

Development of a Model to Simulate the Performance of Hydronic Radiant Cooling Ceilings

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Abstract

A significant amount of the electrical energy used to cool non-residential buildings equipped with all-air systems is drawn by the fans that transport the cool air through the thermal distribution system. Hydronic radiant cooling systems have the potential to reduce the amount of air transported through the building by separating the tasks of ventilation and thermal conditioning. Because of the physical properties of water, hydronic radiant cooling systems can transport a given amount of thermal energy using less than 5% of the otherwise necessary fan energy. This improvement alone significantly reduces the energy consumption and peak power requirement of the air conditioning system.

Hydronic radiant cooling systems have been used for more than 30 years in hospital rooms to provide a draft-free, thermally stable environment. The energy savings and peak-load characteristics of these systems have not yet been analyzed systematically. Moreover, adequate guidelines for design and control of these systems do not exist. This has prevented their widespread application to other building types.

The evaluation of the theoretical performance of hydronic systems could be made most conveniently by computer models. Energy analysis programs such as DOE-2 do not have the capability to simulate hydronic radiant systems yet. In this paper the development of a model that can simulate accurately the dynamic performance of hydronic radiant cooling systems is described. The model is able to calculate loads, heat extraction rates, room air temperature and room surface temperature distributions, and can be used to evaluate issues such as thermal comfort, controls, system sizing, system configuration and dynamic

response. The model was created with the Simulation Problem Analysis and Research Kernel (SPARK) developed at the Lawrence Berkeley Laboratory, which provides a methodology for describing and solving the dynamic, non-linear equations that correspond to complex physical systems.

1. Approach

Recent experience has shown that radiant cooling has the potential to be an energy efficient alternative to all-air cooling systems (see [1, 2]). Radiant cooling systems can achieve high savings, especially if used with alternative cooling sources and elements with large thermal mass. Unfortunately, there are not enough design data available for these cooling systems, and the strong influence of the transient response of this cooling system causes difficulties in defining simple design rules. A survey sent to about 300 researchers and practitioners in building science showed general interest in a design tool for radiant cooling and heating systems. A building simulation program appears to be a useful tool for the understanding and predicting the thermal behavior of buildings equipped with a radiant cooling system.

Existing building simulation programs like DOE-2¹ are often not very flexible in incorporating new technologies [3]. For the simulation of cooling systems in general, there is a need for highly modular programs which are easily extendable and easy to use. The program RADCOOL has been designed to accurately simulate the dynamic performance of hydronic radiant cooling systems. The ultimate goal for RADCOOL is to perform as a DOE-2 SYSTEMS module, since DOE-2 cannot model radiant cooling systems yet. At the same time, DOE-2 has strong capabilities in modeling the weather and HVAC components that now constitute input to RADCOOL.

RADCOOL is designed to calculate loads, heat extraction rates, room air temperature and room surface temperature distributions. It can already be used to evaluate issues such as

1. DOE-2 is a thermal building simulation program developed by the Simulation Research Group at LBL

dynamic response and controls, and it can be extended to evaluate thermal comfort, system sizing, system configuration, and energy use. The simulation program was created with the LBL Simulation Problem Analysis and Research Kernel (SPARK) [4], which provides a methodology for describing and solving the dynamic, non-linear equations that correspond to complex physical systems. The design of the program allows the addition of new modules in a straightforward way. This quality is important in situations where new assumptions need testing.

SPARK was chosen as the environment for RADCOOL because of its ability to describe and solve dynamic equations. More generally described, SPARK is an environment that was developed to treat problems as networks. A major advantage of this feature is that fairly complex problems may be defined in an intuitive way. On the other hand, if the problem is of a simple nature, the usage of the SPARK environment might be rather cumbersome. The use of SPARK is recommended only in problems where there are many correlations between different variables (a real *network*, or a large number of simultaneous equations).

Logical statements are difficult to use in SPARK. Such logical statements are by nature bound to a sequential approach to a problem, while the network approach solves a problem for all variables simultaneously. It is not only very cumbersome for the programmer to use logical statements in SPARK, it also means a waste of computer time and disk space. In order to avoid wasting disk space, and computer time, simple tasks and case distinctions are done in RADCOOL outside the SPARK environment. Therefore, a program flow as shown in Figure 1, was chosen for RADCOOL to reduce the number of acyclic calculations and to avoid most of the logical statements.

In the "preliminary data processing" section of RADCOOL, the complete description of the simulation problem is created. This includes the room geometry, building materials, load schedules, weather data, running period assumptions for the calculation, and other numerical data. The calculations performed in this section are sequential and as a result

are not treating any variables of the cyclic type. Among these are the calculation of shape factors, of convection film coefficients, and the weather calculations which, even though considered as cyclic because of their yearly cycles, are only performed once for a certain SPARK problem. To address these considerations, it is highly recommended that the user perform the above-mentioned calculations with programs written in FORTRAN or C, but *not* in SPARK.

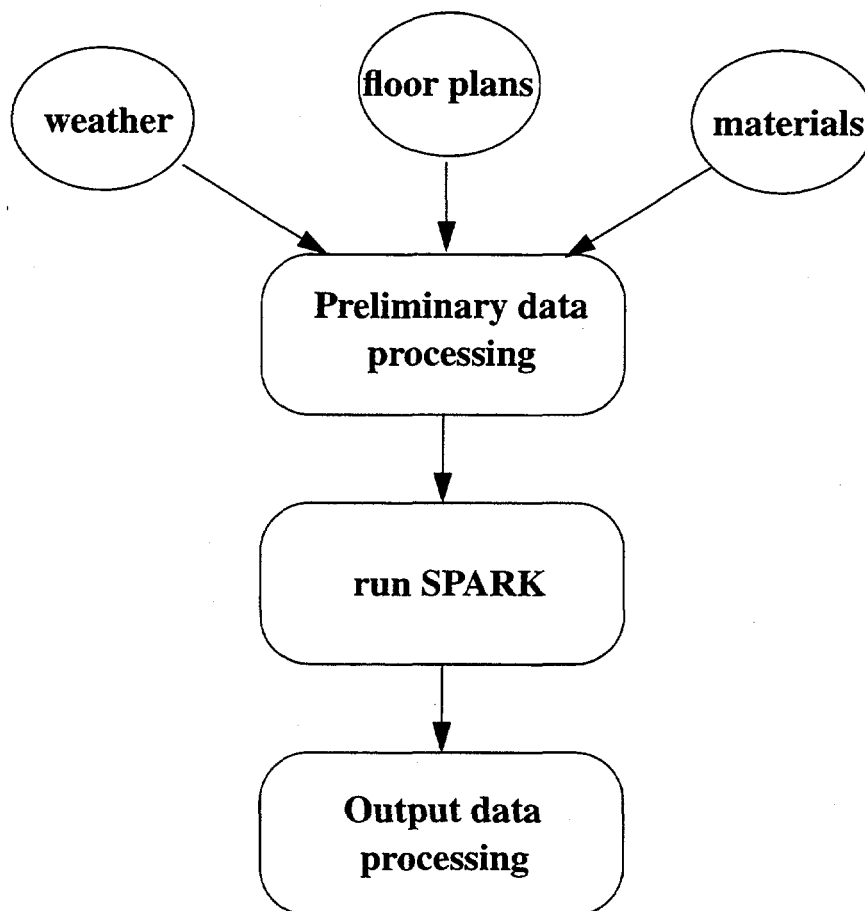


Figure 1. Broad Program Flow for RADCOOL

1.1 The SPARK passive elements

1.1.1 Unidimensional heat transfer

A good approximation of building components such as passive walls and windows is to consider that their surfaces (and/or each imaginary internal plane parallel to the surfaces) are isothermal. This approach neglects surface temperature gradients and edge effects in these building components. The benefits of analytical unidimensional heat transfer mechanisms offset the impreciseness of the model.

1.1.1.1 The unidimensional heat conduction/storage equations

Consider an infinitely long and wide wall slab, with homogenous and isotropic material properties. Assume a one-dimensional heat flux on the surface of this slab. The temperature at each point over the thickness of the slab can be defined as a time- and space-dependent function: $T = T(x, t)$ (x is the space variable over the thickness of the wall, and t the time variable).

Consider a volume element ΔV of this slab, with a heat flux of $q(x, t)$ incident on one surface and a heat flux $q(x + \Delta x, t)$ emerging from the opposite surface, as shown in Figure 2.

