Simplified Modeling of Indoor Dynamic Temperature Distributions

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It is well-known that there exist indoor temperature distributions. To have more precise predictions of indoor thermal comfort and better control of indoor thermal conditions, a both detailed and fast model of the dynamic indoor temperature distributions is needed. Unfortunately, very few papers studied such models due to the complexity of fluid (air) flows. CFD can be used as a detailed model. But it is too time consuming. This paper discusses two models in this respect, the fixed-flow-field model and air-zonal model. Both models are validated with experimental results. In order to design better control systems, the zonal model is converted into a state space representation form which can be easily used by control system designers.

1. Introduction

In traditional models of indoor thermal responses, the indoor air volume is assumed to be "perfectly-mixed" and no temperature distribution is considered. Such models are obviously too crude and not suitable for precise indoor thermal comfort predictions and optimal control system designs.

It is well-known that there exist indoor temperature distributions (Figure 1) in reality. The temperature of the "working zone" of the room occupant is usually different from that of the "discharge zone" of the conditioning air or that of the "temperature sensor zone". Our literature study shows that almost all air-conditioning control systems have not taken the temperature distributions into consideration since a dynamic model in such respect is simply not available. This negligence of temperature variations is one of the main reasons of the malfunction of the air-conditioning control system, which in

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turn causes the complaints of the room occupant about indoor thermal comfort. Therefore to have more precise predictions of indoor thermal comfort and better control of indoor thermal conditions, a both detailed and fast model of the dynamic indoor temperature distributions is needed.

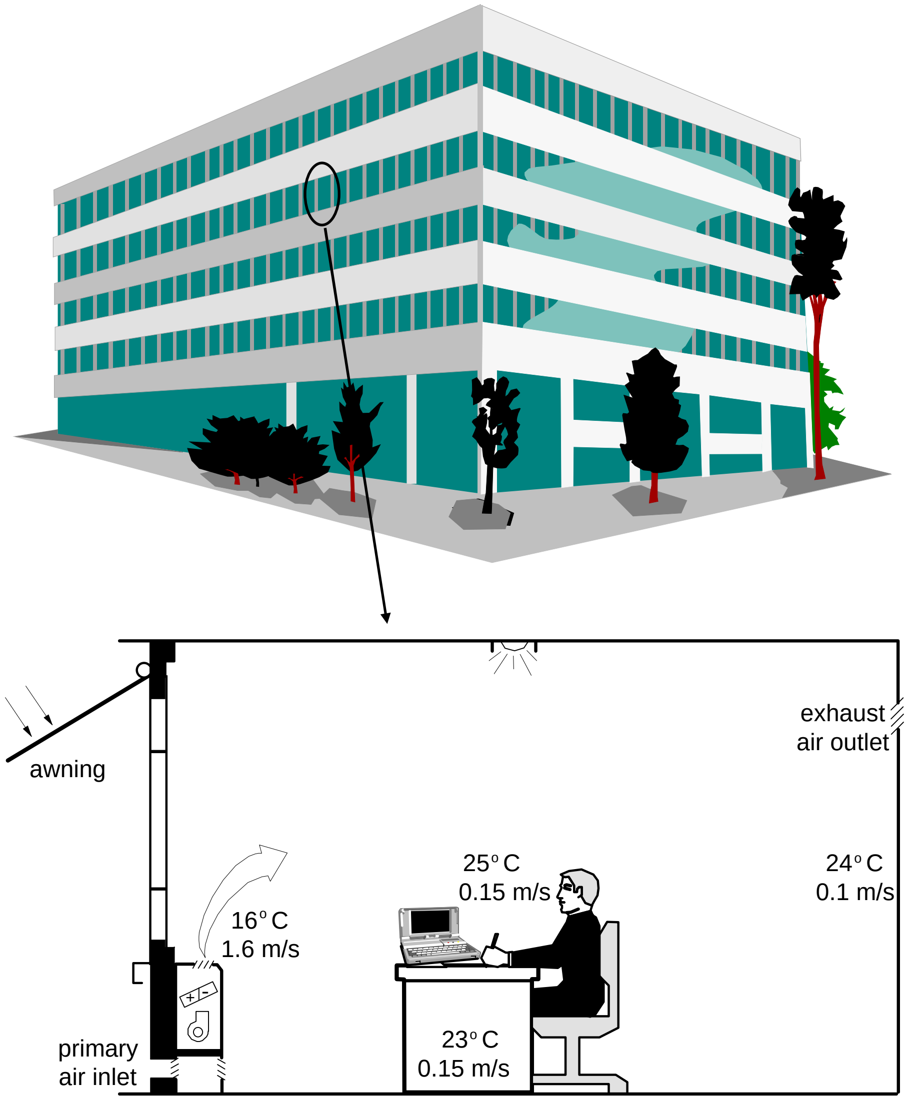

Figure 1 Indoor temperature distribution of a room

Hemmi (1967) developed a model which divided the room air volume into several ideally mixed zones that were connected both in series and parallel with respect to the flow in the room. van der Kooi and Förch(1985) presented a model in which the room center (in the middle of the ceiling and floor) was assigned a temperature T_a . Since there was indoor temperature gradient, the air temperature near the ceiling was represented with $T_a + \Delta T_c$ and the air temperature near the floor was represented with $T_a + \Delta T_f$. ΔT_c and ΔT_f were measured first. The room was then represented with three indoor temperatures at different heights, which could easily be calculated with fixed values of ΔT_c and ΔT_f since only one temperature T_a is unknown. In the same paper, van der Kooi presented a more-point model in which the room air volume was divided into several air zones, each of which was represented with one temperature.

