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Effect of Combined Heat and Moisture Transfer on the Predicted Indoor Thermal Environment

Key Words
Radiation heat transfer
Moisture transfer
Condensation
Thermal comfort
Computational fluid dynamics

Abstract
Both building performance and occupants' thermal comfort are influenced by heat conduction, convection and radiation as well as condensation/evaporation. This paper presents an analysis of thermal comfort in buildings subjected to simultaneous heat and moisture transfer using numerical techniques. A model based on computational fluid dynamics has been developed for predicting the indoor thermal environment. Assessment is made of the effects of radiation heat transfer and moisture condensation on the accuracy of predicted indoor thermal comfort.

Introduction

Accurate modelling of combined air, heat and moisture transfer in buildings is important in predicting heating/cooling loads, thermal comfort and air quality as well as degradation and deterioration of building components due to moisture problems and mould growth [1]. The thermal environment in buildings involves complex interactions between air, heat and moisture. For example, temperature and moisture gradients in a naturally ventilated room are important driving forces for the air movement; temperature and vapour pressure differences between different zones will cause interzonal exchanges of air, heat and water vapour.

During the last few decades, researchers have developed computer models for the heat and moisture transfer in and around buildings. Most of the existing building simulation models apply to certain specific aspects of building physics such as indoor climate in association with heating and cooling loads, solar heat gains through windows, air infiltration phenomena and air flow patterns. Heat transfer by conduction in building structures and convection between a room surface and air has been extensively studied. Owing to such studies standard procedures for the calculation of these heat transfer modes have been established [2, 3].

Radiation heat transfer is an important aspect in load calculation and thermal comfort in buildings. Athienitis and Haghighat [4] presented an experimental and numerical study of the effects of solar radiation on the indoor environment in terms of globe temperature. It was shown that diffuse solar radiation could have a significant effect on the thermal comfort of occupants in buildings with large windows. Stefanizzi et al. [5] compared different algorithms for radiation heat transfer and concluded that a more accurate method was required in cases of low surface emissivity or large temperature differences between surfaces. It was also shown that failure to model internal long-wave radiation to and from windows could lead to substantial errors in predicted overall energy balance. Chapman and Zhang [6] developed a three-dimensional mathematical model to compute the radiant heat exchange between room surfaces separated by a transparent and/or opaque medium using the discrete ordinates radiation model. By coupling the radiation model with a convective and conductive heat transfer model and a thermal

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comfort model, it was envisaged that comfort levels throughout the room could be easily and efficiently mapped for a given radiant heater location.

Radiation heat exchange models have also been incorporated with computational fluid dynamics (CFD) in order to provide more realistic modelling of air flow in buildings. Chen and van der Kooi [7] presented a methodology for the computation of airflow and space cooling load of a room with a displacement ventilation system. Air flow and temperature distribution in the room were predicted using the CFD method. A radiation model was used to calculate solar radiation through windows and heat exchange between room surfaces. Ozeki et al. [8] developed a numerical simulation system for temperature and flow distributions taking account of various radiative heat transfer mechanisms including the direct solar radiation, the diffused solar radiation and the diffused reflection of the solar radiation. Such numerical methods were also used for predicting steady state distributions of air flow and thermal comfort in rooms with floor panel-heating [9], with natural ventilation [10] and displacement ventilation [11].

Indoor air humidity influences the performance of a building as well as thermal comfort and air quality. One of the most serious problems relating to moisture transport in a building is condensation. Condensation of water vapour depends upon temperature and humidity levels. Therefore, studies of transport of moisture in building structures and in spaces have to be carried out in combination with heat transfer processes. Examples of condensation studies include analysis of the thermal performance of curtain walls and moisture condensation on glass and mullion surfaces [12]; interaction between moisture condensation and surface temperature [13] and improvement of numerical predictions of window condensation potential by including interglazing convection and intraglazing radiation heat transfer [14]. Moisture transfer in hygroscopic building materials involves absorption and desorption. The hygroscopic behaviour of building materials was taken into account by a number of investigators in modelling moisture transfer in structures [15], in evaluating room air-conditioning load [16] and in determining air humidity in an enclosure [17] or in a multizone space [18] and water vapour flow through a ventilated or non-ventilated room [19].