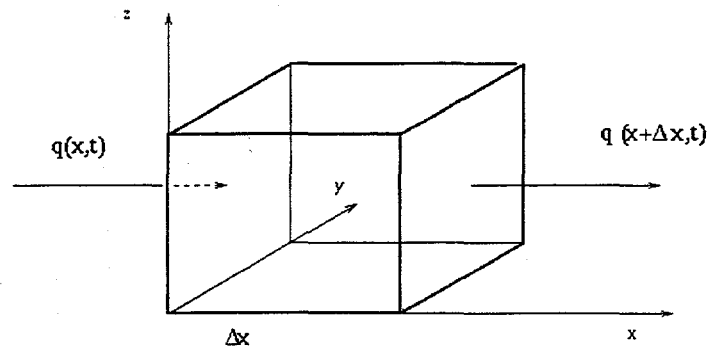


Figure 2. Volume element

The Fourier equation describes conduction inside the wall layer:

$$q(x, t) = -k \frac{\partial T}{\partial x}(x, t) \quad (1)$$

where

k is the thermal conductivity of the material [W/m K].

The heat diffusion equation describes the interdependence of the temporal and spatial effects inside this volume element:

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2} \quad (2)$$

where α is the thermal diffusivity [m^2/s]:

$$\alpha = \frac{k}{\rho c_t} \quad (3)$$

ρ is the density of the material [kg/m^3]

c_t is the heat capacity of the material [$\text{J}/\text{kg-K}$].

1.1.1.2 The RC approach to solve the heat conduction/storage equations for one solid layer in SPARK

Consider the wall slab from Section 1.1.1.1. Equations (1) and (2) are the differential expressions of heat conduction and heat diffusion through this layer. The scope of this section is to simplify the two equations and bring them into a form that a SPARK program can easily solve. To this end, the space dependence of the temperature needs to be expressed by means of finite differences.

The analogy of the Fourier equation and the heat diffusion equation with Ohm's law and the electrical diffusion equation is obvious. By virtue of this analogy, the following can be defined:

- a "lumped thermal resistance" R_t as:

$$q = \frac{\Delta T}{R_t} \quad (4)$$

- a "lumped thermal capacitance" C_t as:

$$q = C_t \frac{\partial T}{\partial t} \quad (5)$$

The idea of the RC approach is to determine the resistance and capacitance in equations (1) and (2), and to express these equations in finite format. After some calculations, the following results can be obtained:

- the thermal resistance of a layer of thickness Δx can be defined as:

$$R_t = \frac{\Delta x}{k} \quad (6)$$

- the thermal capacity of a layer of thickness Δx can be defined as:

$$C_t = \rho c_t \Delta x \quad (7)$$

Using (6), (7), and (3), yields the result:

$$\alpha = \frac{(\Delta x)^2}{R_t C_t} \quad (8)$$

Equations (4) - (7) give the "lumped" RC model of a homogenous, isotropic wall slab. This model is however a crude approximation of the real case, in which each infinitesimal layer dx of the slab has its own resistance and capacity. In order to use the RC approach in a more accurate way, a wall slab has to be modeled as composed of a number of layers, each having a resistance and capacity expressed by (6) and (7) respectively. The larger the number of layers simulated, the thinner each layer becomes (Δx decreases), and the more the model approaches the real case.

1.1.2 The structure of the passive wall in SPARK

Based on the general scope of RADCOOL, the SPARK module corresponding to the heat conduction/storage sub-component of a wall should be able to model up to four layers of different materials. Therefore, a RC model of each layer was needed. Then, the overall wall module was designed to solve the system formed by the RC equations for each layer. Several test programs were written in SPARK in order to determine the number of sub-layers that need to be defined, so that the results of the simulation would be in fairly good

agreement with the corresponding theoretical results. The results show that a combination of three resistances and two capacities differs only in the order of a few percent from a combination of four resistances and three capacities, but that the computation time increases significantly for the second case as compared to the first. It was therefore considered appropriate that each layer have a maximum of three resistances and two capacities. Another important consideration in selecting the final "equivalent circuit" for the wall was that, out of the four layers of the wall, the two in the middle are not exposed to any other radiation sources, whereas the surface layers are exposed to convection, long wave (IR) radiation, solar radiation (if exterior layer), etc. It was therefore considered appropriate that for each of the two layers in the middle, a 2R, 1C circuit be modeled, while for each of the two surface layers, a 3R, 2C circuit be modeled. The resulting RC circuit is shown in Figure 3.

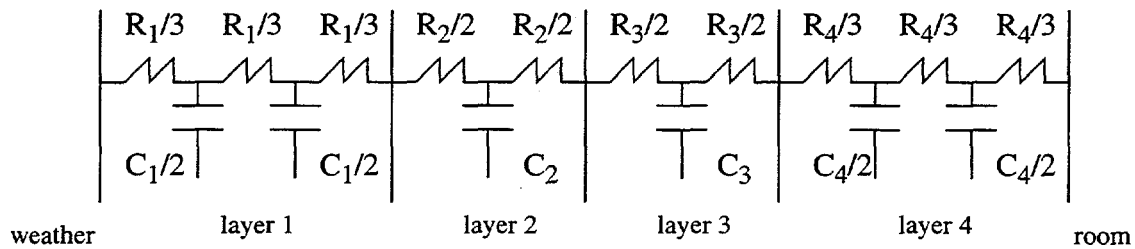


Figure 3. The RC model of the 4-layer wall

1.1.2.1 The equations for the temperature nodes in SPARK

The SPARK module that simulates the heat conduction/storage in a 4-layer passive wall solves the system of equations that describe the heat balance for each temperature node of the wall. The temperature nodes can be identified from Figure 3: each surface layer contains two temperature nodes, each interior layer contains one temperature node, each interface contains one temperature node, and each surface contains one temperature node.

Consider interior node i . Denote by $i-1$ and $i+1$ the nodes located immediately to the left and right of node i . Denote by R_{i-1} and R_{i+1} the resistances of the sub-layers $(i-1, i)$ and $(i,$

$i+1$), and by C_i the capacity corresponding to node i . The heat flux balance equation for interior node i is:

$$\frac{T_{i-1} - T_i}{R_{i-1}} = C_i \frac{dT_i}{dt} + \frac{T_i - T_{i+1}}{R_{i+1}} \quad (9)$$

Consider interface node i . Denote by $i-1$ and $i+1$ the nodes located immediately to the left and right of node i . Denote by R_{i-1} and R_{i+1} the resistances of the sub-layers $(i-1, i)$ and $(i, i+1)$. The heat flux balance equation for interface node i is:

$$\frac{T_{i-1} - T_i}{R_{i-1}} = \frac{T_i - T_{i+1}}{R_{i+1}} \quad (10)$$

Consider surface node i . Denote by $i+1$ the node located immediately inside the wall. Denote by R_{i+1} the resistances of the sub-layers $(i, i+1)$, and by q_i the exterior heat flux incident on the wall. The heat flux balance equation for the surface node i is:

$$q_i = \frac{T_i - T_{i+1}}{R_{i+1}} \quad (11)$$

The system of equations for a 4-layer wall with the equivalent circuit shown in Figure 3 is composed of

- 6 differential equations of type (9), corresponding to the interior nodes
- 2 linear equations of type (10), corresponding to the interface nodes, and
- 2 linear equations of type (11), corresponding to the surface nodes.

1.2 The core cooling ceiling

1.2.1 Two-dimensional heat transfer analysis

The unidimensional heat transfer analysis method presented in Section 1.1.1 cannot yield good results in the case of a building component that represents a heat source or sink. In this case, a two-dimensional heat transfer analysis method, capable of describing the behavior of the wall temperature in both “main directions” of a wall cross-section, is necessary.

1.2.1.1 The heat conduction/storage equations in 2-D

Consider a solid wall slab with homogeneous and isotropic material properties. Consider that the temperature is a function of only two dimensions of the wall, and is constant in the third dimension (e.g. the temperature varies over a cross-section of the wall, but cross-sections parallel to each other have the same temperature profile).

In analogy with Section 1.1.1.1, the temperature in the cross-section can be considered a function of space and time, $T = T(x, y, t)$, (where x and y are the space variables, and t the time variable).

Consider a volume element of this slab and a one-dimensional heat flux incident on one surface, as in Figure 2. The 2-D Fourier equation for heat transfer in one direction is analogous to equation (1), and has the expression

$$Q(x, t) = -k\Delta A \frac{\partial}{\partial x} T(x, y, t) \quad (12)$$

where

Q is the total heat incident on the surface [W]

k is the thermal conductivity of the wall material [W/m K]

ΔA is the area of the face of the volume element normal to the heat flux [m^2].

The two dimensional diffusion equation has the expression

$$\frac{\partial T}{\partial t} = \alpha \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (13)$$

where α is the thermal diffusivity given by equation (3) [m^2/s].