The precondition for the models of Hemmi and van der Kooi was that the air mass flow between one zone and another could be prescribed. But both Hemmi and van der Kooi did not give a theoretical method of how to prescribe these mass flows. The method they gave was measurement, i.e. the air mass flows were estimated according to the visualization of indoor air flows and the measurement of air speeds.

Hill (1985) described a model based on the one-point model of Borrresen (1981). Although the total air mass was represented with two temperatures, the room air was still treated as fully mixed and represented with one temperature. The second temperature was that of the air in the air duct.

Chen and van der Kooi (1987, 1988) presented the idea of combining a cooling load program and an air flow program in order to find better agreement between computed and measured results of annual heating/cooling loads.

Dalicieux (1992) presented a simplified model to represent indoor air motions based on the degradation of fluid mechanics equations. In his paper, the room air volume was divided into two parts: the plume areas where the supply air inlet or heating panel was located and the standard areas which were the remaining parts and where the air movement was slow. The mass flow between one part and an adjacent part is calculated based on Bernoulli's law.

Peng, Chen and van Paassen (Peng *et al* 1994, Chen *et al* 1995) studied the prediction of indoor dynamic temperature distributions by using a fixed-flow-field obtained from a CFD calculation. A good agreement between calculations and measurements was obtained.

With the advent of faster and faster computers in recent years, CFD (computational fluid dynamics) method is more and more used to investigate indoor temperature distributions and air flows. The fundamental differential equations in CFD are the continuity, momentum conservation and energy conservation equations. The basic idea of CFD is that the flow domain is first divided into thousands of finite volumes by setting a grid (Figure 2). For every finite volume of the grid, the conservation equations are solved iteratively until the solutions of all variables for all finite volumes have converged.

In contrast to the perfectly-mixed air model, CFD goes to the other way round — too complicated theory and too time consuming computation. Until recently, most published CFD results for predicting indoor temperature distributions and air flows are steady state simulations. Due to the speed limitation of the present-day widely available computers such as work stations, PC-486s, PC-pentiums, etc., dynamic CFD simulations for the thermal responses of buildings are seldom seen in literature. On the other hand, it is impracticable to use CFD as a dynamic model for control system designs. A compromise model between the perfectly-mixed air model, which is simple but dynamic, and CFD model, which is the complicated but more realistic, should be found out for the present time. It could be anticipated that during the search of such a dynamic model, some assumptions will have to be made and some information will have to be sacrificed.

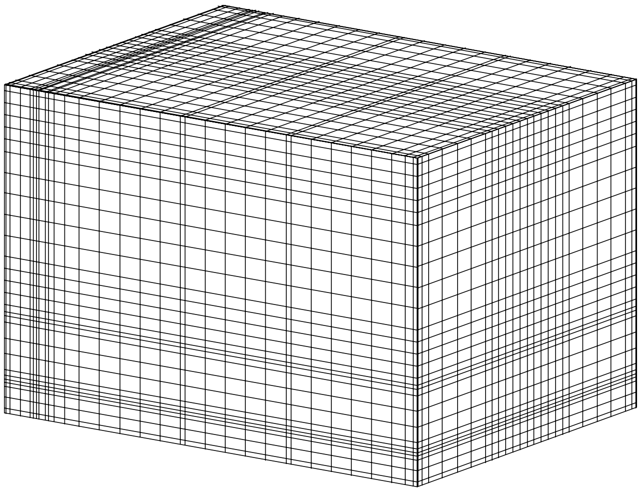

Figure 2 Grid used by a CFD calculation

2. Fixed-Flow-Field Model

We may have noticed such a phenomenon from our experience of daily life and experiments that for a typical heating (or cooling) situation, although the indoor air flow in some room locations may vary with time due to the existence of turbulence and disturbances, the prevailing air flow field that has dominant effects on air mass transportation and temperature distributions is relatively stable (Peng *et al* 1994, Chen *et al* 1995, Peng 1996). This phenomenon is especially apparent when the heating (or cooling) energy is supplied through an air-conditioning unit with forced air flows.

From the experience of using CFD codes, we also know that due to the nonlinearity of the momentum equations, under-relaxation has to be introduced in order to get converged results. This means that we intentionally slow down the solving process in order to get the correct air velocities. In fact, it is the solving process of the momentum equations (the velocity components) that takes most of the computing time of a CFD simulation.

What we face now is such a situation: on the one hand, the solving process of velocities takes huge amount of time; on the other hand, the indoor air flow pattern does not change much for forced convection air flows. Then, why should we spend so much computing time on calculating the air velocities at every time step. If we could use a fixed air flow field for dynamic temperature calculations, then only the energy equation is left to be solved and a great of amount of computing time should be saved.

From books on CFD we know, the energy conservation equation is:

$$\frac{\partial \rho\theta}{\partial t} + \mathbf{div}\left(\rho \vec{V} \theta - \Gamma_{\mathrm{H,eff}} \,\mathrm{grad}\theta\right) = \frac{S_H}{C_p} \tag{1}$$

where $\Gamma_{\mathrm{H,\,eff}} = \lambda/C_p + \rho\nu_t/\sigma_H$ is the effective exchange coefficient,

θ = temperature of the finite air volume [K or ^{o}C],

ρ = density of air [kg/m^3],

t = time [s],

S_H = heat generation rate per unit volume [W/m^3]

C_p = specific heat of air [J/kg·K],

λ = thermal conductivity of air [W/m·K],

ν_t = turbulent viscosity [m^2/s],

σ_H = turbulent Prandtl number (dimensionless)

If the velocity field $\vec{V}$ and turbulence viscosity ν_t are calculated with a CFD code (PHOENICS is used in this paper), it would be much easier to solve the energy equation (1). We call this method Fixed-Flow-Field (FFF) model.

