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PAPER NO. 42

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SUMMARY

The paper deals with experimental and theoretical examinations of the vertical temperature gradient in rooms with convective flows under transient conditions. The measurements are carried out in a laboratory test room of three different sizes. A small room (7.25 m^2) with a normal room height of 2.4 m (only the result of this experiment is described in the paper), a small room with a large room height of 3.3 m and a room (10.70 m^2) with a room height of 3.3 m.

For normal rooms with only convective air flows it is concluded that it is possible to find two dimensionless temperature profiles, one for periods with rising air temperatures and one for periods with falling air temperatures (heating/cooling periods). The dimensionless gradients are possibly independent of the room dimension because of the turbulence intensity in the boundary layer. By using the dimensionless temperature profiles it is possible to find the horizontal location of a simulated mean air temperature and with a good approximation it is possible to determine the actual vertical temperature profile, even under transient conditions, if only two air temperatures are known (simulated), an air temperature in an upper zone and lower zone of the room, respectively. Furthermore, it is concluded that this is a better description of the vertical temperature gradient than a linear description, which is often used when only two air temperatures are known.

INTRODUCTION

Our knowledge of and the demands on the indoor climate have grown very fast for the last decades. The demands on the indoor climate are becoming more important because more and more people are working under indoor conditions. Therefore, it must be a human right not to be exposed to bad indoor climate. Specially because we cannot be sure that bad indoor climate cannot be a long term healthy risk for the people being exposed.

Another thing is that we for the last decades have come to see energy consumption in another light. Since the early 70'th there have been two energy crises and we are aware of the fact that the climate of the earth is influenced by the global consumption of fossil energy. For lowering the energy consumption this has affected that our buildings are being better insulated and are more tight. At the same time the rapid growth in the use of electronics or other electrical equipments has resulted in more internal heat gains.

Seen in that light it is very important that every new large building where people are working is designed with a good indoor climate. This has resulted in a wish for more accurate calculations of the indoor climate before the final technical solution of a building is chosen.

An accurate modelling of the dynamic thermal behaviour of a room requires the solution of a thermal network representing all convective and radiant heat transfers within the room. The method involves the solution of a set of equations: one for each heat transfer surface and one for the room air. In this way the room air is assumed to have one homogeneous temperature which is never the case in a normal occupied room. To calculate the room temperature gradients it requires not one but several equations for the room air, one for each air node, and it also requires a calculation of the convective energy transport between every air node in the room. The consumption of CPU time is increased enormously to make an accurate calculation that involves modelling of the air flow patterns for determination of the convective air transport in the room.

In this paper it is shown that in a relatively simple way it is possible to calculate a vertical average gradient in a room with only convective air flows. The model is validated by comparing simulated results with measured temperatures in two laboratory test rooms. The model described in this paper is implemented in the simulation programme Suncode which is a PC-version of the mainframe programme SERIE-RES [1]. The necessary chances taken to implement the model in Suncode are described in [2, 3]. Except these changes the main structure of the original programme is unchanged.

DESCRIPTION OF A TWO-ZONE ROOM MODEL

As described in [2, 3] there is made a two-zone room model that calculates the exchange of air and energy between an upper zone and the occupied zone of a room. Normally Suncode and other transient simulation programmes only calculate one temperature for the air in the room. This temperature is regarded as the mean air temperature in the room.

Room model with two air nodes

By using a two-zone air model the calculated air temperatures are also mean air temperatures. In the two-zone case the calculated temperatures are the mean temperature for the upper zone and the mean temperature for the occupied zone, both for the air and the vertical surfaces. The connection R_7 in figure 1, in the two-zone model, describes the exchange of convective energy and air mass between the upper and the occupied zone.

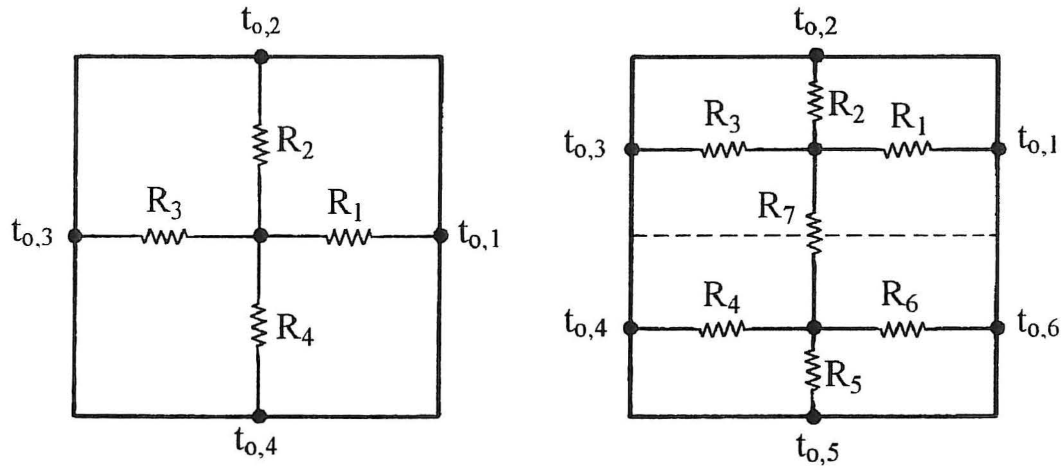

Figure 1. Connections between air node and surface in one and two-zone models.

