MEASUREMENT OF THERMAL COMFORT PARAMETERS IN A SPACE WITH RADIANT COOLED CEILING

Vladimír Zmrhal, František Drkal
Czech Technical University in Prague, Faculty of Mechanical Engineering
Department of Environmental Engineering
Technicka 4, 166 07 Praha 6, Czech Republic
Vladimir.Zmrhal@fs.cvut.cz

ABSTRACT

The paper presents experimental measurement results of thermal comfort parameters in a space with radiant cooled ceiling. The experiments were carried out in a test chamber with radiant cooled ceiling, which represents standard office room. Displacement ventilation system was used for fresh air supplying. Internal heat gain models were used.

From the measured thermal comfort parameters the predicted mean vote *PMV* and draft risk *DR* were evaluated. Also vertical air temperature distribution in such a space was measured during the experiment.

INTRODUCTION

The operative temperature t_o is the evaluation criterion for thermal comfort in air-conditioned spaces – ČSN ISO Standard 7730. The operative temperature depends on air temperature t_a, mean radiant temperature - *MRT* and air velocity w_a. Operative temperature t_o is the uniform temperature of a radiantly black enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the non-uniform environment. The following formula is used (ČSN ISO Standard 7730)

$$t_o = At_a + (1 - A)MRT \qquad [°C] \qquad (1)$$

where *A* is a function of relative air velocity (from 0,5 to 1). When the air velocity is lower than 0.2 m/s, the operative temperature approximately equals the globe temperature t_g, and can be calculated as the average of air temperature and *MRT*

$$t_o = \frac{t_a + MRT}{2} \cong t_g \qquad [°C] \qquad (2)$$

The influence of the surface temperature on thermal comfort is significant; therefore if there is a cooled surface in the room it is possible to keep a higher indoor air temperature to achieve the same thermal comfort.

The fact, mentioned above, implicate in energy savings associated with cooling of the outside air (Zmrhal 2005).

Thermal comfort and thermal sensation can be predicted several ways. For the thermal sensation is commonly used seven points ASHRAE scale from –3 (cold) to +3 (hot). Fanger related the comfort data to physiological variables. Thermal comfort in conditioned spaces can be predicted by means of the predicted mean vote *PMV* or predicted percentage dissatisfied *PPD* (ISO Standard 7730, Fanger 1972). The *PMV* index predicts the mean response of large group of people according to the thermal sensation scale. The *PMV* index can be calculated from the conditions of thermal environment (t_a, *MRT*, w_a, φ), activity level *M* and various clothing insulation I_{cl}.

The comfort range of thermal comfort in air-conditioned spaces is normally considered as

$$-0,5 \le PMV \le +0,5 \qquad (3)$$

corresponding to

$$PPD < 10\% \qquad (4)$$

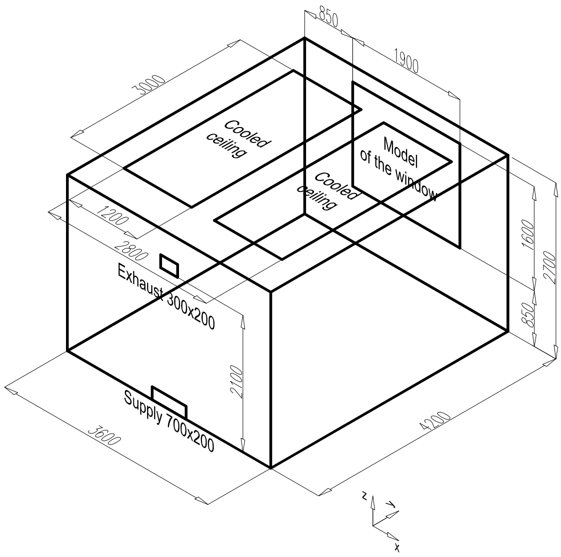

Figure 1 *Schematic drawing of the test chamber*

Table 1 Measurement conditions (CC – cooled ceiling)

Conditions	Heat gains					Ventilation		Supply air temp			Cooling ratio of CC		
	Window	Occupants	PC	Total heat load		Air change	Rate	Case 1	Case 2	Case 3	Case 1	Case 2	Case 3
	[W]	[W]	[W]	[W]	[W/m^2]	[h^{-1}]	[m^3/h]	[°C]	[°C]	[°C]	[%]	[%]	[%]
A	0	120	240	360	23.81	2.45	115	23.3	26.1	27.9	89	94	98
B	160	120	240	520	34.39	2.45	115	22.2	23.4	25.3	72	90	99
C	160	120	240	520	34.39	4.90	210	22.7	23.5	26.0	63	72	78
D	160	120	240	520	34.39	7.35	310	21.7	23.8	25.9	50	68	79

In a space with radiant cooled ceiling and displacement ventilation the low air velocity w_a (≤ 0,2 m/s) in occupant zone can be predicted (Skistad 1994).