1.2.1.2 The RC approach to solving the 2-D heat conduction/storage equations

The same approach as in Section 1.1.1.2 was taken in order to express the space and time dependence of the temperature by means of finite differences. The wall is described as a collection of parallel boxes, and the discretization is made in a plane normal to the surface of the wall. The boxes have one dimension equal to the length of the wall, while the other two are normal to this direction, and much smaller.

In an analogy to electrical circuits, a lumped thermal resistance can be defined in relation to the heat conduction through each box in a given direction, as:

$$Q_{\xi} = \frac{\Delta T_{\xi}}{R_{\xi}} \quad (14)$$

where ξ denotes the direction in 3-D. Similarly, a lumped thermal capacity can be defined in relation to the heat stored inside each box, as

$$Q = C \frac{\partial T}{\partial t} \quad (15)$$

The resistance and capacity can be calculated based on the thermal properties of the wall material as

$$R_{\xi} = \frac{\Delta \xi}{k \Delta A_{\xi}} \quad (16)$$

$$C = \rho c_v \Delta V \quad (17)$$

where

ΔA_{ξ} is the area of the box surface normal to the direction ξ [m²]

ΔV is the box volume [m³].

1.2.2 The 2-D model of the cooled wall in SPARK

SPARK programs were written in order to simulate the 2-D heat transfer. Figure 4 shows an example of a 2-D RC circuit that has been "translated" into a program. This model assumes that there is some periodicity over the slab cross-section, so that the analysis of a "sample" (in these cases, the sample has a thickness of 4 dx) correctly describes the temperature profile of the whole wall. Examples of periodicity of the temperature over the cross-section of the wall are the case in which the isotherms are parallel to each other (passive wall) and the case in which there are heating or cooling sources in the wall normal to the cross-section and periodically spaced (a hydronically cooled or heated wall).

The resistances and capacities in the figures are calculated as

$$R_x = \frac{dx}{kdydz} \quad (18)$$

$$R_y = \frac{dy}{kdx dz} \quad (19)$$

$$C = \rho c_i \Delta V \quad (20)$$

and

$$dx = \frac{\text{dist-between-pipes}}{n_x} \quad (21)$$

$$dy = \frac{\text{thickness-of-wall}}{n_y} \quad (22)$$

$$\Delta V = dx \times dy \times z \quad (23)$$

where $dz = z$ is the length of the wall.

The heat balance at the temperature nodes can be derived by analogy with the 1-D situation (equations (9)-(11)). The heat balance for the interior node (i,j) is

$$\frac{T_{i-1,j} - T_{i,j}}{R_{x,i-1}} + \frac{T_{i,j-1} - T_{i,j}}{R_{y,j-1}} = C_{i,j} \frac{dT_{i,j}}{dt} + \frac{T_{i,j} - T_{i+1,j}}{R_{x,i+1}} + \frac{T_{i,j} - T_{i,j+1}}{R_{y,j+1}} \quad (24)$$

where $R_{x,k}$ and $R_{y,k}$ are thermal resistances connecting the node (i,j) with the rest of the network in the x and y directions, respectively, and C_i is the capacitance of a cell in row i.

The heat balance for the surface node (i,j) is

$$Q_{i,j} = \frac{T_{i,j-1} - T_{i,j}}{R_{y,j-1}} \quad (25)$$

If the wall is heated or cooled by water running through imbedded pipes, the contact between the pipes and the wall is modeled as a boundary condition. The thermal resistance between the wall node in contact with the pipe, and the water, is equal to the pipe wall resistance. In the case of a cylindrical pipe, this resistance is defined as:

$$R_{\text{boundary}} = \frac{\ln\left(\frac{r_{\text{ext}}}{r_{\text{in}}}\right)}{2\pi Lk} \quad (26)$$

where

r_{ext} is the exterior radius of the pipe [m]

r_{in} is the interior radius of the pipe [m]

L is the length of the pipe [m].

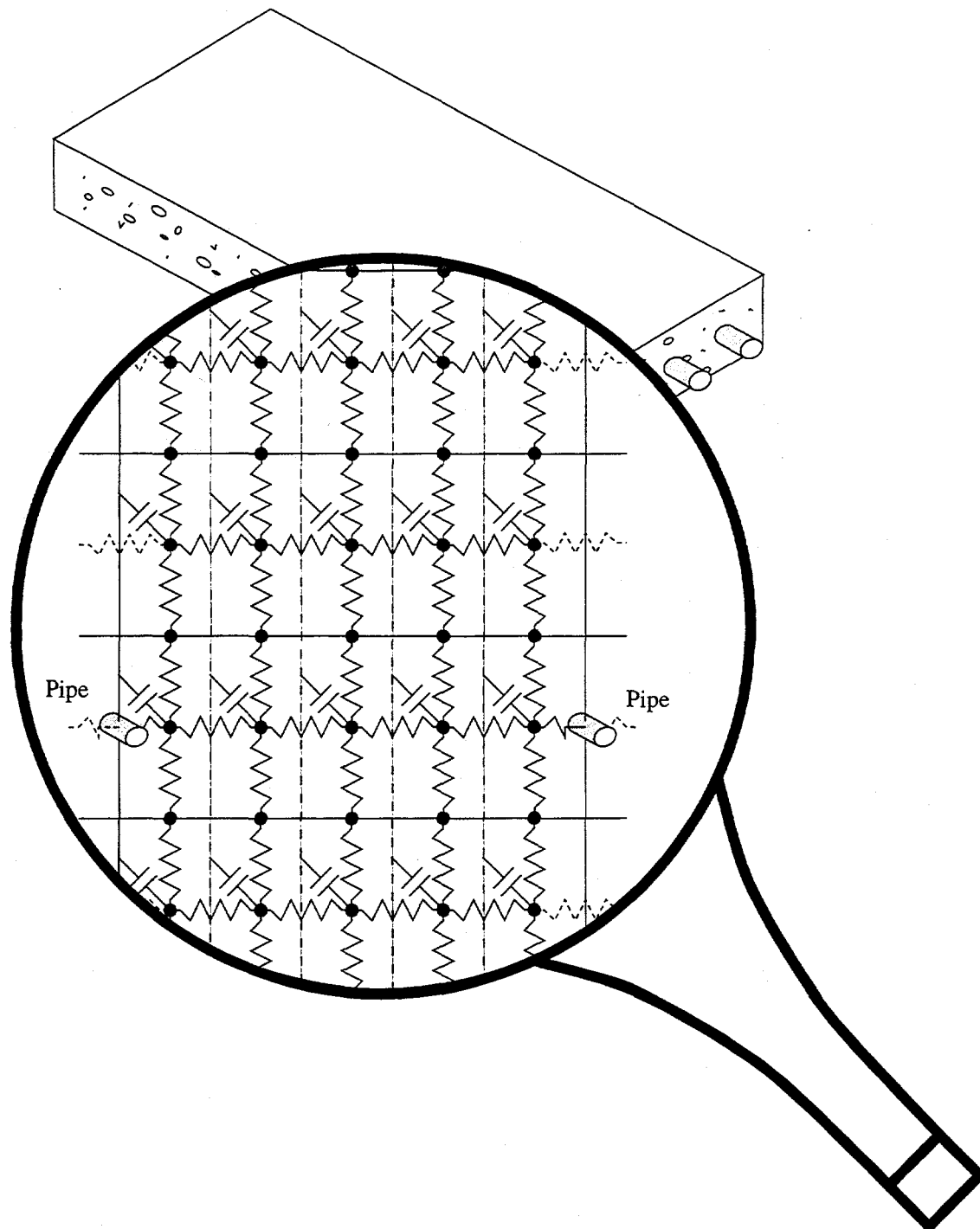


Figure 4. The two-dimensional circuit of the cooled wall.

2. Model evaluation

In order to evaluate the proposed RADCOOL model, three types of tests have been carried out. These are:

- a comparison between the results for an individual wall module and the analytical solution of the conditions modeled
- a comparison between results for a passive structure, and DOE-2 results for the same structure
- a comparison between the results for an active structure and measurements for the same structure.

2.1 Test to determine the accuracy of the RC wall model

To determine the accuracy of the RC wall model, the results from the SPARK program set up to model a given problem were compared to the analytical solution of the problem.

2.1.1 The problem

Consider the one-dimensional heat transfer problem of a homogenous and isotropic wall slab $0 < x < l$, with zero initial temperature, and with the planes $x = 0$ and $x = l$ kept at temperatures zero and $\sin(\omega t + \epsilon)$ respectively [5].

