_i$ of the N_i-th wall are described by a state space model with state matrix A given by

$$A = \begin{bmatrix} 1-2k & k & 0 & \dots & 0 \\ k & 1-2k & k & \dots & 0 \\ \dots & \dots & \dots & \dots & \dots \\ \dots & \dots & \dots & 1-2k & k \\ \dots & \dots & \dots & k & 1-2k \end{bmatrix} \quad (18)$$

where, for notational convenience, $k = Fo$. It can be shown (see e.g. Rocha and Zampieri [1995]) that the eigenvalues of the $(N_i - 1) \times (N_i - 1)$ matrix A are given by

$$\lambda_\ell = 1 - 2k\,(1 - \cos\frac{\ell\pi}{N_i}), \quad \ell = 1, \ldots, N_i - 1\,. \tag{19}$$

The system is asymptotically stable if and only if $|\lambda_\ell| < 1$, $\ell = 1, \ldots, N_i - 1$, that is,

$$\left|1 - 2k\,(1 - \cos\frac{\ell\pi}{N_i})\right| < 1\,, \ \ \ell = 1, \ldots, N_i - 1 \tag{20}$$

or equivalently

$$0 < k < \frac{1}{1 - \cos\left[\ell\pi/N_i\right]}\,, \ \ \ell = 1, \ldots, N_i - 1\,. \tag{21}$$

The most restrictive bound is achieved when

$$\cos\frac{\ell\pi}{N_i} \simeq -1 \tag{22}$$

that yields the *conservative* bound corresponding to the Fourier condition (16)

$$0 < k < \frac{1}{2}\,. \tag{23}$$

In fact, for a given dimension of the state-space system, an exact bound is provided by (21), which is however close to (23) when the system dimension $N_i - 1$ is sufficiently large.

It has to be remarked that the state matrix A for the generic room structure is not symmetric tridiagonal as in (18), due to the presence of terms accounting for the convective heat exchange, infrared mutual and high-frequency radiation for each wall. Furthermore, the equations for the first two states, namely T_{ia} and T_{op}, have a different structure. However, A can be considered to be a perturbation of a tridiagonal, symmetric matrix, and this is confirmed by the fact that in all the tested situations, the spectrum of A consists of real eigenvalues only.

5. VALIDATION TESTS

In this section results of some of the tests carried out to validate the model are reported. The solution of the Fourier conduction equation for the single wall, the energy balance of the room in steady-state conditions and the dynamic behavior both of the inside air and the building envelopment are analyzed.

5.1 The conduction equation

The Fourier thermal conduction equation is solved by using the LTI state space model described above. In order to verify the accuracy of the LTI model solution, a comparison is carried out with a program which solves the conduction equation via the electrical analogy (Karplus [1958], Yu and van Paassen [2004]).

In Table 2 one of the tested multi-layer wall structures is shown, where the layers are reported from the internal to the external one. A sinusoidal external temperature signal variation of amplitude 1 °C is assumed. No other heat source or flux is considered.

For each case a simulation time of 15 days with a sampling time of 60 seconds is sufficient to reach temperature steady-state conditions for all layers. The considered outputs are the absolute values of the internal and external layer temperatures, the heat fluxes through these layers and the phase difference θ of all these variables with the external temperature. The results are reported in Table 3 while in Figure 2 the steady-state profile of the external air and of the wall temperatures are shown.

5.2 The energy equation

The steady-state energy balance equation is verified by means of a simple simulation. The simulated room is square with length and width of 4 m and an height of 2 m. No window is present. The walls are all considered to be in contact with the external environment and the structure 1 of Table 2 is used. If a value of 10 W/m^2K for both the internal and the external surface heat transfer coefficient is assumed, the calculated transmittance for these structure is 0.524 W/m^2K.

As far as the external conditions are concerned, the external temperature is constant and equal to 0 °C, while no solar radiation contribute is taken into account. The internal heat gain is equal to 52.5 W/m^2 so the total internal heat gain is 840 W. All initial states are equal to zero.

In these conditions from the energy balance equation:

$$q(t) \;=\; \sum_{i=1}^{N_{wl}} U_i\, S_i \left[T_{ea}(t) - Tia(t)\right]\Big|_{\,t=\infty} \tag{24}$$

we can obtain the value of the internal air temperature and compare it to the value supplied by the model once it has reached steady-state conditions. The difference between the program value and the calculated one becomes negligible after 15 simulated days.