Most of the analyses of moisture transport in buildings involve only heat and mass balances without considering the spatial variation of humidity. The transport of moisture can be predicted more accurately using a CFD technique that takes account of various heat transfer processes (radiation, convection and conduction). It also enables comprehensive evaluation of thermal comfort in buildings. In 1994 the author made use of such a technique to study thermal comfort in ventilated rooms [20]. The study revealed the importance of accurate predictions of air movement, moisture transfer and radiation heat transfer for assessing the indoor thermal environment. Based on the method developed in the previous investigation, this paper presents a CFD analysis of air, heat and moisture behaviour in a typical office space. The principal purpose of this analysis is to examine the effects of radiation heat transfer and surface condensation on the predicted indoor thermal comfort.

Mathematical Model

A complete mathematical model for predicting the indoor thermal environment requires the CFD model for general air flow and models for thermal comfort.

CFD Model

The CFD simulation is based on a two-dimensional TEAM computer code [21]. The original 2-D code is here extended to the 3-D conditions in order to predict air, heat and moisture movement in enclosures. Jacobsen [22] evaluated the TEAM code by comparing the numerical prediction with experimental measurement. It was found that the code produced satisfactory results with respect to the general room air flow patterns and temperature distribution when thermal radiation between room surfaces was taken into account.

In the present CFD model turbulent air, heat and moisture movement is represented by transport equations for continuity, momentum, turbulence, enthalpy and concentration. For an incompressible flow the time-averaged steady state equations can be written as

$$\underbrace{\frac{\partial}{\partial x_i}(\rho U_i \phi)}_{\text{convection}} + \underbrace{\frac{\partial}{\partial x_i}\left(\Gamma_\phi \frac{\partial \phi}{\partial x_i}\right)}_{\text{diffusion}} = \underbrace{S_\phi}_{\text{source}} \quad (1)$$

where ϕ denotes the dependent variable which may stand for mean velocity component U_i in x_i direction, turbulent kinetic energy and its dissipation rate, mean specific enthalpy and mean concentration of moisture; ρ is the air density; Γ_ϕ is the diffusion coefficient for variable ϕ; S_ϕ represents the source term for variable ϕ (table 1).

The boundary conditions for solving the model equations are summarised in table 2 and the details are given in reference 20.

Table 1. Source terms of flow equations

Equation	ϕ	Γ_ϕ	S_ϕ
Continuity	1	0	0
Momentum	U_j	μ_e	$-\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_i}\left(\mu_e \frac{\partial U_i}{\partial x_j}\right) - \frac{2}{3}\frac{\partial(\rho k\, \delta_{ij})}{\partial x_j} + g_j(\rho_r - \rho)$
Kinetic energy	k	Γ_k	$G_s + G_B - C_D\rho\varepsilon$
Dissipation rate	ε	Γ_ε	$C_1(G_s + C_3G_B)\varepsilon/k - C_2\rho\varepsilon^2/k$
Enthalpy	H	Γ_e	q
Concentration	C	Γ_c	$C_s\rho$

Definitions

$$T = \frac{H - 2501000C}{1000 + 1805C} \qquad \rho = \frac{p_aM_a + p_vM_v}{R(T + 273.15)} = \frac{p_{at}M_a}{R(T + 273.15)}\frac{1}{1 + \left(\frac{M_a}{M_v} - 1\right)C}$$

$$\mu_e = \mu + \mu_t \qquad \mu_t = C_\mu\rho k^2/\varepsilon$$

$$\Gamma_k = \mu + \mu_t/\sigma_k \qquad \Gamma_\varepsilon = \mu + \mu_t/\sigma_\varepsilon$$

$$\Gamma_e = \mu/\sigma + \mu_t/\sigma_t \qquad \Gamma_c = \mu/\sigma_s + \mu_t/\sigma_c$$

$$G_s = \mu_t \frac{\partial U_i}{\partial x_j}\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right) \qquad G_B = \frac{g_i}{\rho}\frac{\mu_t}{\sigma_t}\frac{\partial \rho}{\partial x_i}$$