2.1.2 The analytical solution

The temperature of a node situated at x from the 0 plane of the wall is [5]:

$$t = A \sin(\omega t + \epsilon + \phi) + 2\pi\alpha \sum_{n=1}^{\infty} \frac{n(-1)^n (\alpha n^2 \pi^2 \sin \epsilon - \omega l^2 \cos \epsilon)}{\alpha^2 n^4 \pi^4 + \omega^2 l^4} \sin \frac{n\pi x}{l} e^{-\frac{\alpha n^2 \pi^2 t}{l}} \quad (27)$$

where

$$A = \left| \frac{\sinh kx (1+i)}{\sinh kl (1+i)} \right| = \left\{ \frac{\cosh 2kx - \cos 2kx}{\cosh 2kl - \cos 2kl} \right\}^{\frac{1}{2}} \quad (28)$$

$$\phi = \arg \left\{ \frac{\sinh kx (1 + i)}{\sinh kl (1 + i)} \right\} \quad (29)$$

and

$$k = \left(\frac{\omega}{2\alpha} \right)^{\frac{1}{2}} \quad (30)$$

2.1.3 The actual data

In order to compare the results from the SPARK program and the analytical solution, a 20 cm concrete wall slab was modeled and the temperature at half the thickness of the slab ($x=10$ cm) was studied. The value of the thermal diffusivity of the wall material was $\alpha = 7.2 \times 10^{-7} \text{ m}^2/\text{s}$. The sine temperature function on the $x = l$ surface was chosen with a period of 24 hours ($\omega = 7.3 \times 10^{-5} \text{ s}^{-1}$) and no time lag ($\epsilon = 0$).

To determine the analytical solution, a FORTRAN program was written in which 100,000 terms were summed to calculate the temperature at each time step.

The SPARK program was designed so that all the four layers of the wall slab have the same thermal properties: density $\rho = 2400 \text{ kg/m}^3$, heat capacity $c_t = 1040 \text{ J/kg-K}$, and conductivity $k = 1.8 \text{ W/m-K}$ and the same thickness, 5 cm.

2.1.4 Results

Figure 5 compares the variation of the temperature in the node $x = 10$ cm of the slab, given by the analytical solution (26) - (29), with the results of the SPARK RC model. The sinusoidal temperature on the $x = 20$ cm plane of the slab is also shown. As it can be seen, there is good agreement between the results of the SPARK model and the analytical solution of the problem. The error introduced by the model is 6%.

2.1.5 Test to determine the accuracy of the 2-D RC model

The same problem was solved for the 2-D wall model (Figure 4). It was found that the results from the 2-D model with the dimensions $dx = 2.5$ cm ($x=10$ cm), $dy = 4$ cm ($y=20$ cm) and $z = 3$ m agree with the analytical solution similarly to the 1-D model.

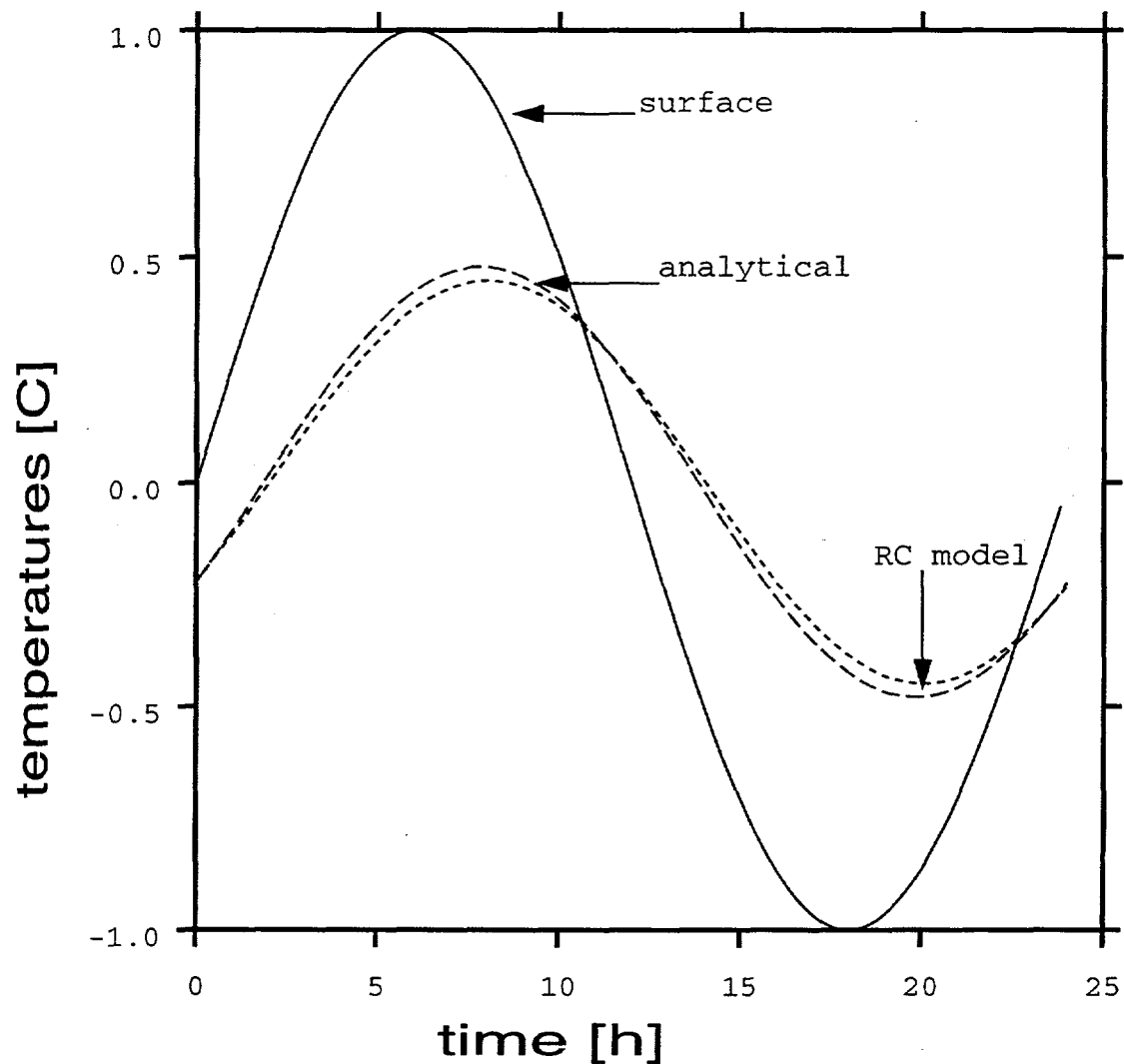


Figure 5. Temperature at the mid point of a wall slab - comparison of SPARK model and analytical solution in the context of the surface wall temperature

2.2 Comparison with DOE-2

To evaluate the performance of RADCOOL in describing the heat transfer mechanisms inside a building structure, a SPARK “passive” test room was created. The test room simulates the thermal transfers occurring in a one-room building. The test room

- is of rectangular shape

- may have several exterior surfaces with several windows
- can have different wall structures (wall layer sequences)
- can be ventilated
- can have several cooling systems (including non-radiant), and cooling sources.

The same test room was modeled with DOE-2, and the results of the two models were compared.

2.2.1 The test room

For the purpose of evaluating the results from a RADCOOL calculation, the test room has the dimensions of 4 m x 5 m x 3 m, and all its walls are exterior. The floor is in direct contact with the ground. A window was modeled on the wall facing west. The dimensions of the test room and the geometry of the wall facing west are shown in Figure 6.

To compare the results of the RADCOOL simulation with the results of a DOE-2 calculation, a test room with the same dimensions was modeled in DOE-2, in the same climate, and with the same wall structure. By simulating the heat transfer in the test room using the same set of assumptions, the comparison of the two sets of results becomes legitimate.

The climate selected for the test room location corresponds to the Red Bluff DOE-2 weather file (a DOE-2 weather file is representative for the climate where the data acquisition system is placed). To match the DOE-2 model, a pre-heating period was simulated: the weather corresponding to a chosen day was repeated several times to provide the conditions for the building structure to adjust for the thermal storage effects. The selected test day was June 1, and the pre-heating period was set to 7 days.

No internal loads or mechanical systems were modeled inside the test room. The comparison of the two models is aimed at showing the similarities, or discrepancies, in their respective approaches to simulate heat transfer in a building.