The percentage of people dissatisfied because of draft intensity predicts index *DR*. The comfort range because of draft risk is normally considered as

$$DR < 15\% \qquad (5)$$

EXPERIMENT

The presented experiments were carried out in an existing test chamber with a floor space of 4.2 by 3.6 m and a room height of 2.7 m (*Figure 1*). The radiant cooled ceiling of 7.2 m^2 area is located symmetrically in the chamber. Cooled ceiling covers the 48% of total ceiling area. The test chamber is ventilated by displacement ventilation (AHU). The exhaust and supply air openings are installed symmetrically in the front wall (*Figure 1*).

Supply air opening with dimensions of 0.7 by 0.2 m is located near the floor (*Figure 1*). The exhaust air opening with dimensions of 0.3 by 0.2 m is located 2.1 m over the floor. Exhaust air opening is joined with laboratory space. The test chamber is placed in a larger, air-conditioned enclosure providing the chamber with required ambient conditions. The surrounding air was conditioned to provide minimal heat flux between both spaces, i.e. the temperature of the surrounding air was adjusted identically to the air temperature in the test chamber. Moreover, all chamber surfaces are very well heat-insulated (30 mm polystyrene) to minimize heat loss.

The models of internal heat gains were used (2 models of occupants, 2 models of PC and the model of the window). Window model represents shaded window without direct solar heat gains. The model of the window was made from three electric heating sheets of 1.8m length and 0.6m width. The galvanised steel sheet with thickness of 1mm covered the black heating sheets. For presented experiments the surface temperature of the window model was set to 40 °C, corresponding to electricity input of 160 W.

The simplified models of occupants were made to model the sensible heat gain. Occupant model contains from steel duct on the stand and the bulb (60 W) inside the duct (*Figure 2*). The free convection stream was able to flow along the duct. Real PC case with two bulbs (2x60 W) was used for PC modelling. Two occupant models and two of PC models were used. Experimental set-up in the test chamber is illustrated in *Figure 2*. Measurement conditions are presented in *Table 1*.