The air movements between the upper zone and the occupied zone are caused by temperature differences between the air and the surfaces in the zone or by air plumes generated over local heat sources (figure 2).

For every time step in the transient calculation the air movement along the vertical surfaces and over heat sources is determined. If there is a difference in the upstream and the downstream air masses it is accepted that this difference is exchanged directly between the zones, so that the total sum of air masses passing the fictive line (figure 2) that separates the zones is equal to zero.

$$\sum G_{up} - \sum G_{down} + G_{direct} = 0 \quad (1)$$

G_{up} upstream air mass (kg/s)

G_{down} downstream air mass (kg/s)

G_{direct} air mass directly exchanged between zones for preservation balance of mass (kg/s)

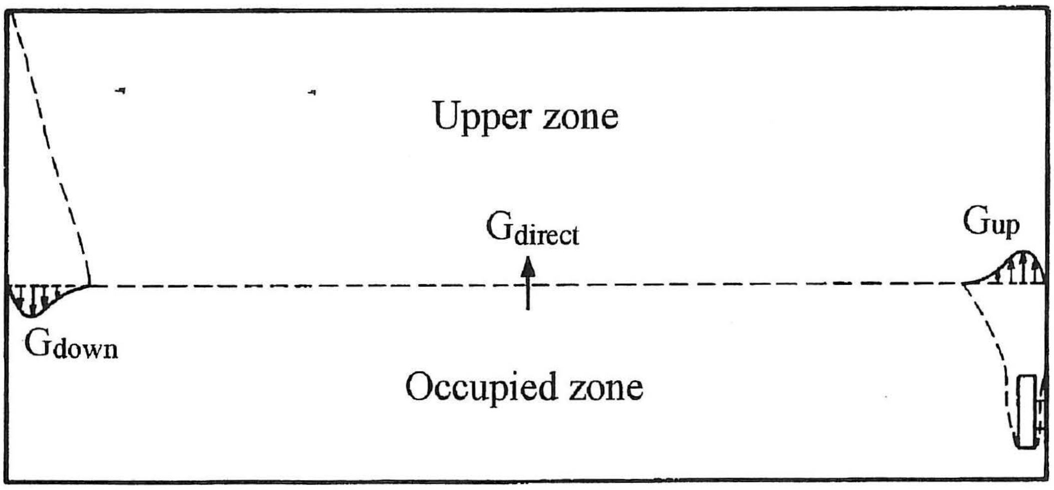

Figure 2. Model for air movements between upper and occupied zone.

The model for exchange of convective energy (figure 3) is built upon the main concept that the plume entrains air from the zone in which it is generated, and that the plume in the generating zone thermally separates the wall from the zone air. That means that there is not any mixing between the air and the plume in the generating zone and therefore the calculation of energy that leaves the generating zone can be based on equation 2 and the energy that passes through the fictive line that separates the zones can be based on equation 3. When the plume enters the receiving zone it is presumed that the plume will be totally mixed with the zone air in the receiving zone.

$$q_g = G_g \, c_p \, t_z \quad (2)$$

q_g energy that leaves the generating zone pr. metre surface (W/m)

G_g amount of air mass that passes through the zone dividing line (kg/s m)

c_p fluid specific heat at constant pressure (J/kg °C)

t_z air temperature in generating zone (°C)

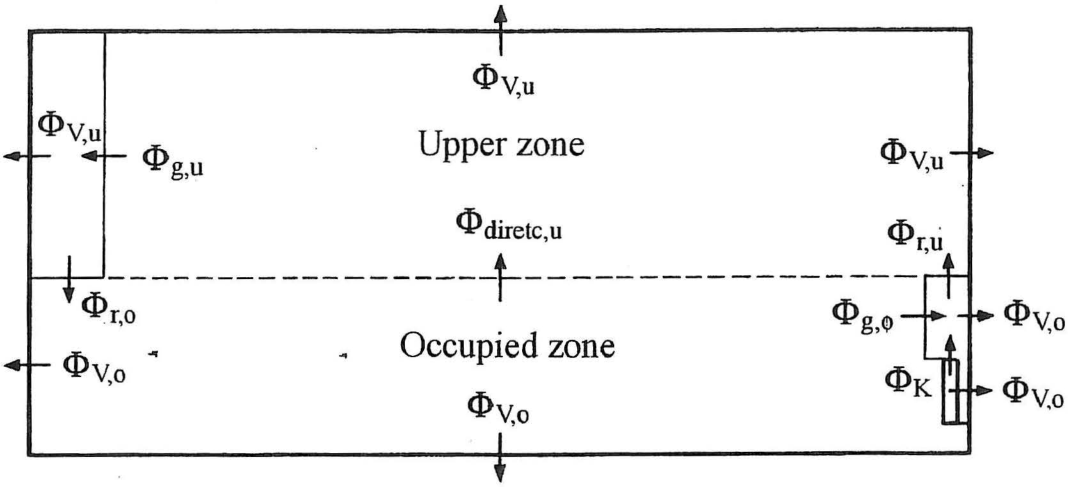

Figure 3. Model for convective energy exhange between zone and heat loss from the room.