When equation (1) is discretized, the following difference equation can be obtained:

$$a_P\theta_P(t) = a_E\theta_E(t) + a_W\theta_W(t) + a_N\theta_N(t) + a_S\theta_S(t) + a_T\theta_T(t) + a_B\theta_B(t) + b \tag{2}$$

Detailed descriptions of the coefficients a_p, a_E, a_W, a_N, a_S, a_T, a_B and b can be found in Patankar (1980).

To validate this model, some dynamic measurements are made in a test room. The measures of the room are as follows (Figure 3):

Internal (L × W × H)	=	4.1 m × 3.1 m × 2.7 m
External (L × W × H)	=	4.4 m × 3.4 m × 3.0 m

Construction and materials of walls, roof and floor:

Walls and roof:	Polystyrene with steel plate layers on both sides.
Floor:	Soil, polystyrene, plywood with aluminum plate on the top.

Hear source box:	=	0.50 m × 0.60 m × 0.40 m
Fan-coil unit:	=	0.45 m × 1.00 m × 0.80 m

The heat source is to emulate the heat generated from a computer and a seated person. An incandescent lamp of 150 W is mounted inside the heat source box.

The fan-coil unit is installed below the window (Figures 3 and 4). Hot water and chilled water are supplied to the coil (heat exchanger). The flow rates of the hot and chilled water can be changed by the control device (controller) through two proportional solenoid valves. The size of the supply air grill at the top of the fan-coil unit is 0.80 m × 0.13 m. The direction of the grill can be changed so that the supply air from the fan-coil unit can be guided either toward the room or toward the window.

Figure 4 shows the locations of three thermocouples in the central plane of the room. The heights of the thermocouples are arranged in such a way that the temperatures at the ankle and neck positions of the room occupant (when seated) and the temperature at the head position (when standing) should be measured. For clarity, the names “high”, “mid” and “low” are assigned to the thermocouples to reflect their positions.