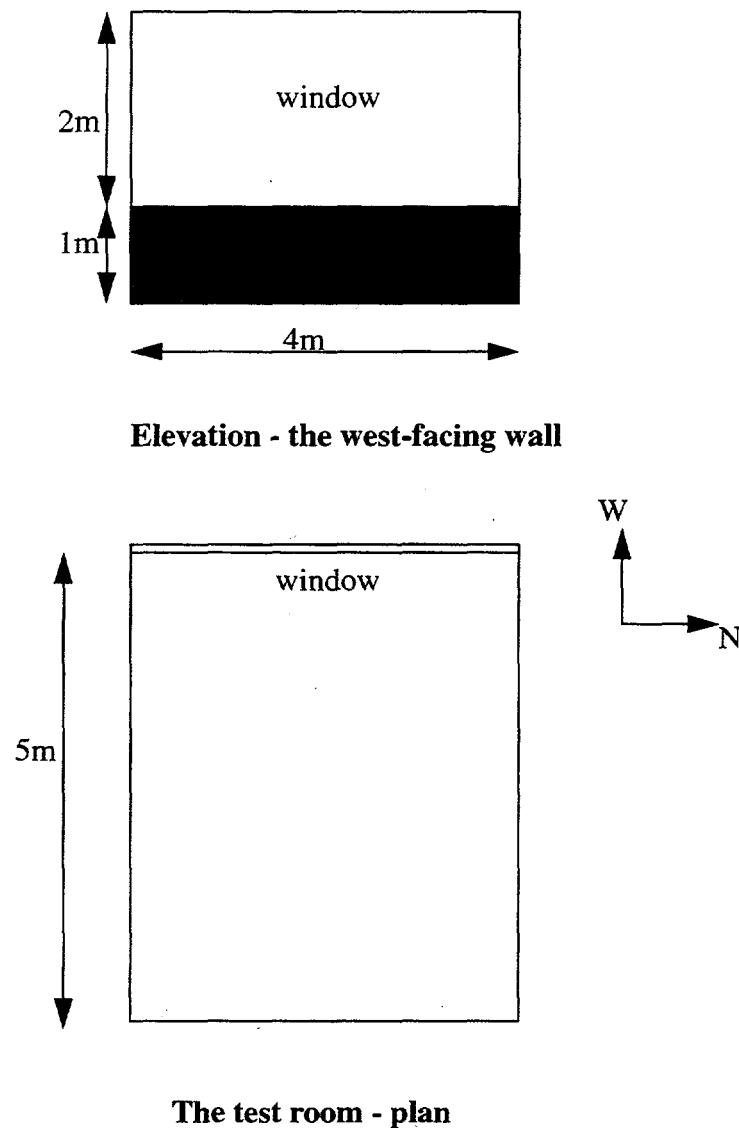


Figure 6. Test room description for performance evaluation in California

In the set of simulations aimed at testing the capacity of RADCOOL to model the room conditions in the given climate, the approach taken was to model the same structure in all six passive walls. To determine how flexible the wall module is to structure changes, three layer structures were modeled for the purpose of the evaluation. Figure 7 shows cross-sections of the three structures.

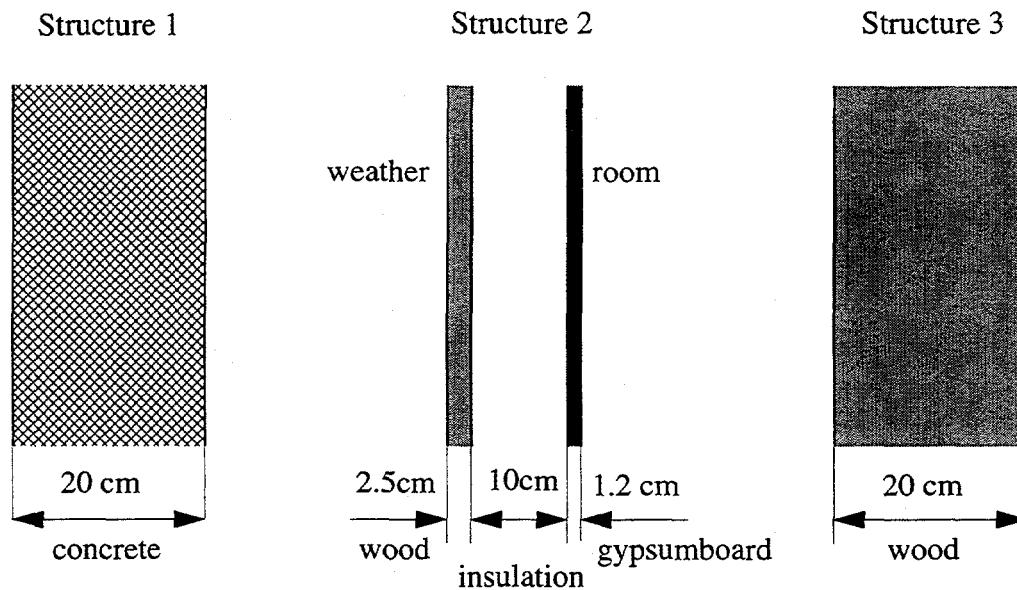


Figure 7. The material structure of the walls in the test room

2.2.2 Results

The parameter chosen for the comparison of the two models was the indoor air temperature. Figure 8 shows a comparison between SPARK and DOE-2 results for the second structure. The figure presents the indoor air temperature profile inside the test room, in the context of the outside air temperature.

The second structure represents a typical exterior wall. An insulation layer is sandwiched between the wooden exterior and the gypsumboard interior layers. The whole structure is designed to minimize the heat conducted through the building envelope. As a result, the test room becomes very hot during the day as the radiation entering the room through the window is not lost to the exterior by means of conduction. The daily swing of the indoor air temperature is much higher compared to that of the outside air, and the time of the peak is delayed as a result of radiation effects, and of the insulated character of the structure.

The results from the two models are very similar, and agree they in the same way for the other two structures.

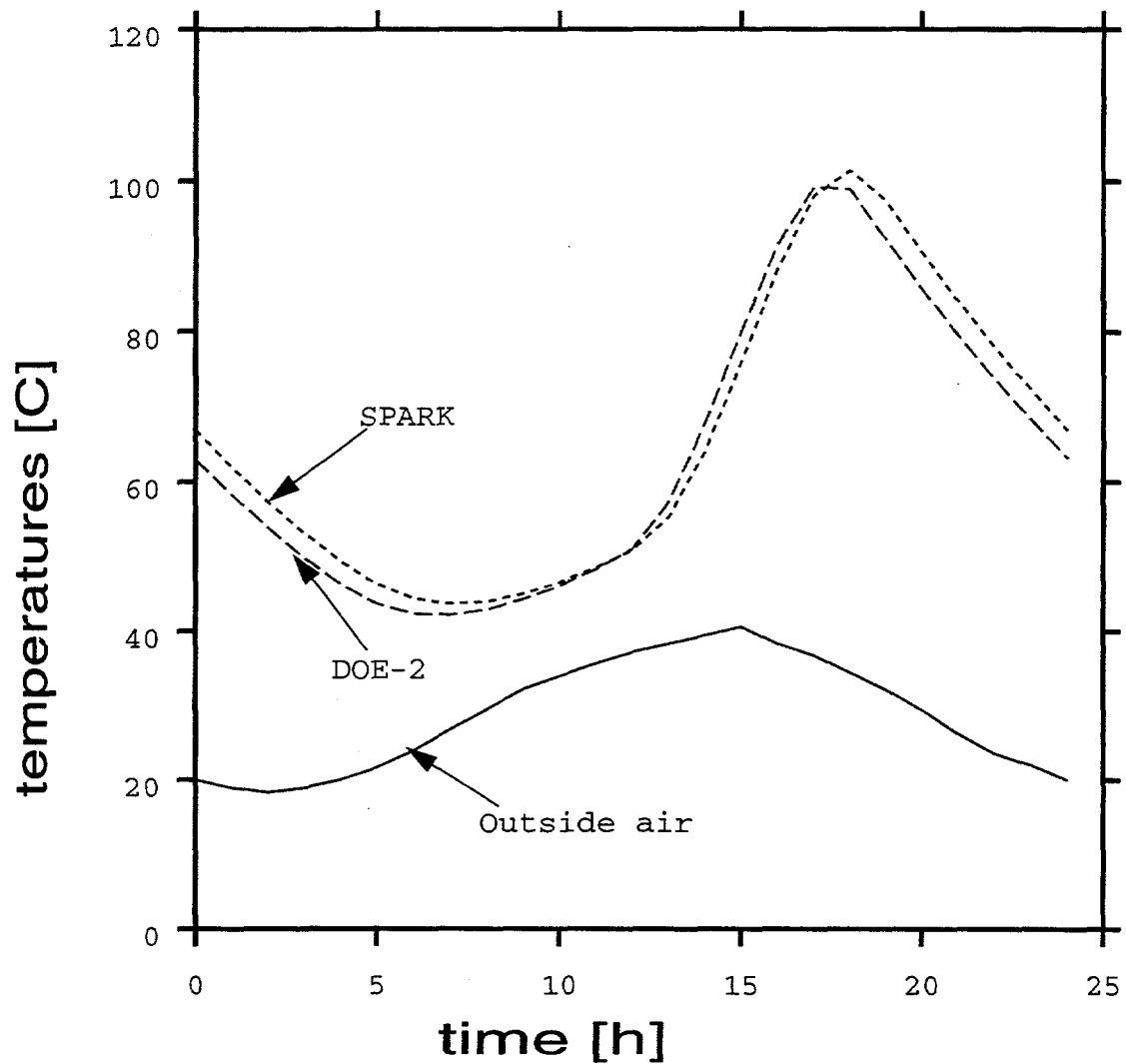


Figure 8. Indoor air temperature: structure 2 (wood - insulation - gypsumboard)

The comparative analysis was found satisfactory as a result of the fact that the two models agreed within 2% for all three structures.