$$q_r = q_g - q_V \tag{3}$$

q_r energy that entrains the receiving zone from the generating zone (W/m)

q_g energy that leaves the generating zone pr. metre surface (W/m)

q_V energy that leaves the air because of heat loss to the surface (W/m)

By this physically simple acceptance of the convective air movements it is possible to determine two mean temperatures for the room in every time step of the transient calculation, one temperature for the upper zone and one temperature for the occupied zone above the floor.

DIMENSIONLESS TEMPERATURE PROFILE

In [2] it is shown that it is possible to determine the real vertical temperature gradient by the way of a dimensionless temperature profile. Later experiments in test rooms of different room heights [3] have shown that under transient conditions it is necessary to use not one but two dimensionless temperature profiles (the drawn dimensionless temperature profiles are mean values based on 100 measured temperature profiles) one for periods with rising air temperatures (heating periods) and one for periods with falling air temperatures (cooling periods) (figure 4).

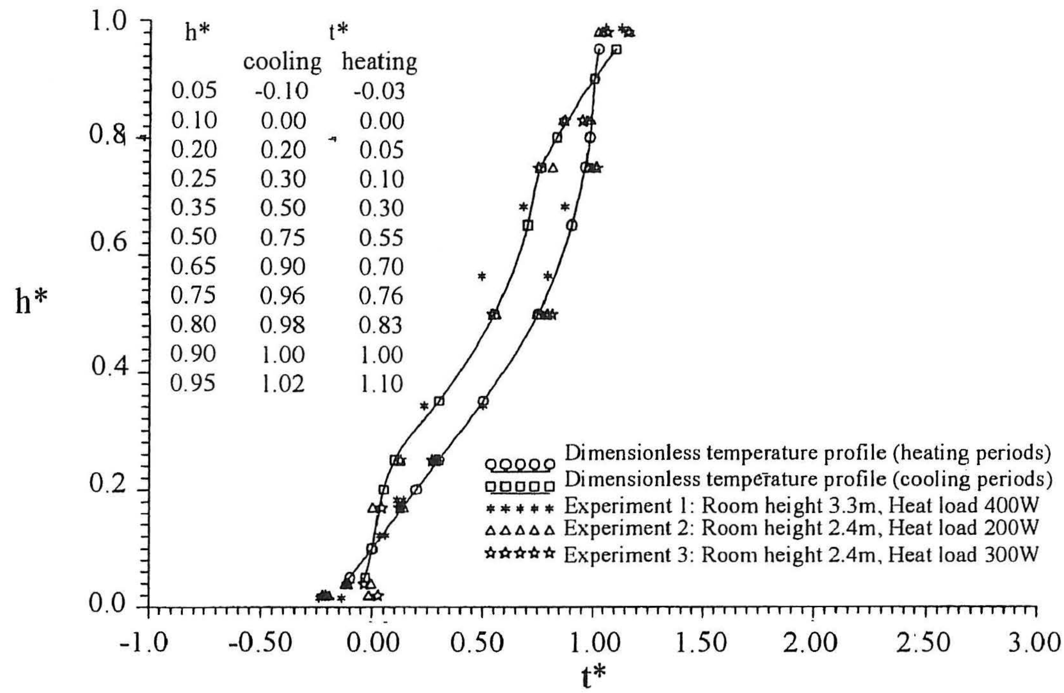

Figure 4. Dimensionless temperature profiles (assumption: t_o, t_u are measured in the dimensionless heights 0.1, 0.9).

The relationship between the real vertical temperature gradients and the dimensionless temperature profiles is according to equation 4 and 5.

$$t^* = \frac{t_h - t_o}{t_u - t_o} \qquad \textbf{(4)}$$

t^* dimensionless temperature

t_h temperature at the real temperature profile at the height h (°C)

t_o mean temperature for the upper zone (°C)

t_u mean temperature for the occupied zone (°C)

$$h^* = \frac{h}{H} \qquad (5)$$

h^* dimensionless room height

h the height in which the temperature t_h is measured (m)

H the total height of the room (m)

Determination of the height in which the mean temperature for a zone is placed

By means of the equations it is possible to calculate different mean air temperatures because the fictive dividing line between the upper zone and the occupied zone can be placed in an arbitrary height. Unfortunately, the mean air temperature for a zone cannot be related to the zone mean height. It is therefore necessary to find a method of determining in which height the mean air temperature is placed.

The only thing known about the real temperature profile and its shape is the connection to the dimensionless temperature profile. Therefore, the only way to determine the height in which the mean temperature in a zone is located is by way of the dimensionless temperature profiles. The conditions for using the dimensionless temperature profile to determine the height in which the mean temperature is located are that the height in which the mean temperature is located is the same on both the real temperature profiles as well as on the dimensionless temperature profile.