$$C_3 = \tanh(V_g/V_h)$$

Empirical constants in the model equations

C_μ	C_D	C_1	C_2	σ_t	σ_c	σ_k	σ_e
0.09	1.0	1.44	1.92	0.9	1.0	1.0	1.22

Table 2. Boundary conditions for flow equations

Variable	Supply	Extract	Solid boundary
U_i	specified V_s	continuity	wall function
k	$k = 0.0225V_s^2$	$\partial k/\partial n = 0$	modified k*
ε	$\varepsilon = \frac{C_\mu k^{1.5}}{0.07L}$	$\partial\varepsilon/\partial n = 0$	$\varepsilon = C_\mu^{3/4}k^{3/2}/\kappa n$
H	specified	$\partial H/\partial n = 0$	wall function
C	specified	$\partial C/\partial n = 0$	wall function

* k is obtained by the direct solution of the k-equation with modifications given in reference 21.

The flow equations are solved for the 3-D cartesian system using the SIMPLE algorithm [23]. Equation 1 is first discretised by means of a control-volume based finite difference technique on a staggered grid with centred grid nodes. The convection and diffusion terms in equation 1 are then integrated using the Power-Law [23] or QUICK [24] difference scheme. In this study the QUICK scheme is applied to momentum equations whereas the Power-Law scheme is applied to other transport equations.

Thermal Comfort Indices

Thermal comfort is evaluated in terms of predicted mean vote and predicted percentage of dissatisfied derived by Fanger [25]. The predicted percentage of dissatisfied, PPD (%), is given by [26]

$$PPD = 100 - 95e^{-(0.03353PMV^4 + 0.2179PMV^2)} \quad (2)$$

where PMV is the predicted mean vote; it is a function of resultant air velocity (V), mean air temperature (T), mean radiant temperature (T_{mr}), water vapour pressure of air (P_v), clothing thermal resistance and occupant's metabolic rate. Details of its expression are given by Fanger [25].

The air velocity, air temperature and vapour pressure are predicted using the flow equations. The mean radiant temperature and associated radiant temperature asymmetry are calculated from plane radiant temperatures using a radiation heat exchange model [20]. The radiation model is also used to calculate the temperatures of inner room surfaces based on the heat balance equations for conduction, convection and radiation as well as the latent heat release in the case of condensation. The room surface temperatures are then used as boundary values in the CFD prediction.

In addition, indices such as operative temperature and resultant temperature are often used in practice for simplified description of the thermal environment.

The operative temperature, T_o (°C), is the average of the mean radiant and air temperatures weighted by their respective heat transfer coefficients and is given by [2]

$$T_o = \frac{h_r T_{mr} + h_c T}{h_r + h_c} \quad (3)$$

where h_r and h_c are the radiative and convective heat transfer coefficients respectively (W/m²K) and are evaluated at the outer surface of the clothed body.

The resultant temperature, T_{res} (°C), is recommended for use in the UK [3] and calculated using