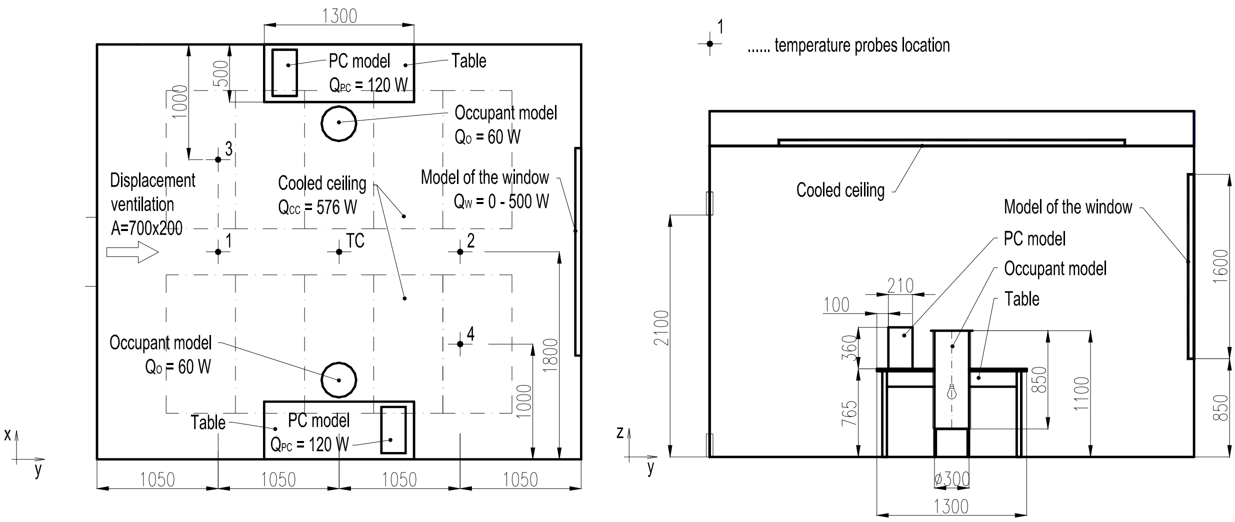

Figure 2 *Experimental set-up in the test chamber*

METHODOLOGY

Measurement of vertical temperature distribution was carried out along four vertical axes in the test chamber (*Figure 2* - location 1,2,3 and 4). The special multiple temperature probes were made. There were used 4x10 Pt100 sensors (class A) with minimal proportions (1,6 x 3,2 x 1,0 mm). Each of multiple probe contains 10 sensors for air temperature measurement. For data collection the data logger Ahlborn ALMEMO 5590-3 was used.

The "*Indoor Flow System*" (Dantec) and globe thermometer were installed in the middle of the room (*Figure 2* – TC location). The measurement of thermal comfort parameters measurement (air temperature t_a, globe temperature t_g, air velocity w_a and turbulence intensity *Tu*) was performed in four room elevations - 0.1, 0.6, 1.1m and 1.7m above the floor during the steady state. Thermal conditions in test chamber were monitored.

Measurements of capacity parameters were carried out for experiment evaluation. The supply air temperature was measured in supply air opening (*Table 1*). Temperature of indoor air was calculated from vertical temperature profiles (*Figure 6*) and also exhaust air temperature was measured. Eight temperature probes were located around the test chamber for outside air temperature measurement. Two water temperature probes Pt100 (supply and return pipe) and flow meter were installed in pipe system to measure heat flux supplying into the room from the cooled ceiling.

RESULTS

Thermal Comfort

The predicted mean vote *PMV* was evaluated on the basis of thermal comfort parameters measurement. The operative temperature t_o is the evaluation criterion for thermal comfort in air-conditioned spaces (Czech legislative act no. 523/2002 Sb., ČSN ISO Standard 7730). If the air velocity w_a is lower than 0.2 m/s, the effect of air velocity is insignificant. The main goal of the experiments was to find out if such a neglect of air velocity causes variability of thermal sensation *PMV*.

The *PMV* index was calculated (Fanger 1972) for measured parameters (t_a, *MRT*, w_a), activity level of 1 met, clothing insulation of 0.5 clo, and relative air humidity of 50 %. Operative temperature was calculated according to equation (1) also on the basis of measured thermal environment parameters (t_a, *MRT*, w_a).

Figure 3 presents the relationship between the predicted mean vote index *PMV* and operative temperature t_o for all presented cases. Linear theoretical dependences (valid for assigned air velocity) are calculated from *PMV* equation (ISO Standard 7730, Fanger 1972). The points in graph are drawn for one measurement state and represents *PMV* index calculated from measured parameters.

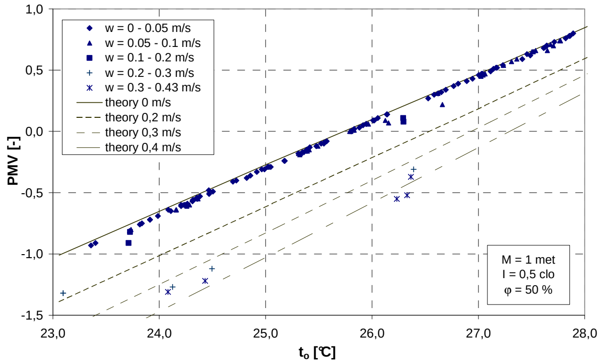

Figure 3 *Predicted mean vote PMV = f(t_o)*

For the same operative temperature t_o the different *PMV* value will be achieved how it is shown in *Figure 3*. For example if operative temperature t_o is 25°C and air velocity w_a = 0 m/s, *PMV* index will be equal to -0.28 (neutral); for w_a = 0.2 m/s, *PMV* index will be equal to -0.61 (slightly cold). In a space with radiant cooled ceiling the air velocity will be very low and mostly near to 0 m/s. The experiments confirm, that neglect of air velocity (w_a = 0 - 0.2 m/s) in operative temperature calculation (1) doesn't cause variability of thermal sensation *PMV*. If the air velocity will be higher $w_a \geq 0.2$ m/s (for the same operative temperature t_o), thermal sensation will be slightly cold.