It can mathematically be shown that the above described conditions are valid (figure 5)

The temperature t and the dimensionless temperature t^* are both functions of the dimensionless height h/H (equations 6 and 7).

$$t = f_1(h/H) \qquad (6)$$

$$t^* = \frac{t - t_o}{t_u - t_o} = f_2(h/H) \tag{7}$$

If $\overline{t_o}$ is the mean air temperature for the occupied zone as shown in figure 5, then the following integrals must be equal (equation 8).

$$\int_0^a (\overline{t_o} - t)\, d\left(\frac{h}{H}\right) = \int_a^b (t - \overline{t_o})\, d\left(\frac{h}{H}\right) \tag{8}$$

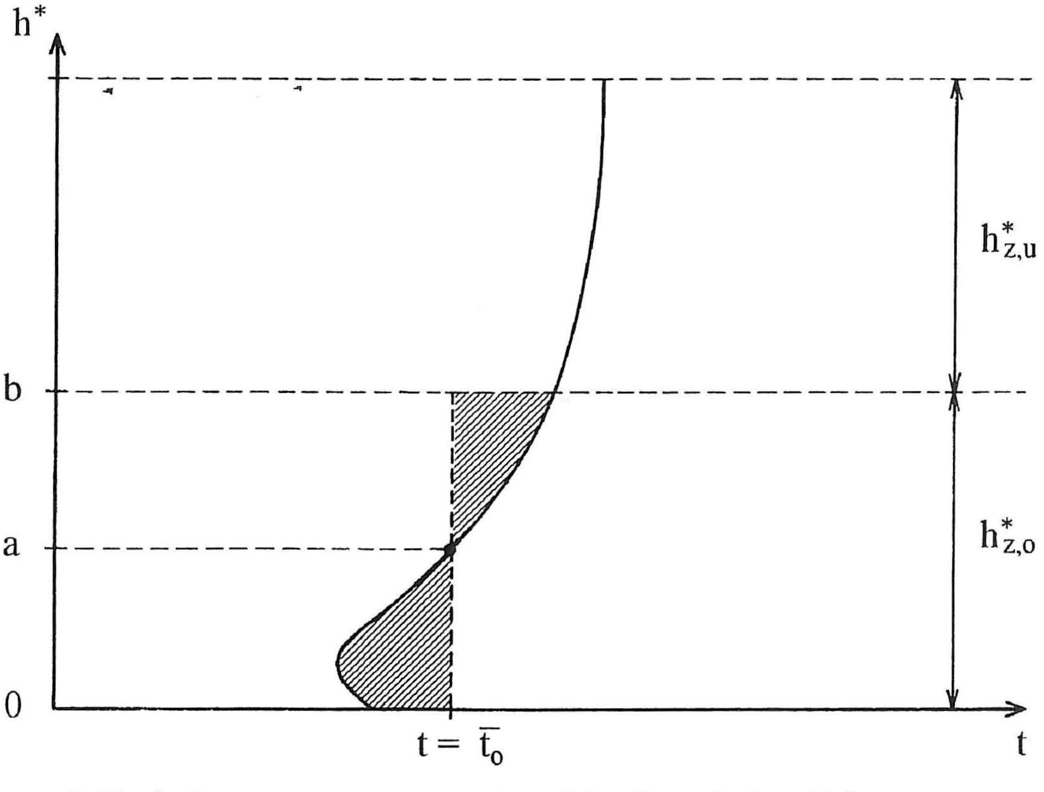

Figure 5. Vertical temperatures as a function of the dimensionless hight.

The mean temperature for the occupied zone $\overline{t_o}$ can therefore be determined by reducing equation 8. The result of this is given by equation 9.

$$\overline{t_o} = \frac{1}{b} \int_0^b t\, d\left(\frac{h}{H}\right) \tag{9}$$

It is possible to make the following changes of equation 8 by addition and subtraction of t_o , which gives equation 10.

$$\int_0^a (\overline{t_O} - t_o + t_o - t)\, d\left(\frac{h}{H}\right) = \int_a^b (t - t_o + t_o - \overline{t_O})\, d\left(\frac{h}{H}\right)$$
$$\Downarrow \tag{10}$$
$$\int_0^a [\, (\overline{t_O} - t_o) - (t - t_o)\,]\, d\left(\frac{h}{H}\right) = \int_a^b [\, (t - t_o) - (\overline{t_O} - t_o)\, d\left(\frac{h}{H}\right)$$

In the same way it is possible to multiply and divide equation 10 by $(t_u - t_o)$. This gives equation 11.

$$(t_u - t_o) \int_0^a \left[\frac{\overline{t_O} - t_o}{t_u - t_o} - \frac{t - t_o}{t_u - t_o} \right] d\left(\frac{h}{H}\right) =$$
$$(t_u - t_o) \int_a^b \left[\frac{t - t_o}{t_u - t_o} - \frac{\overline{t_O} - t_o}{t_u - t_o} \right] d\left(\frac{h}{H}\right) \tag{11}$$