$$T_{res} = \frac{T_{mr} + T\sqrt{10V}}{1 + \sqrt{10V}} \quad (4)$$

Simulation Conditions

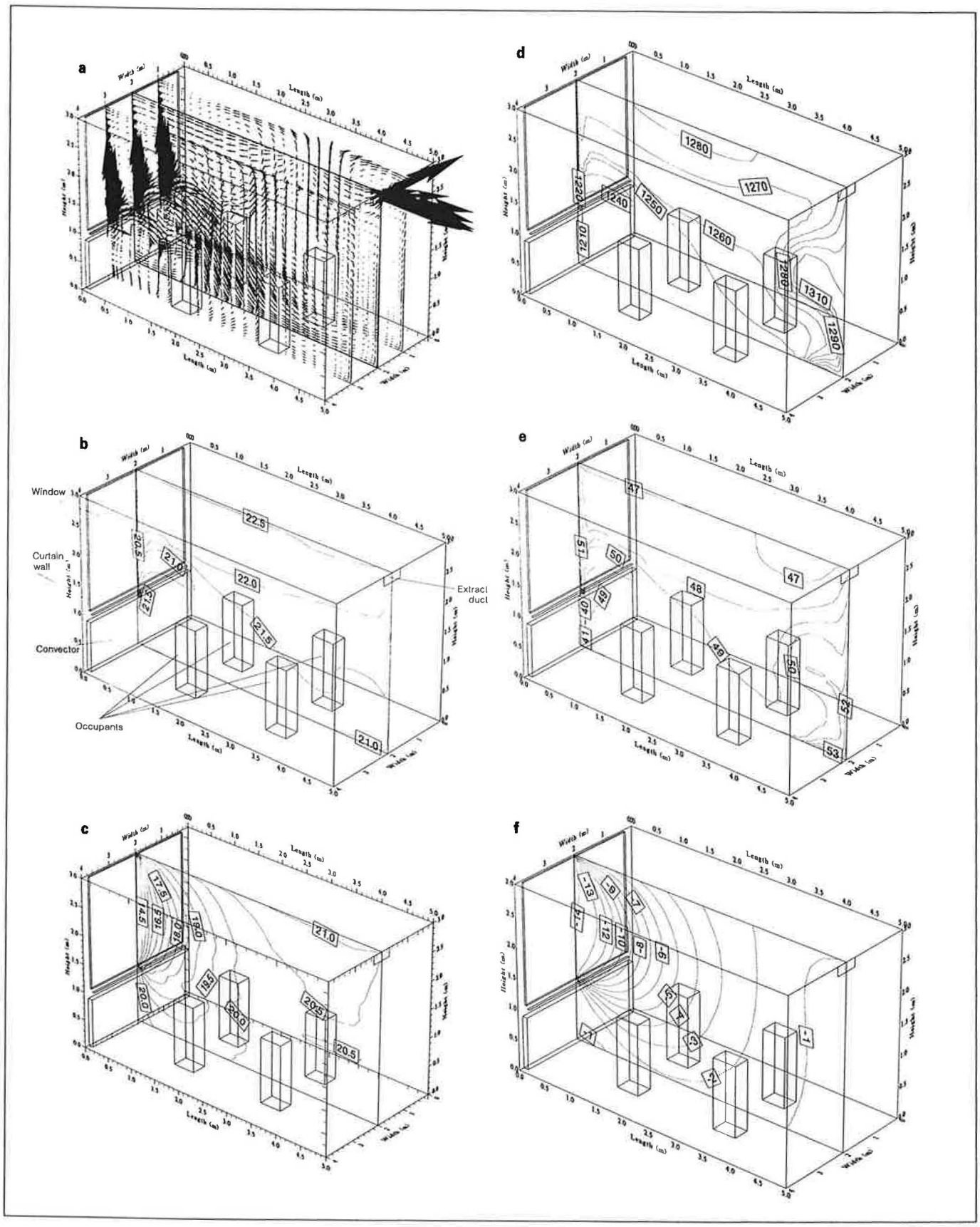
The numerical method is used to predict the thermal environment of a mechanically ventilated office in winter conditions. The office module under consideration was selected as an example calculation by CIBSE [3]. The office is 5 m long, 4 m wide and 3 m high (fig. 1b). It is located on an intermediate floor. The external wall (curtain wall) is constructed from two layers of brickwork each 105 mm thick with 50 mm mineral fibre between and 13 mm inside plaster, giving an equivalent U-value (overall thermal transmission coefficient) of 0.5 W/m²K. It has a single-glazed window of 3.5 m wide, 2 m high. The U-value for the window is 5.7 W/m²K. For comparison a double-glazed window with a U-value of 2.9 W/m² will also be used in place of the single-glazed window. The U-values are used only for estimating the amount of heat supply. In the process of flow computation, the thermal conductivity of the building materials rather than the U-value is used to calculate the heat transfer rate through the fabric. It is assumed that the surface temperature of the adjacent rooms is equal to that of the room under consideration and hence there is no net heat transfer through the internal walls. All the room surfaces are assumed to be grey and have an emissivity of 0.9.

The office is occupied by four persons. Each occupant generates sensible heat of 100 W and latent heat of 40 W. The moisture production rate is estimated from the amount of latent heat and is assumed to be a source at a node point close to head level. The occupants wear clothes equivalent to a clothing level of 0.8 clo (= 0.124 m²K/W). Heat production by lighting is 20 W/(m² floor area) and is assumed to be a heat source on the ceiling. The radiant component of heat from lighting is transferred to other room surfaces by radiation.