Draft Risk

During experiments, the air temperature t_a, air velocity w_a and turbulence intensity *Tu* were measured in elevations 0.1, 0.6, 1.1 and 1.7 m above the floor. The draft risk *DR* (ISO Standard 7730) can be calculated from these quantities, which predicts the percentage of people dissatisfied because of draft intensity. *Figure 4* and *5* shows the experimental result of the draft risk *DR* as a function of air velocity w_a for all cases.

Results of thermal comfort parameters measurement show no risk of draft (*DR* < 5%) in a space with radiant cooled ceiling combined with displacement ventilation for elevations 0.6, 1.1m and 1.7m above the floor (*Figure 4*). Local discomfort due to draft risk can cause by airflow increasing in ankle zone (*h* = 0.1 m above the floor) as it is shown in *Figure 5*).

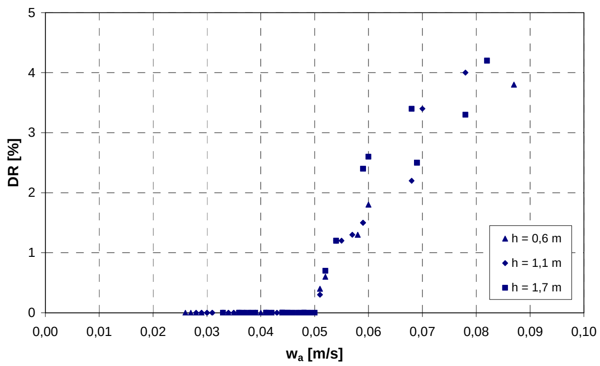

Figure 4 *Draft risk DR = f(w_a) (0.6; 1.1 and 1.7 m over the floor)*

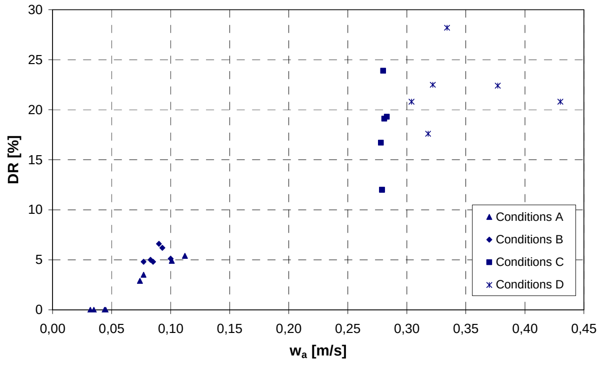

Figure 5 *Draft risk DR = f(w_a) (0.1 m over the floor)*

Temperature distribution

Vertical temperature profiles shown in *Figure 6* represent air temperature distribution in a space with radiant cooled ceiling. Air temperatures were evaluated during measurement steady state. Information about cases 1,2,3 and conditions A,B,C,D are present in *Table 1*, Probes 1,2,3 and 4 means location in *Figure 2*. For conditions A (cases 1 – 3) the straight temperature profiles in occupied zone are visible because minimal heat load (360 W) and the isothermal supply air were used.

Under the ceiling, the peaks caused by convection flow along heat gains are visible in all cases. The peaks in location 3 and 4 for conditions A (*Figure 6*) are caused by internal heat gains (PC model and occupant model). The effect of convection airflow along the model of window is visible for conditions B, C and D (*Figure 6* - location 2).

The effect of displacement ventilation is present in all cases near the supply air opening (*Figure 6* - especially location 1). The peak of temperature profiles caused by the lower temperature of supply air is visible in the height of 0.1 m above the floor. If the cooling capacity of the ceiling is sufficient and the airflow is isothermal, the peak of temperature profile is negligible (conditions A, case 3).

Conditions B represent measurement with maximal heat load (520 W) and minimal supply of air (115 m^3/h). The portion of the total heat load is removed by supplying non-isothermal airflow, which cause deformation of vertical temperature profiles in the occupied zone. Conditions C and D bring information about variations of temperature profiles in a room with cooled ceiling caused by increasing of supply airflow. The effect of increasing airflow on temperature profiles is positive. As it is shown in *Figure 6* (conditions D3) the bigger airflow and higher air temperature (higher heat transfer along the cooled ceiling) makes the temperature profiles line-up.

Based upon thermal comfort requirements (ISO Standard 7730) for sitting person in the summer, the air temperature difference between head (1.1 m above the floor) and ankle (0.1 m above the floor) must be less than 3 K. Temperature profiles document the executed requirement in all cases (Figure 3). For standing person, the difference between head (1.7 m above the floor) and ankle is maximally 2.9 K.