5.3 Dynamic behaviour

To test the dynamic behavior of the simulation model, its outputs are compared with the reference values given in a recent standard proposal (CEN [2001]). In this draft, a standard on calculation methods for the evaluation of the thermal performance of buildings and building components is discussed. This proposal provides 21 validation tests to be performed on a single zone with air temperature control. Any calculation method meets the standard if it provides, for the various considered tests, results in accordance with given reference values.

For the simulation a room of width 3.6 m, length 5.5 m, height 2.8 m is used. The west wall is external and has a window of 7 m^2 surface while the other walls are to be considered in contact with rooms at the same temperature of the simulated one. In the tests, two types of glazing system are present, a double-pane glass (DP) and a double-pane glass with an external shading device (SDP). The structures and the thermophysical characteristics of these surfaces are given in CEN [2001]. The characteristics of the different walls, ceilings, and floors, the hourly values of the solar radiation and of the external air temperature used in the tests are reported in Table 4 and 5. The main simulation parameters for the 8 considered tests are summarized in Table 6.

As an example of the obtained results, the hourly values of the cooling power for test 1 is reported in Figure 3. In Table 7 the values of the cooling daily energy and of the maximum cooling power calculated with THESIS and the same values given in CEN [2001] for each of the considered tests are summarized, together with their percentual difference. As far as the daily energy requirement is concerned, the data are in very good accordance with the reference values, with a mean difference of 1.9%. The peak loads deviates more from the reference (mean difference of 8.3%). Both results show that THESIS has a satisfactory dynamic behavior in all of the considered tests, with different wall structures and in presence of solar radiation.

We remark that it is reasonable to find differences between the results provided by different computational techniques, in particular as far as the maximum cooling power is concerned, since each algorithm employs different models for the evaluation of the boundary conditions (i.e. solar radiation, internal and external surface heat transfer coefficient, windows absorption and transmission coefficients). However, in a recent paper (Cecchinato *et al.* [2002]) the simulation results on the standard tests with six different programs (including THESIS) are presented, showing that all the results are located in the band given by the mean value ±15%, both for the maximum cooling power and for the daily cooling energy.

6. CONCLUSIONS

In this paper, THESIS, a Matlab-based software for the thermal-hygrometric dynamic simulation of building, is described, and its performance in terms of load prediction capabilities is evaluated by performing a number of benchmark tests as suggested in CEN [2001]. The performance is fully satisfactory and in agreement with the results of recent studies Cecchinato *et al.* [2002].

The formulation in terms of LTI state space models together with the finite difference scheme applied to the wall grants excellent performance in terms of computational effort. Attention has to be payed in the choice of the simulation time step in order to guarantee asymptotic stability of the LTI model. Furthermore, the proposed technique is particularly suitable when the transfer function technique exhibits some limitations (e.g., the case of walls with relevant thickness).

The implementation in the MATLAB environment is flexible, easy, and efficient. The availability of SIMULINK tools may be very useful for easily testing different control strategies. An interesting possibility given by the adopted small time step employed in the simulations (one minute or even less) is the execution of simulations in real time.

Future works will concern the development of a multi-room program and the coupling different thermal plants, for improving the control strategies of the overall building-plant system.

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	s [m]	λ [W/(m K)]	ρ [kg/m^3]	c_p [J/kg K]
outer plastering	0.02	0.70	1300	840
insulating layer	0.06	0.04	40	840
Masonry	0.12	0.80	1600	840
internal plastering	0.02	0.70	1300	840

Table 2. Multi-layer wall structure tested to verify the solution of the equation.

		Temperature [K]	**Temperature** θ [min]	**Flux** [W/m^2]	**Flux** θ [min]
Internal	B	0.023	374.3	0.228	374.3
	T	0.023	374.3	0.229	374.4
External	B	0.578	101.8	5.383	-110.0
	T	0.578	101.8	5.388	-110.0

Table 3. Results of the conduction test with the structure of Table 2 (B=Benchmark, T=THESIS).