The quantity of supply air is estimated from the heat loss through the curtain wall and heat gains from the occupants and lighting. When the outdoor winter design air temperature is taken to be −1 °C and indoor design temperature 21 °C, heat gains from the occupants and lighting are nearly sufficient (~ 90%) to compensate for the heat lost through the curtain wall with the single-glazed window. Ventilation of the room is therefore mainly to provide fresh air and redistribute the heat generated. Hence, the supply air is set at a temperature of 21 °C and relative humidity of 50%. The ventilation rate is four air changes per hour (i.e. 16.7 litres/s-person), of which up to 50% could be recirculated air according to a minimum fresh air supply rate of 8 litres/s-person [3]. Air is introduced vertically upwards from the heating system beneath the window and room air is extracted through a duct on the wall opposite to the curtain wall.

Results and Discussion

Five simulations were performed using different boundary conditions for heat and moisture transfer. The five cases are designated in table 3.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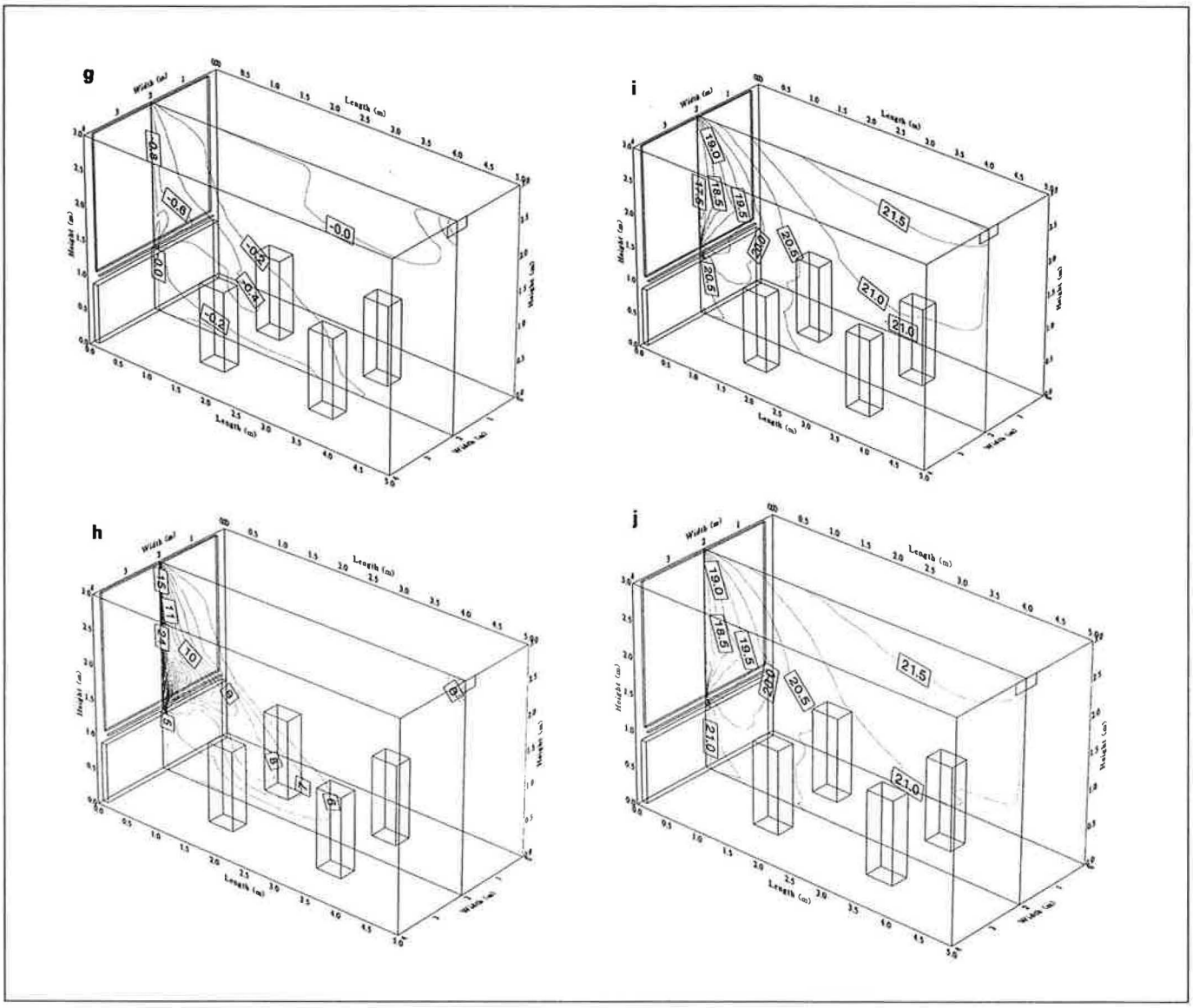