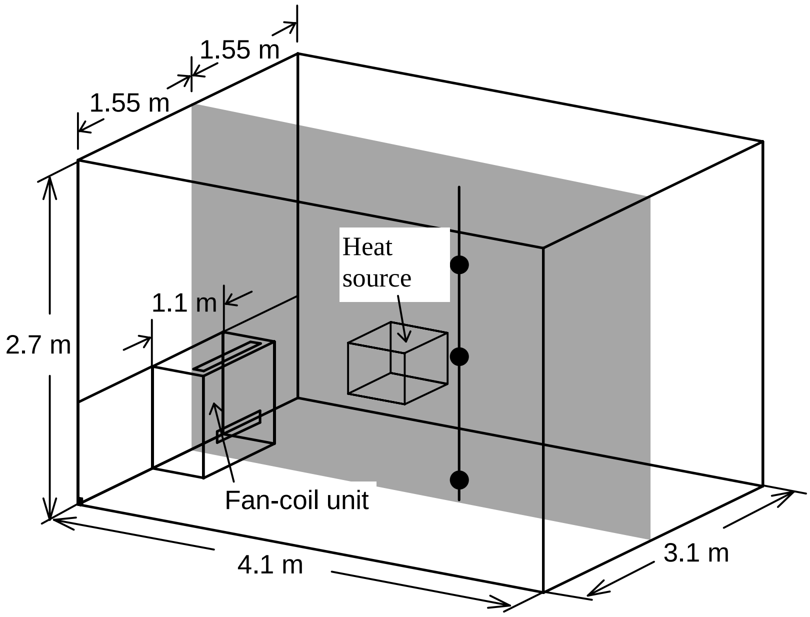

Figure 3 Test room

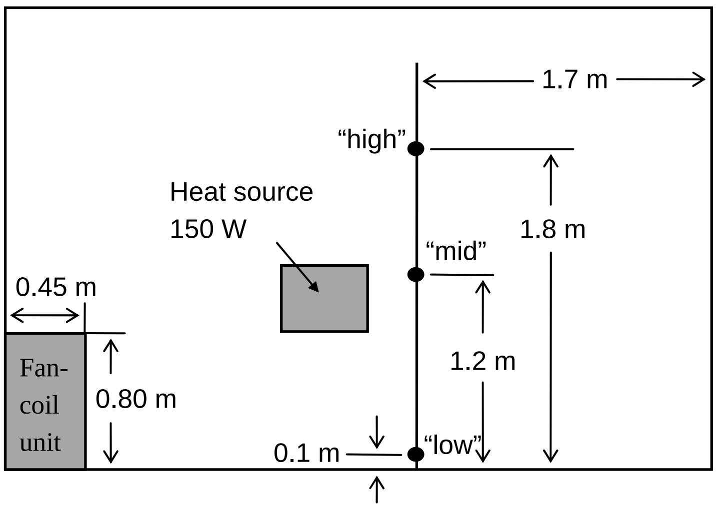

Figure 4 Arrangement of thermocouples

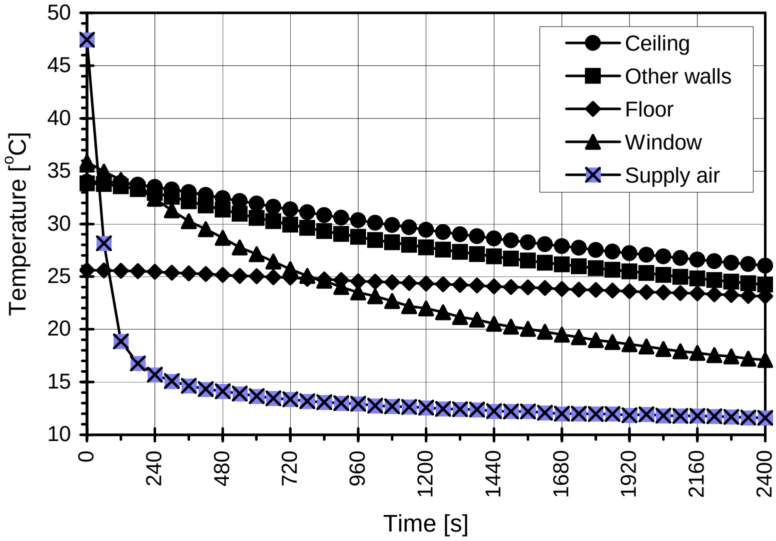

Figure 5 Boundary surface temperatures

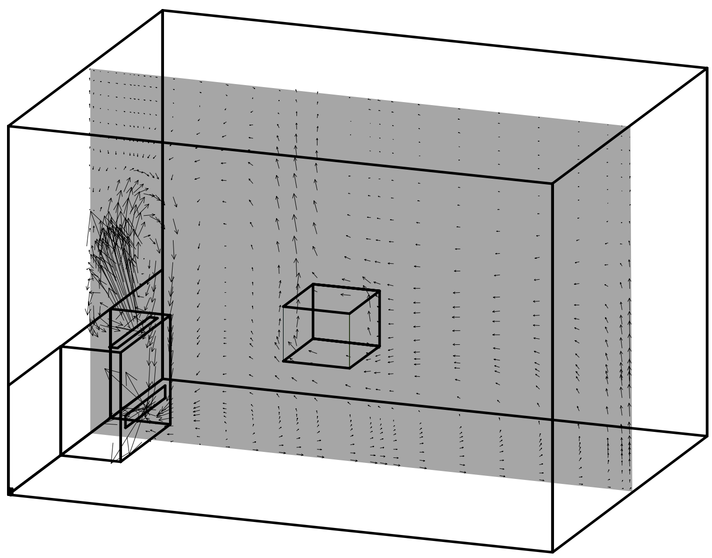

Figure 6 Steady-state air flow pattern at the center plane

To calculate the temperatures of indoor air, internal surface temperatures of the walls as well as the ceiling, floor and window need to be known. There are two ways to obtain the surface temperatures: through calculation and through measurement. To validate the model, any additional and unnecessary computation error should be reduced as much as possible. Thus the surface temperatures are measured here and are shown in Figure 5.

The situation shown in Figure 5 is a cooling case. Cooling air is supplied at low fan speed (the averaged supply air speed = 1.64 m/s) and is guided toward the window. The calculated steady-state air flow pattern at the center plane is shown in Figure 6.

With the calculated velocity field and effective exchange coefficients, the coefficients of the energy equation (2) can be calculated. The indoor dynamic air temperature distributions are then calculated with equation (2). Here, the value of 3.0 $W/m^2{\cdot}K$ is adopted for the convective heat transfer coefficients between the internal surfaces and the adjacent air (van Paassen 1981). Figures 7, 8 and 9 illustrate the calculated temperatures versus the measured temperatures at the three thermocouple positions, where "Cal" stands for "Calculation" and "Mea" stands for "Measurement".