Radiation effect discussion

Although, during the measurement preparation, it was respected eventual radiation effect on temperature sensors and it's minimisation (optimal dimensions of temperature sensors), it's necessary to discuss about this phenomena.

For each of the sensors the radiation effect will be different along vertical temperature distribution in a space with radiant cooled ceiling.

Odds are, that sensor nearly under the ceiling will be more influenced by radiation, than sensors close by the floor. During experiments it was practically impossible to protect all probes against radiation.

Before the experiments the control measurement was carried out, which contained comparison between two vertical temperature profiles located at the same position under the cooled ceiling. For this purpose the sensors were shaded. The first measurement was realised without shading, secondly the sensors at one multiply probe were covered with shading.

By comparison of measured temperatures without and with shading was found, that values of measured temperatures are approximately the same. The error variable is lower than tolerance of using sensors (Pt100 class A: ± 0.1°C). That means measurement temperatures $t_a´$ express air temperature t_a very well.

CONCLUSIONS

Cooling capacity of the cooled ceilings is limited. If the higher cooling capacity is needed, it is possible to cool also by supply air. Ideal air temperature distribution in a space with radiant cooled ceiling and displacement ventilation will be achieved for isothermal supply of air. When the supply air temperature will be lower (higher cooling capacity of the air) the vertical temperature profiles in occupied zone are deformed. On the contrary the higher airflow, the straight vertical profiles.

In a space with radiant cooled ceiling there is no risk of thermal discomfort due to vertical temperature difference between head and ankles, how it follows from vertical temperature profiles evaluation (*Figure 6*). Air temperature difference between the head and ankle for a sitting person is less than 3 K (in presented cases), that is a basic thermal comfort requirement in the summer. The condition is valid also in cases with non-isothermal supply of air, when deformation of vertical profiles is present. The experiments were carried out for temperature difference of displacement ventilation ≤ 3K.

Thermal comfort evaluation presents relationship between the predicted mean vote index *PMV* and operative temperature t_o. In a space with radiant cooled ceiling the air velocity (in occupied zone) is very low. The experiments confirm, the that neglect of air velocity (w_a = 0 - 0.2 m/s) in operative temperature calculation (1) doesn't cause variability of thermal sensation *PMV*.

According to the experiments results, there is no risk of draft in a space with radiant cooled ceiling and displacement ventilation system. In elevations 0.6, 1.1 a 1.7 m over the floor the air velocity is very low for all presented cases. The local discomfort can develop in the ankle zone only (0.1 m above the floor) due to higher air velocity (over 0.4 m/s) in supply air opening.

ACKNOWLEDGEMENTS

The paper is integrated in the framework of CTU Research Aim MSM 6840770011.

REFERENCES

ASHRAE Handbook 2001 Fundamentals, 2001, American Society of Heating, Refrigerating and Air-conditioning Engineers, Atlanta. ISBN - 1-883413-87-7

FANGER, P.O. *Thermal comfort – Analysis and applications in environmental engineering*, Kingsport Press, Inc.,1972.

ČSN ISO Standard 7730, *Moderate thermal environments – Determination of the PMV and PPD indices and specification of the conditions for thermal comfort*. International Standard Organization.

KULPMANN, R.W. Thermal Comfort and Air Quality in Rooms with Cooled Ceilings – Results of Scientific Investigations. In *ASHRAE Transactions*, 1993, vol. 99(2) pp. 488 – 502.

SKISTAD, H. *Displacement ventilation in non-industrial premises*, 1994, REHVA guidebook .

ZMRHAL, V., DRKAL, F. Thermal Comfort Evaluation in a Space with Cooled Ceiling (in Czech). In: *Proc. of 16th National Conference Air-conditioning and ventilation 2004*, Prague 2004.

ZMRHAL V. *Thermal Comfort and Energy Balance of Cooled Ceiling System*. 2005, Doctoral thesis, CTU in Prague.

ZWEIFEL, G. Simulation of displacement ventilation and radiation cooling with DOE2. In *ASHRAE Transactions*, 1993, vol. 99 (2), pp. 548-555.