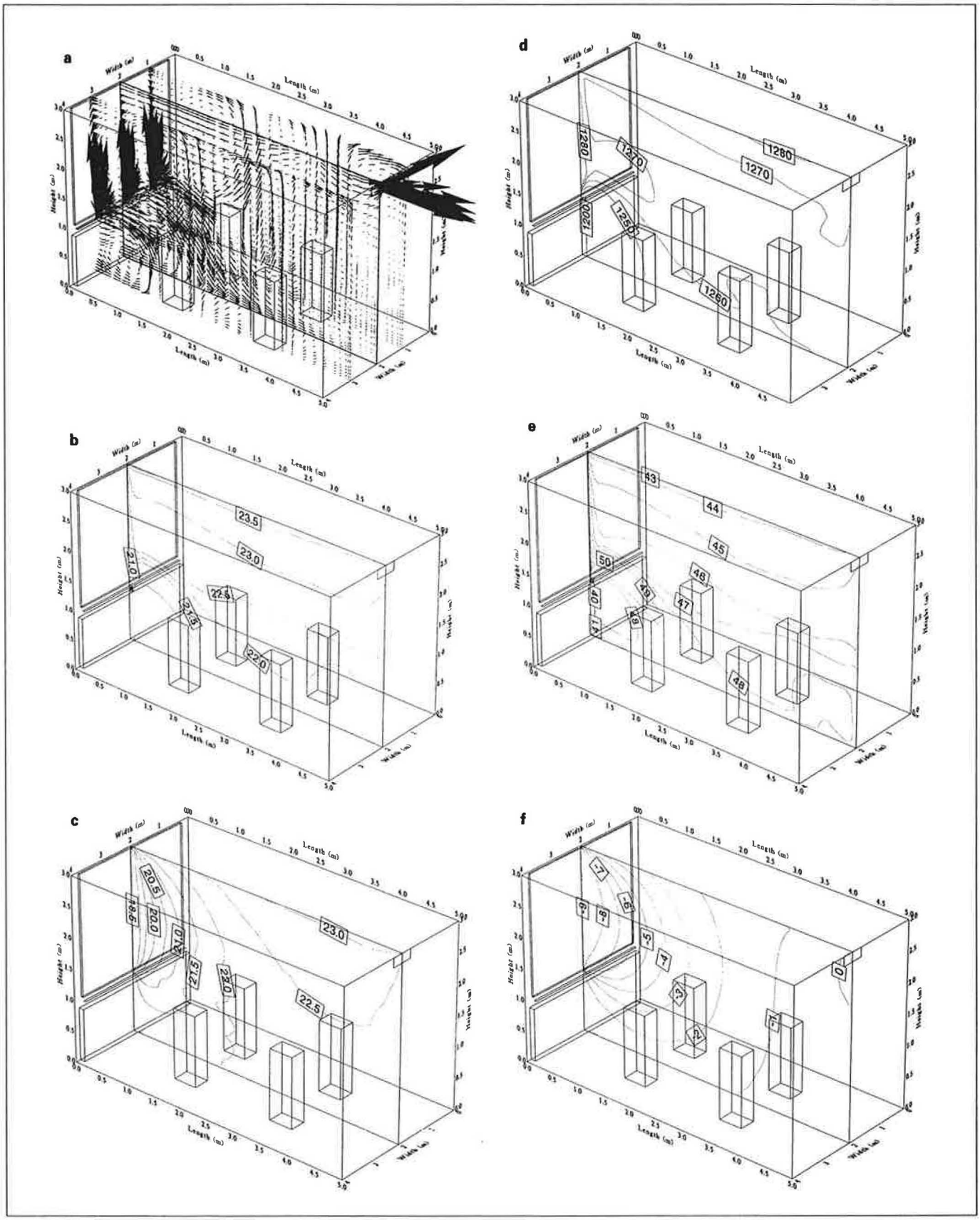
Fig. 1. Predicted thermal environment in the office with a single-glazed window. **a** Air velocity on three vertical sections. **b** Isotherms (°C) on a vertical plane. **c** Mean radiant temperature (°C) on a vertical plane. **d** Vapour pressure (Pa) on a vertical plane. **e** Relative humidity (%) on a vertical plane. **f** Radiant asymmetry (K) on a vertical plance. **g** Iso PMVs on a vertical plane. **h** Iso PPDs (%) on a vertical plane. **i** Operative temperature (°C) on a vertical plane. **j** Resultant temperature (°C) on a vertical plane.