The part $\dfrac{t - t_o}{t_u - t_o}$ is the definition of the dimensionless temperature therefore equation 11 can be reduced to equation 12 according to equation 4

$$\int_0^a (\overline{t_O^{\cdot}} - t^{\cdot})\, d\left(\frac{h}{H}\right) = \int_a^b (t^{\cdot} - \overline{t_O^{\cdot}})\, d\left(\frac{h}{H}\right) \tag{12}$$

In the same way as equation 8 equation 12 can be reduced to equation 13.

$$\overline{t_O^{\cdot}} = \frac{1}{b} \int_0^b t^{\cdot}\, d\left(\frac{h}{H}\right) \tag{13}$$

Equation 13 is the dimensionless mean temperature for the occupied zone. It is therefore shown that by means of the dimensionless temperature profiles it is possible to decide the height in which the mean temperature is located on the real vertical temperature profile,

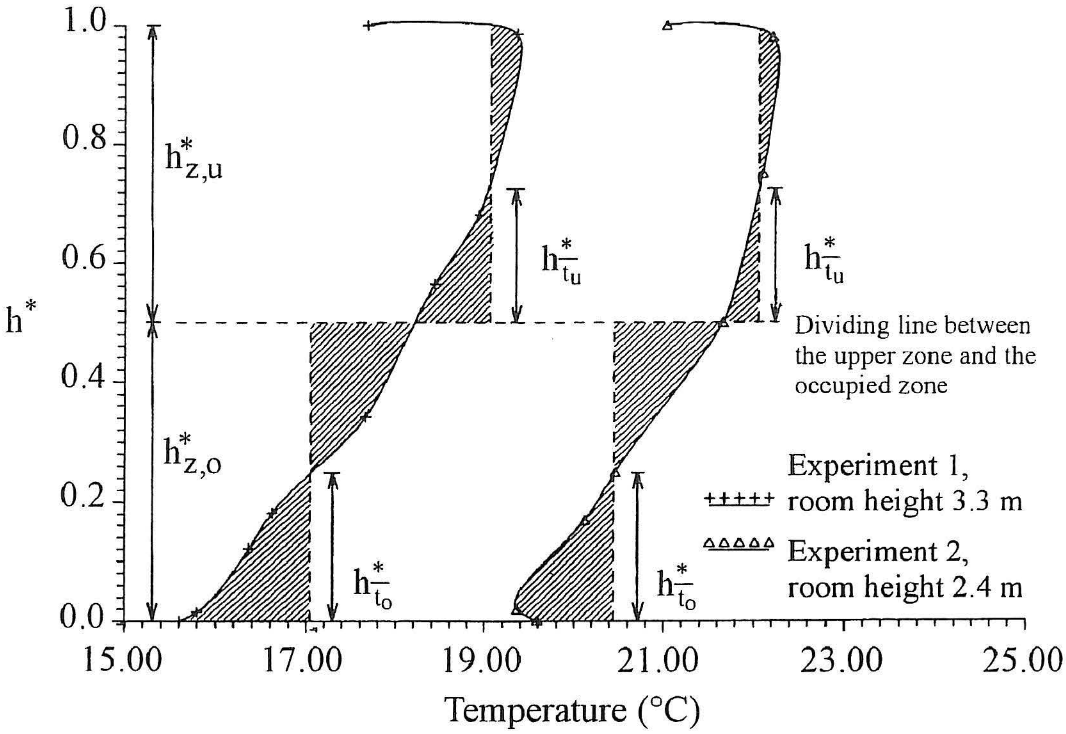

Figure 6. Graphic determination of the heights in which the mean temperatures are placed (two temperature curves measured under heating periods).