NOMENCLATURE

DR	draft risk	%
h	measurement elevation	m
I_{cl}	clothing insulation	clo, m^{2}K/W
M	metabolic heat generation	met, W/m^{2}K
MRT	mean radiant temperature	°C
PMV	predicted mean vote	%
t_a	air temperature	°C
t_a‘	measured temperature	°C
t_o	operative temperature	°C
t_g	globe temperature	°C
x,y,z	axes	-
w_a	velocity of air	m.s^{-1}
φ	relative humudity	%

APPENDIX

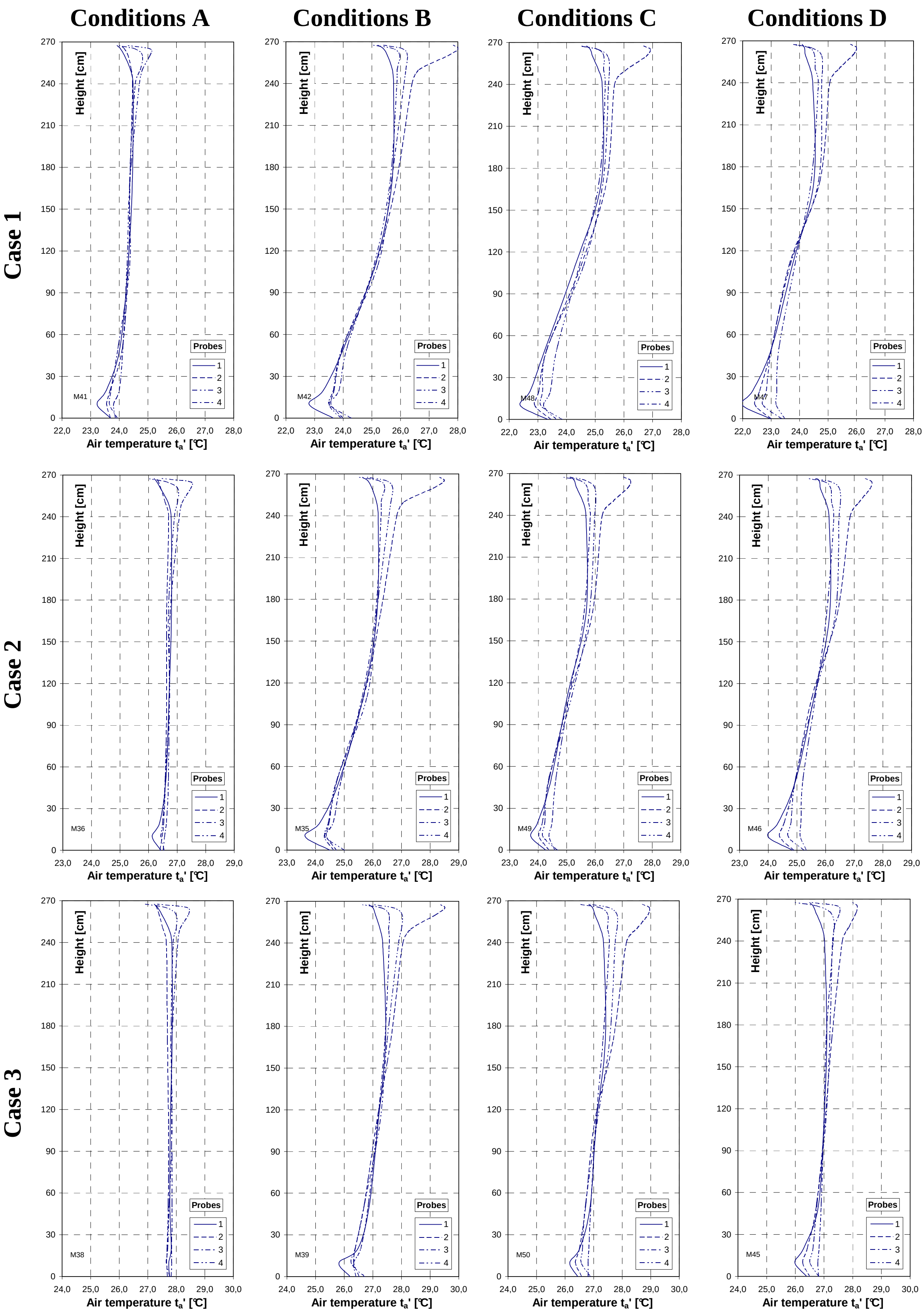

Figure 6 *Temperature distribution in a space with radiant cooled ceiling*