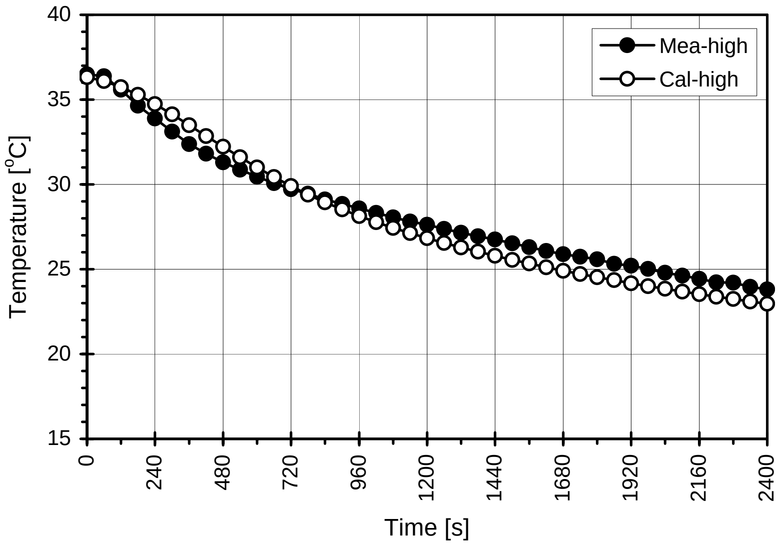

Figure 7 Calculation *vs* measurement at "high" thermocouple position

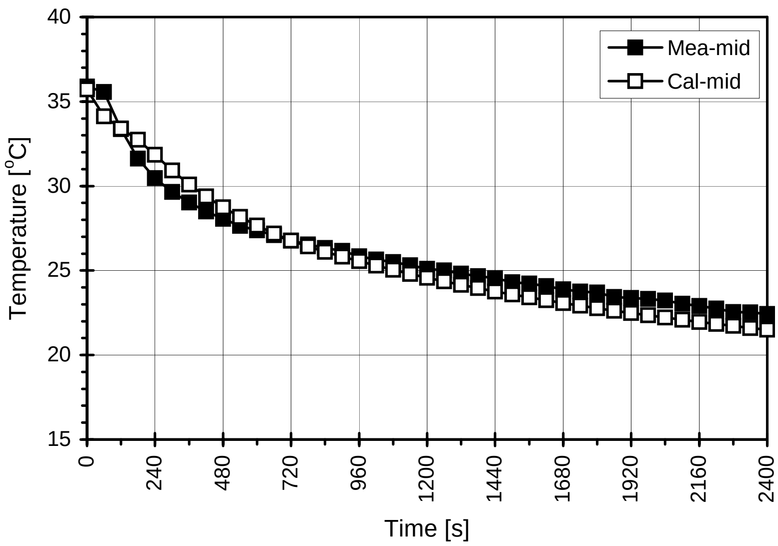

Figure 8 Calculation *vs* measurement at "mid" thermocouple position

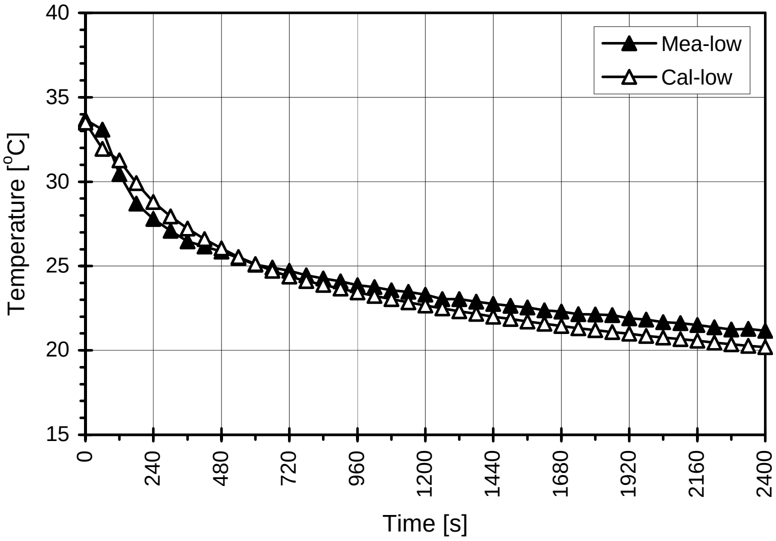

Figure 9 Calculation *vs* measurement at "low" thermocouple position

For the measured temperature responses, the mean radiant temperatures (MRT) of the surrounding surfaces to the three thermocouples have also been taken into account at every time step. Details on such consideration can be found in Peng (1996).

Compare the measured and calculated responses, we see the indoor dynamic temperature distributions (temperature responses and gradient) are well predicted. The computing time for such a dynamic simulation is about 20 CPU minutes on a SUN-Sparc (IPX) station, less than 6% of the CPU time consumed by the dynamic CFD simulation in one time step.

3. Air-Zonal Model

From the standpoints of detailed predictions of dynamic indoor temperature distributions, the Fixed-Flow-Field model really gives more realistic results than the well-mixed-air model, and faster results than the dynamic CFD model. However, since it still needs the calculation of hundreds of thousands of coefficients and temperatures, this method is inconvenient to use and some improvement should be made on it.

As a matter of fact, we are not interested in too detailed information of indoor temperature distributions since most people are insensitive to minor temperature changes (e.g. < 0.5 ^{o} C) inside the room. Generally speaking, there exist significant temperature differences between the working zone where the room occupant sits and works and the heat/cold source zone where the air-conditioning unit is located. Another obvious phenomenon is the temperature gradient of indoor air, which influences the local thermal comfort and the energy consumption of the room.

Thus, the division of the room into several air zones would give enough temperature information and accuracy for simulations and control system designs. If this could be done, the computing time would be further reduced dramatically and a much simpler and easy to use dynamic model would be available. But the critical problem is how to find the mass and heat exchanges between one zone and another (Figure 10).