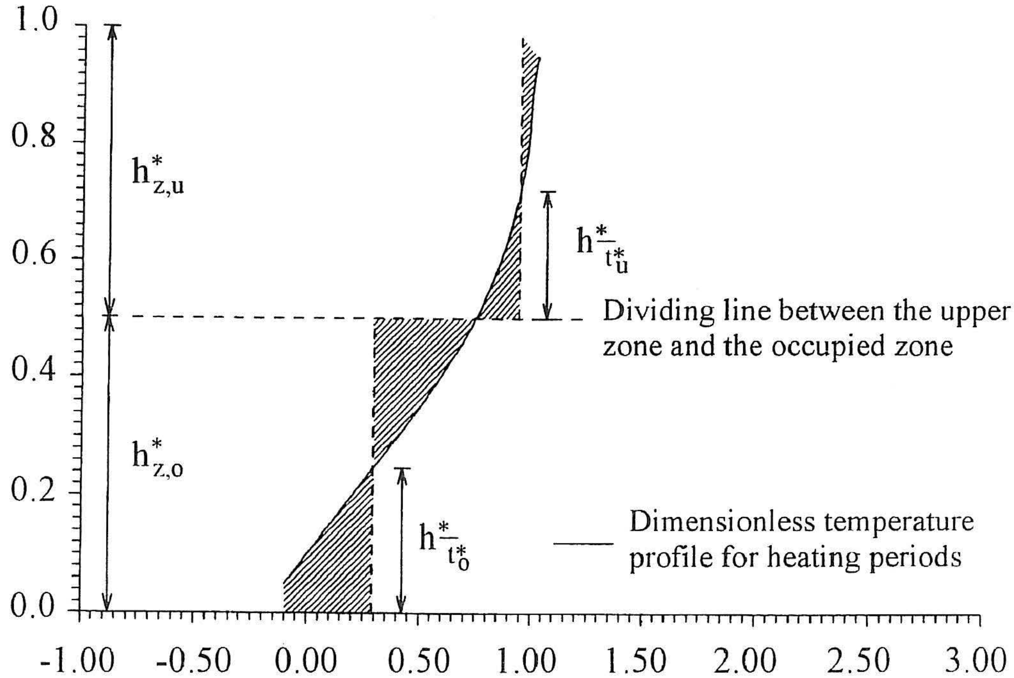

Figure 7. Graphic determination of the heights in which the mean temperatures are placed (dimensionless temperature profile for heating periods).

both for the whole room but also for an arbitrary vertical zone of the room. The relationship is graphical shown on two gradients measured in two different test rooms and on the dimensionless temperature profile for heating periods (figures 5 and 6). Here it can be seen that the heights in which the mean temperatures are located in the two different experiments are very near the heights in which the mean temperatures are placed on the dimensionless temperature profile, both for the upper zone and the occupied zone.

COMPARISON MEASUREMENTS AND CALCULATIONS

The here described experiment is carried out in a laboratory test room. The enclosure of the test room consists of a wooden frame structure. The room is divided into two parts by a heavy brick wall, in which there is a double-glazed window with an area of 1.08 m². Booth the heavy wall and the light frame constructions are insulated with 100 mm mineral wool. The smallest room can be cooled down to approximately -10 °C by two cooler coils. The other part of the enclosure has a floor dimension of 2450 x 2960 mm, the height of the room is in this experiment 2400 mm. This room is regarded as an indoor room with a heavy outside wall and three light inside walls and it is heated by an electric radiator. The test room is located in an open space in the laboratory.

The experiment is started from a thermal steady state condition where the temperature in the "outdoor" room is -10 °C. The indoor steady state situation is reached when the energy transport from the laboratory through the light walls is equal to the energy lost from the room through the brick wall. This steady state temperature for the warm room is approximately 13 °C. After stationary condition has been reached the room is supplied with 200 W (120 W of the energy supply is convective). Under the experiment the heat supply is cyclic 9 hours on and 15 hours off (figures 9 and 10). The total length of the experiment is 180 hours.

Zone	Experiment 2			
	Cooling		Heating	
	h^*	h	h^*	h
Occupied	0.30	0.70	0.25	0.60
Upper	0.80	1.90	0.67	1.61

Table 1. Heights in which the simulated mean temperatures are located.

Experiment 2		
	Heating	Cooling
t_O^*	0.31	0.17
t_U^*	0.93	0.82
h^*	t_s^*	
0.05	-0.66	-0.32
0.10	-0.50	-0.27
0.20	-0.18	-0.19
0.25	-0.02	-0.12
0.35	0.31	0.19
0.50	0.71	0.58
0.65	0.95	0.81
0.75	1.05	0.91
0.80	1.08	1.02
0.90	1.11	1.28
0.95	1.16	1.43

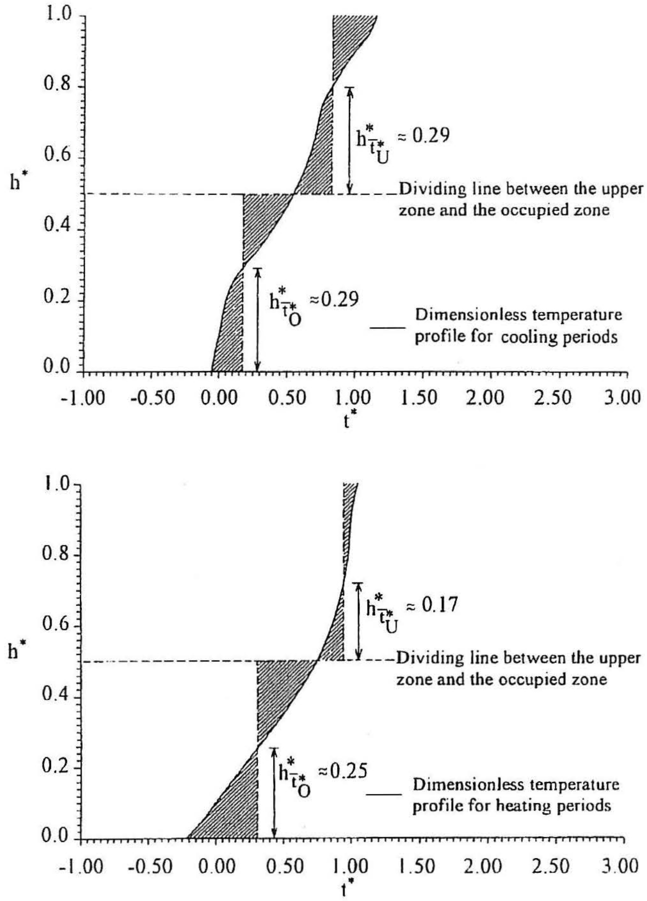

Figure 8. Graphic determination of the heights in which the mean temperatures are simulated for experiment 2.