2.3 Comparison with measured data

The performance of RADCOOL was tested against measured data. Measurements were available from a building equipped with a core cooling system. The weather data at the site during the measurement period was also available for the simulation.

2.3.1 The DOW-Europe test room

Measurements were performed in the Swiss building housing the European headquarters of DOW Chemicals. The building coordinates are 47 °N and 9 °E. The test room is located on the top floor (height = 12.8 m above the ground), and is equipped with a hydronic radiant cooling system. It has the dimensions of 2.9 m x 4.3 m x 2.85 m, and its facade is oriented 65° east of south (see Figure 9).

2.3.2 Wall composition

Figures 10 and 11 show the composition of the test room walls.

The facade (exterior wall) has the dimensions of 2.9 m x 2.85 m. The piece below the window is a 2.9 m x 0.925 m wall, with an overall U value of 0.34 W/m²K. The double pane window has the dimensions of 2.9 m x 1.925 m. The overall U-value of the window is 1.75 W/m²K. Automatic shades are installed over the windows; the shades are operated (closed or opened) when the overall exterior irradiance passes the threshold of 120 W/m².

The interior walls consist of sheetrock and plaster.

The ceiling (also the roof of the building, as the test room is placed on the top floor) has the dimensions of 2.9 m x 4.3 m. Its overall U-value is 0.32 W/m²K.

The floor also has the dimensions of 2.9 m x 4.3 m. It represents a raised floor over the cooled concrete slab of the room below. The overall U-value of the floor is 2.5 W/m²K.

2.3.3 Loads

1. During the measurements, it was considered necessary to model the internal loads, in order to eliminate any unexpected results that might occur due to random occupant behav-

ior. The interior loads were therefore modeled by installing several light bulbs inside the room, and controlling the times they were switched on and off. The following **occupancy pattern** was thus obtained: 35 W/m^2 (a total of 436 W), from 8 am to 12 pm and 1 pm to 5 pm, Monday through Friday.

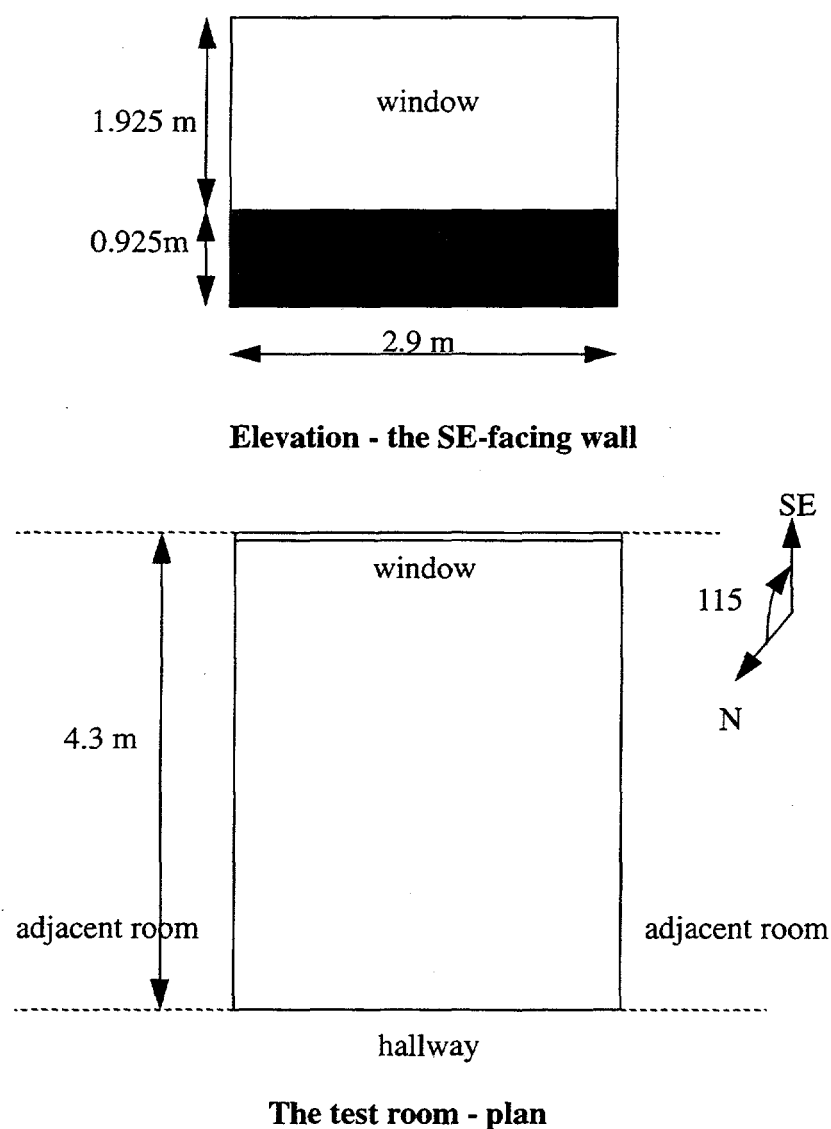


Figure 9. The DOW-Chemicals test room orientation and layout

2. The **solar radiation** intensities needed in the simulation of the experiment were obtained in the form of weather recorded at a station located about 20 km away. There are no tall buildings on the site, so the DOW building is not shaded by any obstacles.

3. 0.2 ACH **infiltration** rates for the test room were measured.

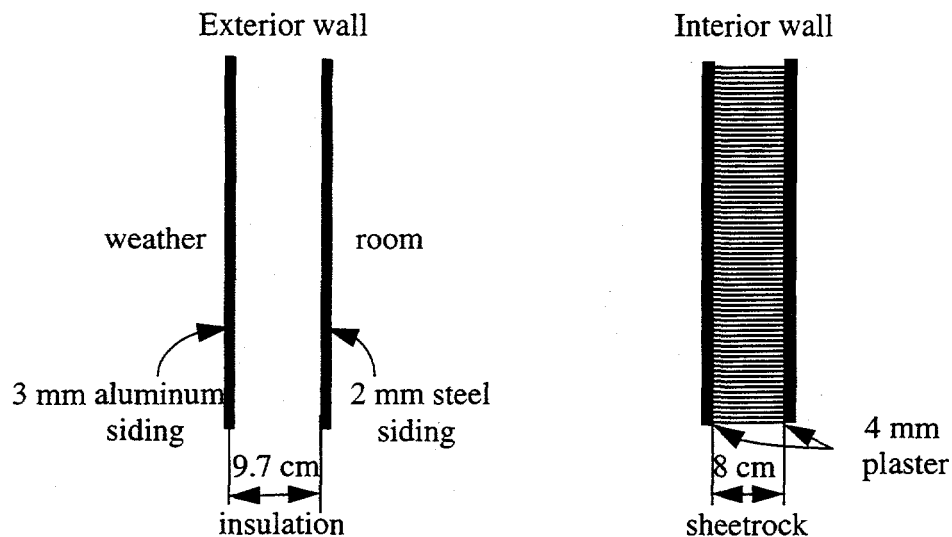


Figure 10. Composition of the vertical walls in the DOW-Chemical test room.

2.3.4 System

1. **The cooled ceiling:** the water flow is 100 l/h per register. Each register is 8.3 m² large. There are 1.5 registers on the cooled ceiling, which gives 150 l/h (0.042 kg/s) total flow. The water is supplied at a temperature that has been measured and is available as input data. The ceiling pipes are made out of polyethylene, have 16 mm exterior and 12 mm interior diameters, and are placed at 15 cm on center, 10 cm deep inside the concrete.
2. **Ventilation:** air at a rate of 1.1 ACH is supplied to the room over the day, when it is occupied. The supply rate is 0.55 ACH at night. The temperature of the supply air has been measured and is available for the simulation.