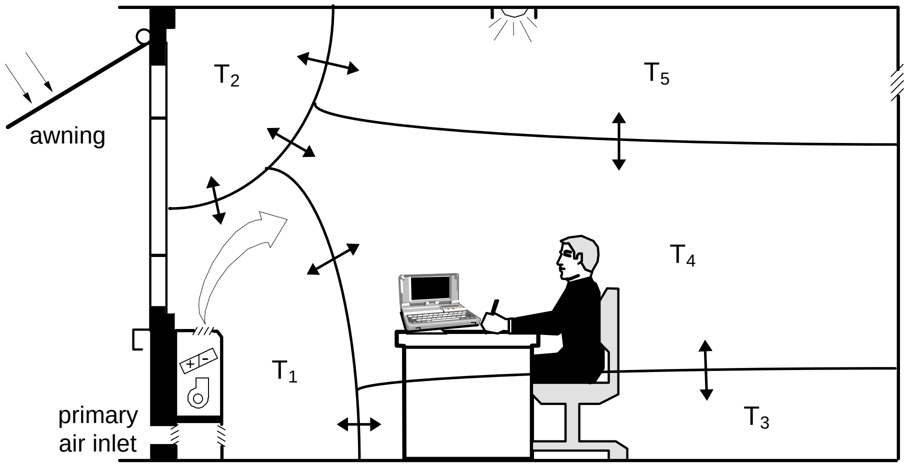

Figure 10 How to find the mass and heat exchanges between two air zones

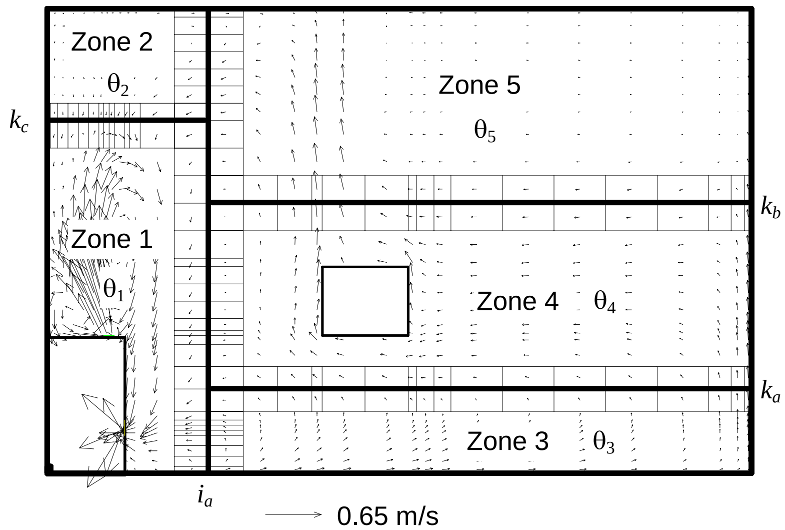

Figure 11 Air zones for cooling case and their borders

According to the CFD theory, if the mass is balanced for every single finite air volume of the grid, then the mass is also balanced for any arbitrary group of finite air volumes. Starting from this point, we may reorganize the thousands of finite volumes used by CFD simulations and transform them into several air zones (Figure 11). By supposing that each air zone is well mixed and is depicted with one temperature, the room thermal response model can be represented with several temperatures and thus be much simplified.

It is obvious that the border between two adjacent air zones is also the border of the grid cells that are located on both sides of the border. The velocities $\vec{V}$ and exchange coefficients $\Gamma_{H,\ eff}$ of these grid cells can be used to calculate the mass and heat exchanges between two air zones.

Suppose each air zone is well mixed and only has one temperature. For zones 1 to 5, which have air temperatures θ_1, θ_2, θ_3, θ_4 and θ_5, and air volumes V_1, V_2 , V_3, V_4 and V_5, we have:

$$\frac{\partial V_1 \rho C_p \theta_1}{\partial t} + a_{P,1}\theta_1 = a_{T,12}\theta_2 + a_{E,13}\theta_3 + a_{E,14}\theta_4 + a_{E,15}\theta_5 + b_1 \tag{3}$$

$$\frac{\partial V_2 \rho C_p \theta_2}{\partial t} + a_{P,2}\theta_2 = a_{B,21}\theta_1 + a_{E,25}\theta_5 + b_2 \tag{4}$$

$$\frac{\partial V_3 \rho C_p \theta_3}{\partial t} + a_{P,3}\theta_3 = a_{W,31}\theta_1 + a_{T,34}\theta_4 + b_3 \tag{5}$$

$$\frac{\partial V_4 \rho C_p \theta_4}{\partial t} + a_{P,4}\theta_4 = a_{W,41}\theta_1 + a_{B,43}\theta_3 + a_{T,45}\theta_5 + b_4 \tag{6}$$

$$\frac{\partial V_5 \rho C_p \theta_5}{\partial t} + a_{P,5}\theta_5 = a_{W,51}\theta_1 + a_{W,52}\theta_2 + a_{B,54}\theta_4 + b_5 \tag{7}$$