Table 2. Dimensionless temperatures calculated for the simulated heights.

The heights (table 1) in which the mean temperatures are simulated, for the described experiment, are graphically determined in figure 8.

Transformation of the constants for the dimensionless gradients

The constants that describe the dimensionless temperature profiles for heating and cooling periods (table in figure 4), are based upon known temperatures in the dimensionless heights 0.1 and 0.9. Therefore, it is not directly possible to use these constants because the known (simulated) temperatures are placed at other heights. It is necessary to have a method to transform the constants in a way they can be used with known temperatures placed in arbitrary heights. By using equation 14 it is possible to calculate a temperature on the real vertical temperature profile in a room. The assumptions for using the equation and the constants t^* from figure 4 are that the known temperatures (measured or simulated) are placed in the dimensionless heights 0.1 and 0.9.

$$t_h = t^*(t_{0,9} - t_{0,1}) + t_{0,1} \tag{14}$$

The same dimensionless temperature profiles can of course be described by temperatures simulated in other heights ($t_{s,O}$, $t_{s,U}$) but the constants (t_s^*) will not be the same as before (equation 15).

$$t_h = t^*(t_{0,9} - t_{0,1}) + t_{0,1} = t_s^*(t_{s,U} - t_{s,O}) + t_{s,O} \tag{15}$$

By rewriting equation 15 the new constants (t_s^*) can be calculated by equation 16.

$$\begin{gathered}
t^*\left(\frac{t_{0,9} - t_{0,1}}{t_{0,9} - t_{0,1}} - \frac{t_{0,1} - t_{0,1}}{t_{0,9} - t_{0,1}}\right) + \frac{t_{0,1} - t_{0,1}}{t_{0,9} - t_{0,1}} = \\
t_s^*\left(\frac{t_{s,U} - t_{0,1}}{t_{0,9} - t_{0,1}} - \frac{t_{s,O} - t_{0,1}}{t_{0,9} - t_{0,1}}\right) + \frac{t_{s,O} - t_{0,1}}{t_{0,9} - t_{0,1}} \\
\Downarrow \\
t^*(t_{0,9}^* - t_{0,1}^*) + t_{0,1}^* = t_s^*(t_{s,U}^* - t_{s,O}^*) + t_{s,O}^* \\
\Downarrow \\
t_s^* = \frac{t^*(t_{0,9}^* - t_{0,1}^*) + t_{0,1}^* - t_{s,O}^*}{(t_{s,U}^* - t_{s,O}^*)}
\end{gathered} \tag{16}$$

But equation 16 can be reduced to equation 17 because $t_{0,9}^* = 1$ and $t_{0,1}^* = 0$. Therefore, it is possible to calculate a new set of constants by using equation 17 and the constants given in figure 4 and the dimensionless temperatures for the simulated heights.

$$t_s^* = \frac{t^* - t_{s,O}^*}{t_{s,U}^* - t_{s,O}^*} \tag{17}$$

The dimensionless temperatures $t_{s,O}^*$ and $t_{s,U}^*$ are given in figures 8 and 9 as the X-coordinate to the simulated heights (the Y-coordinate), because the dimensionless heights of the mean temperature, as previously described, are the dimensionless heights for both the real mean temperature and the dimensionless mean temperature.

In table 2 (page 12) the calculated constants used for simulation of the described experiment are shown.

The air temperatures that describe the vertical temperature profile in the room (experiment 2) can now be calculated by equation 18 where $t_{s,O}$ and $t_{s,U}$ are the temperatures simulated in the occupied zone, and the upper zone and t_s^* are the dimensionless temperatures given in table 2.

$$t_{s,h} = t_s^*(t_{s,U} - t_{s,O}) + t_{s,O} \qquad (18)$$

Measured and calculated air temperatures

In figure 9 the measured and the simulated air temperatures for both the occupied zone and the upper zone of the room are shown. Here it can be seen that the stationary start temperatures are approximately 1 °C too low. These lower start temperatures have an effect during the first 60 - 80 hours of the experiment. After that the differences between the measured and the simulated temperatures are stable and lower than 0.5 °C.