2.3.5 Boundary conditions:

1. Measurements of the air temperature in only one adjacent room are available. We used this air temperature for both adjacent rooms, and the measured hallway temperature as a boundary condition for the “back” wall.

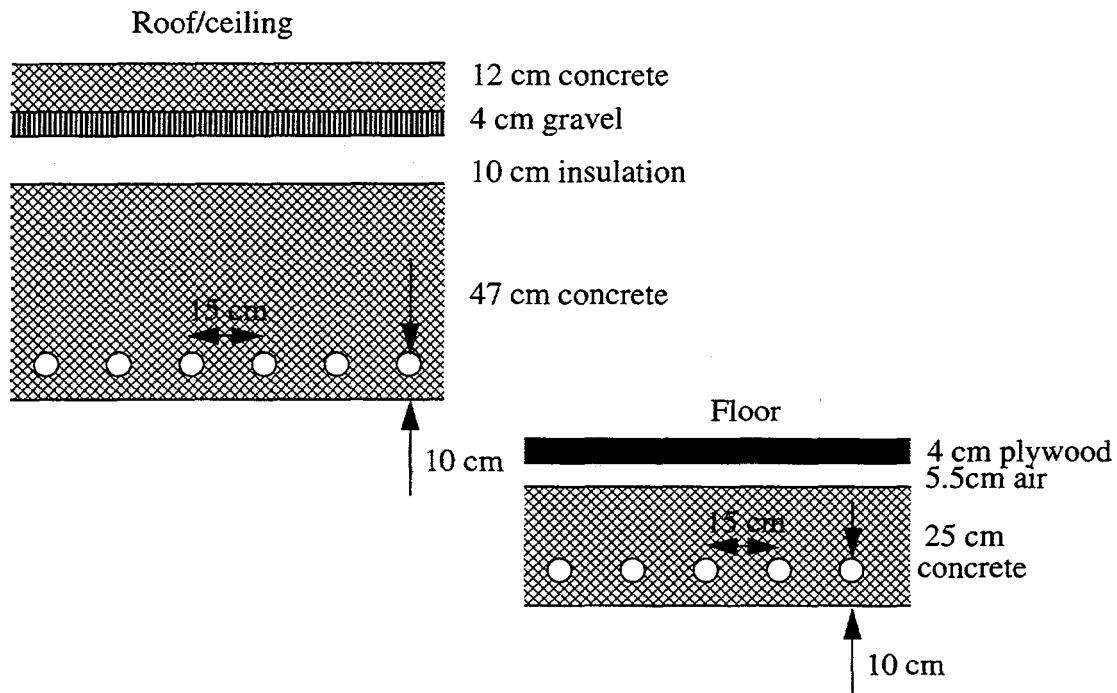


Figure 11. The composition of the roof and floor in the DOW-Chemicals test room.

2. Measurements of the air temperature inside the room are available, but not in the room below. We assumed the air temperature of the room below to be the same as the air temperature of the test room.
3. Measurements of the air temperature 10 cm above the floor are available, and the report [6] accompanying the data states that the floor surface temperature is about equal to this air temperature. We assumed that the air temperature near the floor was equal to the average room air temperature.

4. Measurements of the inlet water temperature in the ceiling of the test room are available, but not these in the cooled floor (ceiling of the room below). We assumed the water temperature in the cooled floor to be the same as the water temperature in the ceiling.
5. Measurements of the outside air temperature near the building, and at the weather station 20 km away from the building are available. We used the temperatures measured near the building, and the solar radiation measurements from the weather station.
6. The shade operation was "measured," but the fact that the shades were declared open over the last two days does not agree with the measured inside air profile. We used a shade operation calculated based on the 120 W/m^2 threshold for the first 5 days, and simulated the shades as being shut during the weekend.
7. In the program, 57% of the solar radiation entering the space was directed toward the floor, 38% equally distributed among the vertical surfaces, and 5% is reflected back out through the window.
8. In the program, 100 W of the internal load were assumed as generated by occupants (50% = 50 W convection and 50% = 50 W radiation) and 336 W by equipment and lights (30% = 101 W convection and 70% = 235 W radiation); this gives a total of 151 W (35%) convection, and 285 W (65%) radiation from internal loads.
9. In the program, the absorptivity of the window panes is assumed constant, and equal to 5% for both direct and diffuse radiation.

2.3.6 Results

For a comparison of the simulation results with measurements, we chose the measured quantities that we had not used as input data in the program. We thus compared the following pairs of quantities:

1. The air temperature profile in the test room. The measurements provide the air temperature measured at 1.1 m above the test room floor. The simulations provide an average air temperature, representing the temperature of well-mixed air in the test room (Figure 12).
2. The cooled ceiling surface temperature (Figure 13).

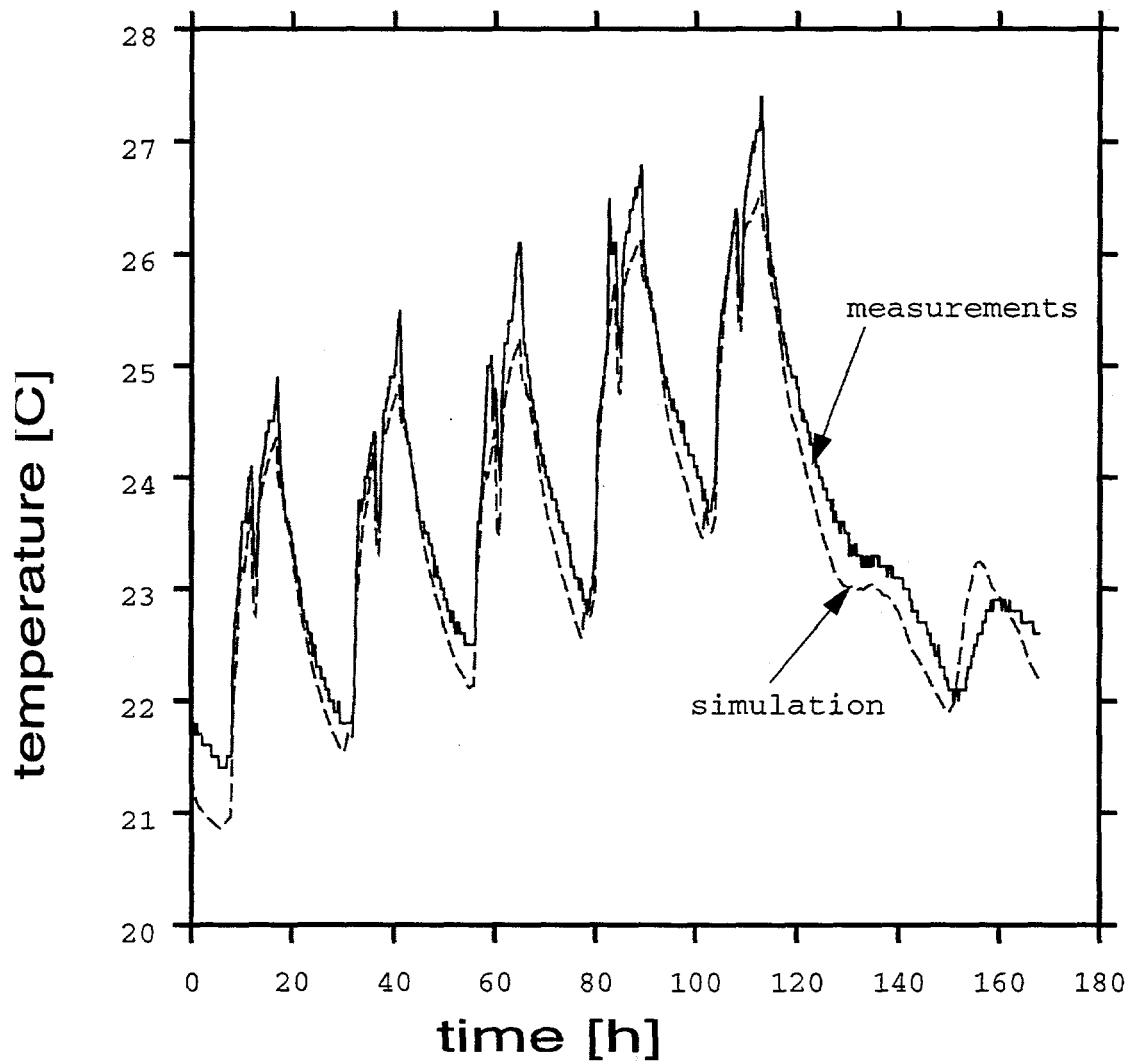


Figure 12. Average air temperature in the DOW-Chemical test room

The simulation results for the room air temperature show good agreement with the air temperature measured at 1.1 m above the floor. The last day presents the highest discrep-

ancy, in that the simulated time of the peak temperature occurs about four hours earlier than the time of the measured peak. This might be caused by a discrepancy between the simulated schedule of the blinds, and the real one.