	s	λ	ρ	c_p
	[m]	[W/(m K)]	[kg/m^3]	[J/kg K]
Structure n.1 (external wall)				
outer layer	0.115	0.99	1800	850
insulating layer	0.06	0.04	30	850
Masonry	0.175	0.79	1600	850
internal plastering	0.015	0.70	1400	850
Structure n.2 (internal wall)				
gypsum plaster	0.012	0.21	900	850
mineral wool	0.10	0.04	30	850
gypsum plaster	0.012	0.21	900	850
Structure n.3 (ceiling/floor)				
plastering cover	0.004	0.23	1500	15000
cement floor	0.06	1.40	2000	850
insulating layer	0.04	0.04	50	850
concrete	0.18	2.10	2400	850
Structure n.4 (ceiling/floor)				
plastering cover	0.004	0.23	1500	15000
cement floor	0.06	1.40	2000	850
insulating layer	0.04	0.04	50	850
concrete	0.18	2.10	2400	850
insulating layer	0.10	0.04	50	850
acoustic board	0.02	0.06	400	840

Table 4. Thermophysical properties of the opaque components of the CEN tests

Hour	**External air temperature** [°C]	**Total radiation (West)** [W/m^2]
1	14,08	0
2	13,28	0
3	12,64	0
4	12,16	0
5	12,00	22
6	12,32	55
7	13,12	80
8	14,56	101
9	16,64	117
10	19,04	128
11	21,76	135
12	24,32	150
13	26,24	366
14	27,52	558
15	28,00	703
16	27,52	778
17	26,40	756
18	24,64	604
19	22,56	271
20	21,44	0
21	18,72	0
22	17,12	0
23	15,84	0
24	14,88	0

Table 5. Solar radiation on the west exposure and external air temperature in the CEN tests

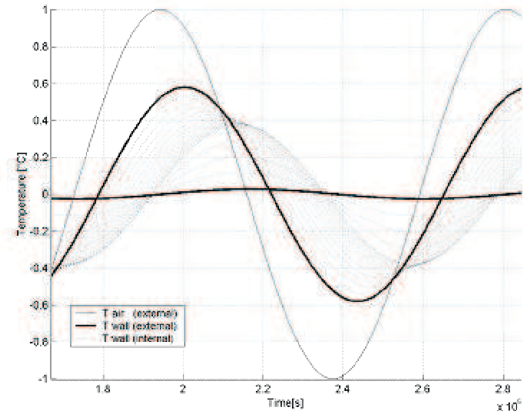

Fig. 2. Air and wall layers temperatures.

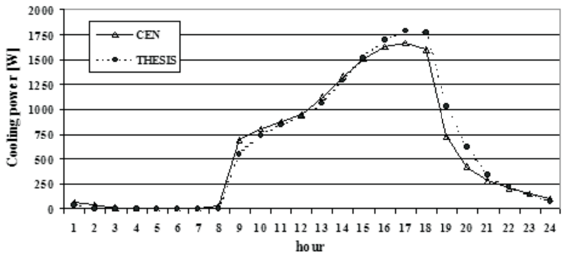

Fig. 3. Hourly profile of the cooling power for test 1.

Test	Shading device	Ceiling/ Floor structure	Internal gains [W/m^2]	Internal gains schedule	System on/off schedule
1	Yes	4/4	20/30	8-18	0-24
2	Yes	3/3	20/30	8-18	0-24
3	Yes	4/4	20/0	8-18	0-24
4	No	4/4	20/30	8-18	0-24
6	Yes	4/4	20/30	8-18	8-18
7	Yes	3/3	20/30	8-18	8-18
8	Yes	4/4	20/0	8-18	8-18
9	No	4/4	20/30	8-18	8-18

Table 6. Parameters of the CEN tests. The internal heat gain value is shown as x/y where x is the value of the specific convective gain(per m^2 of floor) while y is the value of the specific radiant gain

Test	Maximum cooling power			Cooling daily energy		
	Standard	THESIS	Δ	Standard	THESIS	Δ
	[W]	[W]	[%]	[Wh]	[Wh]	[%]
1	1667	1784	7.0	14188	14591	2.8
2	1466	1546	5.5	14259	14569	2.2
3	1173	1282	9.3	8604	8978	4.3
4	3678	3783	2.8	31066	31557	1.6
6	1670	1871	12.0	13469	13687	1.6
7	1646	1739	5.7	13514	13734	1.6
8	1136	1328	16.9	8253	8488	2.8
9	3772	4057	7.5	28406	27921	-1.7

Table 7. Calculated peak loads and total cooling loads; differences with the CEN benchmarks.