Cases I and IV represent simulations using the general principles described above for single and double glazing.

In cases II and V, the direct radiation heat transfer between surfaces is ignored. This refers to the computation of heat transfer by radiation and convection using a combined heat transfer coefficient:

$$h = h_{ci} + 1.2\varepsilon_i h_{ri}, \quad (5)$$

where h_{ci} is the convective heat transfer coefficient for room surface i; ε_i is the emissivity of surface i; h_{ri} ($\approx 4\varsigma T_i^4$ where ς is Stefan-Boltzmann's constant and T_i is the absolute temperature of surface i) is related to thermal properties of the surface involved but, unlike in the full-radiation model, is independent of the rest of room surfaces. Thus, these two cases also involve some form of radiation but are very much simplified.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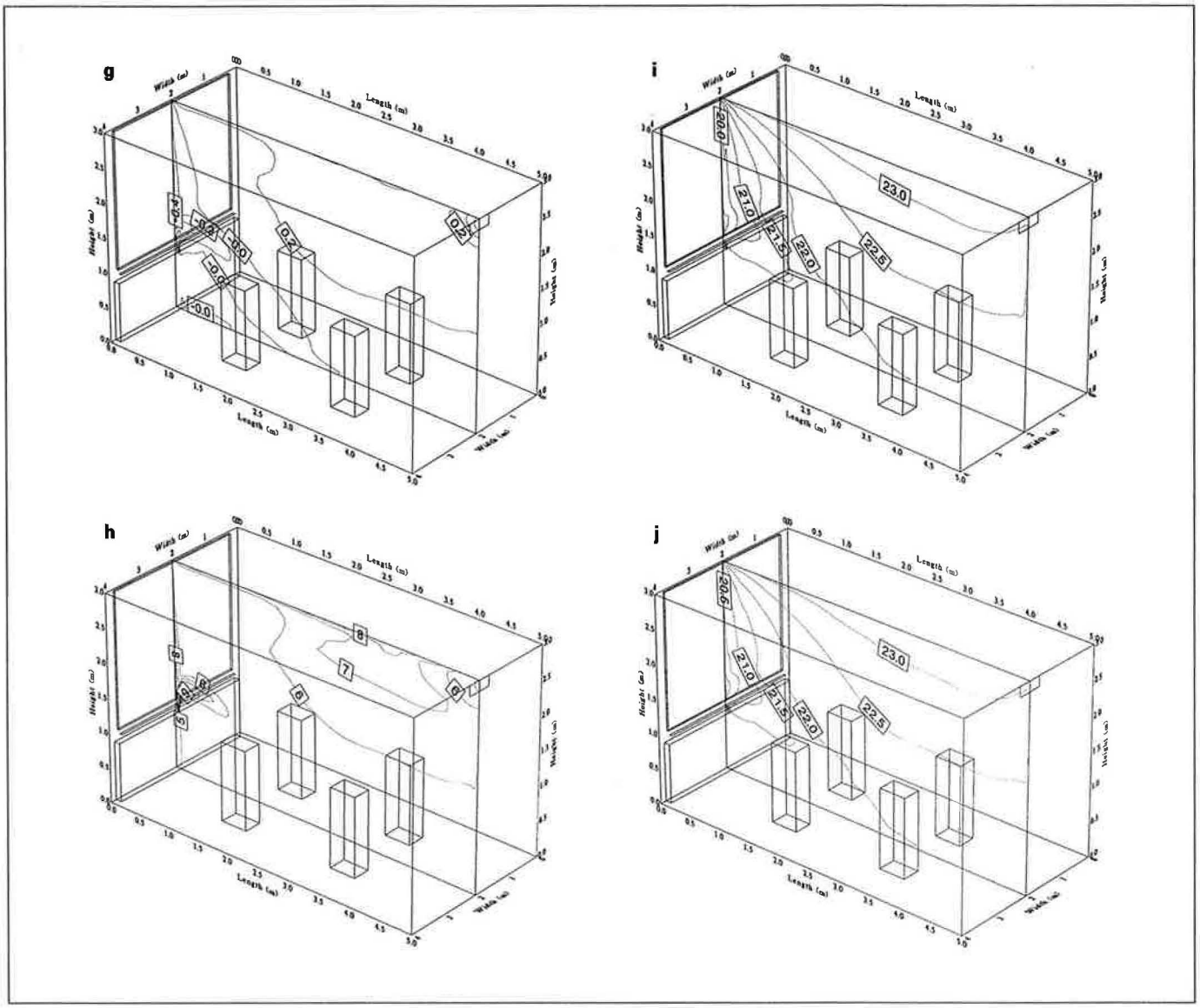