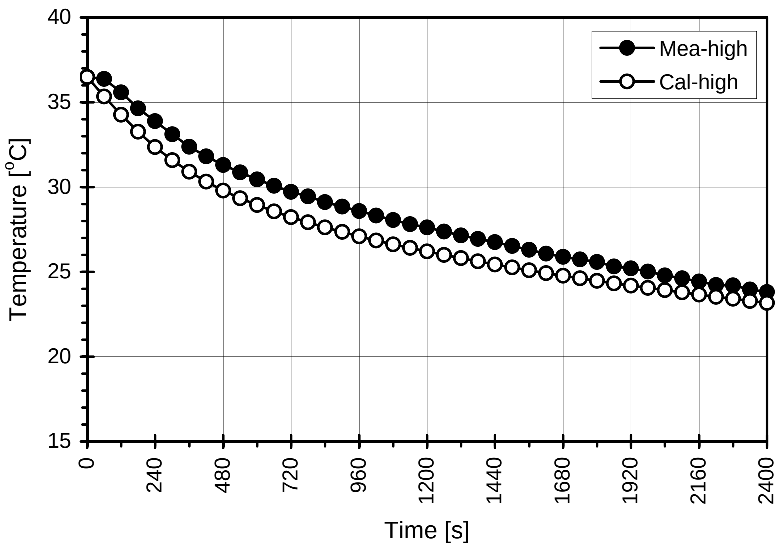

Figure 12 Calculation *vs* measurement at zone 5

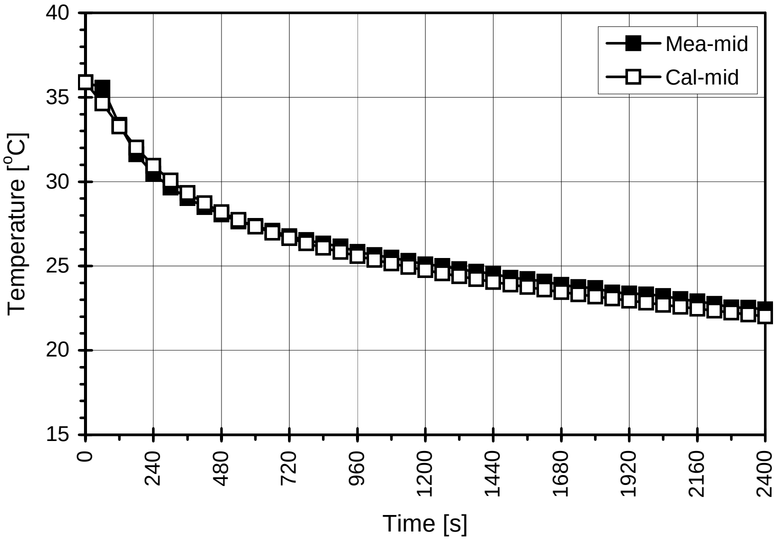

Figure 13 Calculation *vs* measurement at zone 4

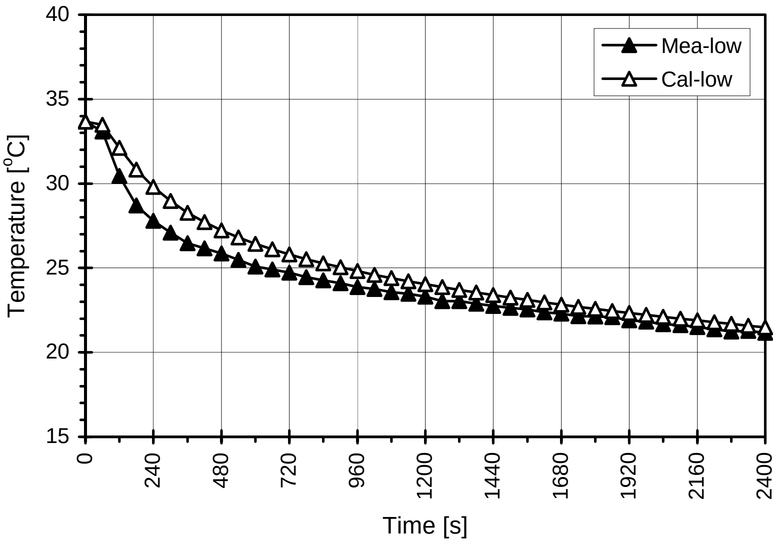

Figure 14 Calculation *vs* measurement at zone 3

where detailed descriptions of the coefficients like $a_{T,\,12}$, $a_{E,\,13}$, etc. can be found in Peng (1996).

<u>*Model Validation*</u>:

For the cooling cases stated above, the test room can be classified into air zones as follows (length by width by height):

Zone 1: 1.14 m × 3.1 m × 2.15 m
Zone 2: 1.14 m × 3.1 m × 0.55 m
Zone 3: 2.96 m × 3.1 m × 0.62 m
Zone 4: 2.96 m × 3.1 m × 1.11 m
Zone 5: 2.96 m × 3.1 m × 0.97 m

The comparisons between the calculated results and the measured results are shown in Figures 12, 13 and 14.

From the compared results, we may see that the indoor dynamic temperature distributions are well predicted with the zonal model. But the computing time has been reduced to less than 1 CPU second on a PC-486 (66 Mhz) computer.

4. State Space Representation of the Air-Zonal Model

Whenever a control system is designed, a model in the form of either transfer function or state space is often needed. The air-zone model presented in the previous section can be easily converted into a state space model.

To be consistent with the modern control theory, we now change the definition of x from coordinate of distance to states in a space and let it denote temperatures of air zones:

$$x_1 = \theta_1,\ x_2 = \theta_2,\ x_3 = \theta_3,\ x_4 = \theta_4,\ x_5 = \theta_5, \tag{8}$$

$$\dot{x}_1 = \frac{\partial \theta_1}{\partial t},\ \dot{x}_2 = \frac{\partial \theta_2}{\partial t},\ \dot{x}_3 = \frac{\partial \theta_3}{\partial t},\ \dot{x}_4 = \frac{\partial \theta_4}{\partial t},\ \dot{x}_5 = \frac{\partial \theta_5}{\partial t}, \tag{9}$$

$$a_1 = V_1\rho C_p,\ a_2 = V_2\rho C_p,\ a_3 = V_3\rho C_p,\ a_4 = V_4\rho C_p,\ a_5 = V_5\rho C_p \tag{10}$$

Then equations (3), (4), (5), (6) and (7) become:

$$\dot{x}_1 = -\frac{a_{P,1}}{a_1}x_1 + \frac{a_{T,12}}{a_1}x_2 + \frac{a_{E,13}}{a_1}x_3 + \frac{a_{E,14}}{a_1} + \frac{a_{E,15}}{a_1}x_4 + \frac{b_1}{a_1} \tag{11}$$

$$\dot{x}_2 = \frac{a_{B,21}}{a_2}x_1 - \frac{a_{P,2}}{a_2}x_2 + \frac{a_{E,25}}{a_2}x_5 + \frac{b_2}{a_2} \tag{12}$$

$$\dot{x}_3 = \frac{a_{W,31}}{a_3}x_1 - \frac{a_{P,3}}{a_3}x_3 + \frac{a_{T,34}}{a_3}x_4 + \frac{b_3}{a_3} \tag{13}$$

$$\dot{x}_4 = \frac{a_{W,41}}{a_4}x_1 + \frac{a_{B,43}}{a_4}x_3 - \frac{a_{P,4}}{a_4}x_4 + \frac{a_{T,45}}{a_4}x_5 + \frac{b_4}{a_4} \tag{14}$$

$$\dot{x}_5 = \frac{a_{W,51}}{a_5} x_1 + \frac{a_{W,52}}{a_5} x_2 + \frac{a_{B,54}}{a_5} x_4 - \frac{a_{P,5}}{a_5} x_5 + \frac{b_5}{a_5} \tag{15}$$

Representing equations (11) through (15) with a state space model we have:

$$\dot{\boldsymbol{x}} = \mathbf{A}\boldsymbol{x} + \mathbf{B}\boldsymbol{u} \tag{16}$$

where

$$\mathbf{A} = \begin{bmatrix} -\frac{a_{P,1}}{a_1} & \frac{a_{T,12}}{a_1} & \frac{a_{E,13}}{a_1} & \frac{a_{E,14}}{a_1} & \frac{a_{E,15}}{a_1} \\ \frac{a_{B,21}}{a_2} & -\frac{a_{P,2}}{a_2} & 0 & 0 & \frac{a_{E,25}}{a_2} \\ \frac{a_{W,31}}{a_3} & 0 & -\frac{a_{P,3}}{a_3} & \frac{a_{T,34}}{a_3} & 0 \\ \frac{a_{W,41}}{a_4} & 0 & \frac{a_{B,43}}{a_4} & -\frac{a_{P,4}}{a_4} & \frac{a_{T,45}}{a_4} \\ \frac{a_{W,51}}{a_5} & \frac{a_{W,52}}{a_5} & 0 & \frac{a_{B,54}}{a_5} & -\frac{a_{P,5}}{a_5} \end{bmatrix} \tag{17}$$