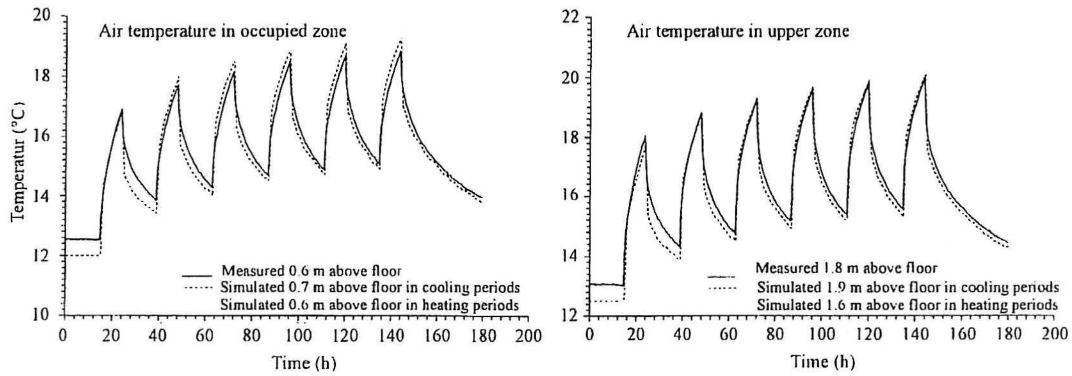

Figure 9. Measured and simulated air temperatures for experiment 2.

In figure 10 three measured and three calculated temperature profiles from the transient part of the experiment are compared.

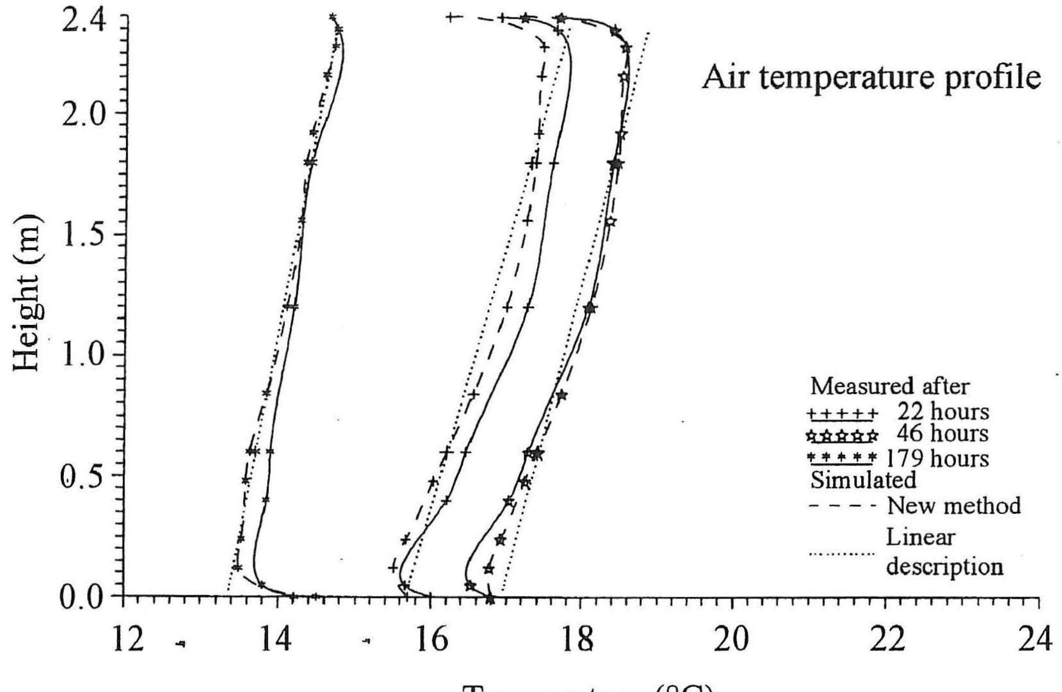

Figure 10. Air temperature profiles measured and simulated in experiment 2.

CONCLUSION

The present paper shows that with a convective air flow model it is possible to calculate the vertical main temperature profile in a heated room with only convective air movements. In the model two temperatures are calculated, the mean temperature in the occupied zone and the mean temperature in the upper zone of the room. To make a total gradient description it also demands a knowledge of two dimensionless temperature profiles for the room, one for periods with rising air temperatures and one for periods with falling air temperatures (heating/cooling periods). The dimensionless gradients are possibly independent of the room dimension because of the turbulence intensity in the boundary layer. The model is built upon the assumption that it is mainly the convective air movements along the room surfaces and over heat sources that generate the vertical temperature gradient, and therefore it is likely that the dimensionless temperature profiles can be used with success in general room loads calculations.

There is shown a way to find the height of a measured or a simulated mean air temperature based on the knowledge of the dimensionless air temperature profiles.

A good estimate of the vertical temperature gradient in the test room is obtained, but in general there is a tendency that calculated vertical temperature profiles are more steep than the real ones. This is because the model gives a little less temperature difference between the occupied zone and the upper zone than measured in the laboratory. It is concluded that this description is a better description of the vertical temperature profile than a linear decription, which is often used when only two air temperatures are known (figure 10).

The described hybrid model shows an improvement that could be included in commercial room load programmes.

ACKNOWLEDGEMENTS

This research work is a part of the IEA research programme Annex 26 within the field zonal models. The future work is, under transient conditions, to make a calculation of the maximal air velocity above the floor near cold vertical surfaces. The models will later be demonstrated at an Annex 26 case-study building.

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