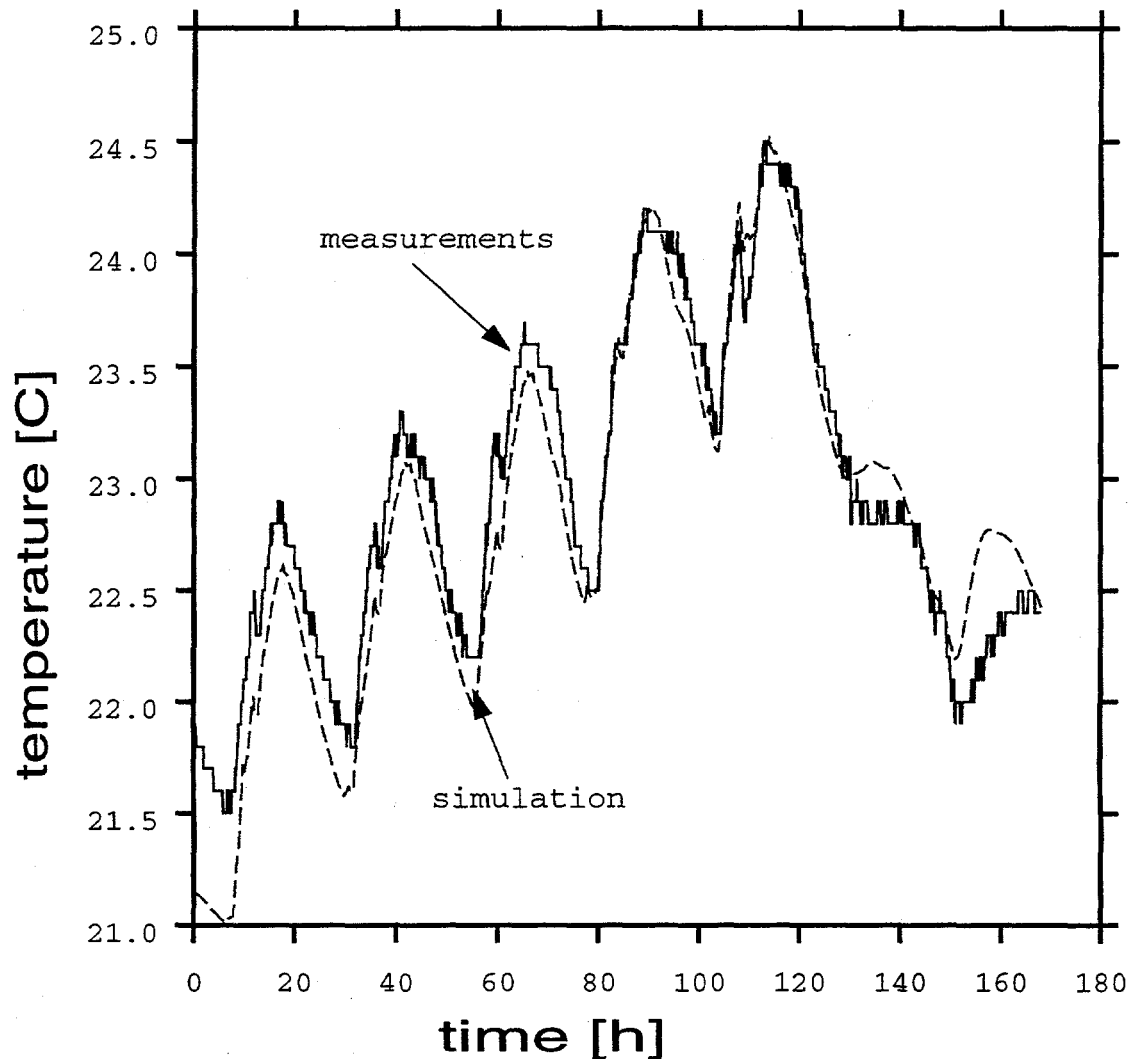


Figure 13. Cooled ceiling surface temperature in the DOW-Chemical test room

The simulation results for the cooled ceiling surface agree well with the measurements. The last two days present, again, the highest discrepancy, which might be caused by a difference between the modeled and the real blinds' operation.

3. Conclusions

3.1 RADCOOL provides reliable results

The results in paragraph 2 show that there is a good agreement between the model and results from analytical problem solving, a different model, or measurements. We are confident that if future RADCOOL modeling is performed in the same manner, the results obtained will be reliable.

3.2 Proposed future development of RADCOOL

A few modules will have to be added to RADCOOL in order to enable it to perform better in a large variety of cases.

3.2.1 Room air stratification

The air stratification that occurs in the case of cooling with radiant ceilings is significantly different from the air stratification that happens in the case of displacement ventilation. The presence of a radiant cooling ceiling produces a relatively uniform vertical temperature profile. However, a steep temperature gradient is created in the vicinity of the ceiling. This gradient plays an important role in the functioning of radiant cooling systems: the higher the gradient, the larger the cooling loads that the system needs to remove. Conversely, if the air in the vicinity of the ceiling is cooled too much, it will move downward, which will impede a displacement system from working efficiently. A secondary effect of air displacement is its bad influence on the air quality: the contaminated air should rise and be removed through the exhaust registers. Downward displacement causes air to be recycled instead.

Results from research in CFD could be used, and expressions could be found that allow the modeling of the air stratification without adding too much computational cost to the program.

3.2.2 Air humidity and condensation at cool surfaces

If the surface temperatures in a room with a radiant cooling system are near the dew point temperature of the ambient air, there is a risk of condensation. Condensation is an undesired effect that may cause damage to the building materials and to the objects in the space. Furthermore, the air humidity is a comfort factor, and moderate relative humidities should be maintained in spaces. A moisture adsorption model for the wall surfaces can be developed [7], which would allow the examination of the relationship between the behavior of thermal and humidity factors.

3.2.3 Thermal comfort and radiant temperature at the occupant location

A SPARK module that allows calculations of the heat exchange between the occupants of a room and the room envelope would be a useful addition to RADCOOL. Once this module is functioning, the user has a complete set of variables that indicate the level of thermal comfort. A similar model for the transfer between the equipment in a space and the room envelope also needs to be added to RADCOOL.

3.2.4 Cooling sources

The present development of RADCOOL allows for ventilation to take place in the test room, but does not state the mechanism by which the cold water is created, nor by which the moisture content of the air in the ventilation ducts is controlled.

There are a number of SPARK modules that model the behavior of cooling sources. Ranval [8] proposed a SPARK module that describes the behavior of a cooling tower. This module can be implemented in RADCOOL. To test several cooling strategies for their performance however, it is necessary that a library of several cooling objects be created.

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6. Nomenclature

α	thermal diffusivity [m^2/s]
ρ	density [kg/m^3]
ΔA	area of the face normal to the heat flux of a two-dimensional element [m^2]
C_t	thermal capacity [$\text{J}/\text{m}^2\text{-K}$]
C_i	thermal capacity of layer i [$\text{J}/\text{m}^2\text{-K}$]
$C_{i,j}$	thermal capacity of two-dimensional element (i,j) [$\text{J}/\text{-K}$]
L	pipe length [m]
Q	heat [W]
R_t	thermal resistance [$\text{m}^2\text{-K}/\text{W}$]
R_i	thermal resistance of layer i [$\text{m}^2\text{-K}/\text{W}$]
$R_{x,i}$	thermal resistance of the horizontal layer i [K/W]
$R_{y,i}$	thermal resistance of the vertical layer i [K/W]
T	temperature [$^{\circ}\text{C}$]
T_i	temperature of layer i [$^{\circ}\text{C}$]
$T_{i,j}$	temperature of two-dimensional element (i,j) [$^{\circ}\text{C}$]
ΔV	volume of a two-dimensional element [m^3]
c_t	heat capacity [$\text{J}/\text{kg-K}$]
k	thermal conductivity [$\text{W}/\text{m K}$]
q	heat flux [W/m^2]
r_{ext}	exterior pipe radius [m]
r_{in}	interior pipe radius [m]