$$\mathbf{B} = \begin{bmatrix} \frac{1}{a_1} & 0 & 0 & 0 & 0 \\ 0 & \frac{1}{a_2} & 0 & 0 & 0 \\ 0 & 0 & \frac{1}{a_3} & 0 & 0 \\ 0 & 0 & 0 & \frac{1}{a_4} & 0 \\ 0 & 0 & 0 & 0 & \frac{1}{a_5} \end{bmatrix} \tag{18}$$

$$\boldsymbol{x} = [x_1, x_2, x_3, x_4, x_5]^{\mathrm{T}} \tag{19}$$

$$\boldsymbol{u} = [b_1,\ b_2,\ b_3,\ b_4,\ b_5]^{\mathrm{T}} \tag{20}$$

where T represents transpose. **A** is usually called system matrix. **B** is named as input matrix. Vector ***u*** contains the heat flux from the supply air.

The output vector is normally defined as:

$$\boldsymbol{y} = \mathbf{C}\boldsymbol{x} \tag{21}$$

where

$$\boldsymbol{y} = \left[y_1, y_2, y_3, y_4, y_5\right]^{\mathrm{T}} \tag{22}$$

$$\mathbf{C} = \begin{bmatrix} c_{11} & c_{12} & c_{13} & c_{14} & c_{15} \\ c_{21} & c_{22} & c_{23} & c_{24} & c_{25} \\ c_{31} & c_{32} & c_{33} & c_{34} & c_{35} \\ c_{41} & c_{42} & c_{43} & c_{44} & c_{45} \\ c_{51} & c_{52} & c_{53} & c_{54} & c_{55} \end{bmatrix} \tag{23}$$

The elements of the output (or measurement) matrix **C**, c_{ij} (i, j = 1, 2, ..., 5), are determined according to the relation between the outputs and the states. For example, if the output vector $\boldsymbol{y}$ is equal to the state vector $\boldsymbol{x}$, then **C** = **I**, where **I** is the unit matrix. If the system only has a single output, then C may be simply written as:

$$\mathbf{C} = \left[c_1, c_2, c_3, c_4, c_5\right] \tag{24}$$

If the system output is the air temperature of zone 4, then:

$$\mathbf{C} = \left[0, 0, 0, 1, 0\right] \tag{25}$$

If the output is the mean value of the air temperatures of zone 3 and zone 4, then:

$$\mathbf{C} = \left[0, 0, 0.5, 0.5, 0\right] \tag{26}$$

5. Conclusions

The traditional modeling assumption of perfectly-mixed indoor air volume can no longer satisfy the requirement of better control system designs of indoor thermal conditions.

For the depiction of dynamic responses of indoor temperature distributions, CFD method should, theoretically, give the best prediction results. But it takes too much computing time and is not cost effective for the dynamic control investigation of indoor thermal conditions.

For rooms installed with air-conditioning units, the indoor air flow patterns are mainly determined by the speed and direction of the supply air. If these two factors do not change, then the indoor air flow field can be thought of as time invariant (fixed). Based on this assumption, the air flow field needs to be solved only once. What is left to be solved is the energy equation. This forms the main idea of the simplified modeling methods of this paper.

In this paper, two simplified dynamic models are presented: the direct solution of the energy equations of all finite air volumes based on fixed flow field (FFF model), and the air-zonal model. The state space model is based on the air-zonal model. Each of the models is validated with experimental results. The validations indicate that the models can give satisfactory and realistic predictions of the indoor dynamic temperature distributions. The total computing time used by the air-zonal model (also state space model) is less than 1 second on a PC-486-(66 MHz) computer, or less than 0.0024% of that used by the dynamic CFD calculation in one time step (the speed of a PC-486-(66MHz) is comparable to that of a SUN-Sparc-IPX station).

The biggest advantage of the state space model is that it can be easily used for control system designs. It can also be transformed into transfer functions so that the frequency response analysis can be carried out. With a state space model, existing modern control theories can find better applications in the prediction and control of the indoor thermal conditions. Another advantage of the state space model is that it can be easily used in the popular software MATLAB/Simulink for any control system simulations.

Peng (1996) gives two illustrations of how the state space model can be used. One example is that it is used for designing state estimators which is then used for temperature predictions in real time. The other example is the precise control of the temperature in the working zone of the room.

Unlike other simplified models of indoor temperature distributions, the zonal model developed in this paper does not ignore the turbulence effect on heat exchanges among air zones. Each element of matrix **A** in the state space model contains the effect of the heat transfer due to air mass flows as well as the heat exchange effects due to turbulence and the free movement of air molecules. Therefore, the elements of matrix **A** can be artificially changed within a certain extent without influencing the mass balance condition of each air zone. The noteworthy point is that not one but two symmetric elements in matrix **A** should be changed simultaneously. The equivalent result is that the turbulent effect is enhanced or attenuated. Such changes might be necessary for model fitting (curve fitting).

Possible applications of the state space model developed in this thesis include:

1. Fast predictions of dynamic indoor temperature distributions.
2. Detailed and fast predictions indoor thermal comfort levels at different room locations.
3. Control system designs and simulations
4. Finding the transfer functions relating the supply air temperature and the temperatures at various room locations.
5. Finding the optimal control of the temperature sensor.
6. Finding control rules of a fuzzy logic controller
7. More precise prediction of the energy consumption of building rooms.

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