Fig. 2. Predicted thermal environment in the office with a double-glazed window. **a** Air velocity on three vertical sections. **b** Isotherms (°C) on a vertical plane. **d** Vapour pressure (Pa) on a vertical plane. **e** Relative humidity (%) on a vertical plane. **f** Radiant asymmetry (K) on a vertical plane. **g** Iso PMVs on a vertical plane. **h** Iso PPDs (%) on a vertical plane. **i** Operative temperature (°C) on a vertical plane. **j** Resultant temperature (°C) on a vertical plane.

Case III simulates conditions without the moisture transfer process in order to assess the effect of surface condensation on the room thermal environment. Moisture transfer can also result in the spatial variation of vapour pressure and thereby influence occupants' thermal comfort. This was discussed in the previous investigation [20].

Effect of Glazing on the General Thermal Environment

Figures 1 and 2 show the predicted air movement and distributions of comfort level for cases I and IV representing single and double glazing, respectively.

When the single-glazed window is used, in spite of the upward supply air, severe downdraught caused by the cold window is seen from the predicted velocity vectors in figure 1a. Apart from the supply air jet and deflected air

Table 3. Simulation regime

Case	Glazing	Radiation transfer	Moisture transfer
I	single	yes	yes
II	single	no	yes
III	single	yes	no
IV	double	yes	yes
V	double	no	yes

stream, the mean velocity in the space is generally low (<0.1 m/s) and the room air is essentially stagnant. The vertical air temperature gradient exists (fig. 1b) but the temperature difference from 1.1 m to 0.1 m above the floor is less than the comfort limit of 3 K [26]. This is due to immediate mixing of upward supply air with downward draughty air near the cold window. The mean radiant temperature (fig. 1c) near the window is much lower than the corresponding air temperature. As a result, the radiant temperature asymmetry occurs near the window (absolute value greater than the comfort limit of 10 K). The radiant asymmetry in figure 1f is for the direction normal to the window and the numerical value at a point in the space is defined as the difference between the plane radiant temperature facing the curtain wall and that facing the opposite wall. The vapour pressure and relative humidity in the space are relatively uniform although near the sources of moisture generation, i.e. occupants, their values are higher than the averages for the whole space (not shown on the plane in fig. 1d, e, though). The air velocity and heat transfer coefficients in the space are such that the predicted operative and dry resultant temperatures are nearly the same (fig. 1i, j). The distribution patterns of operative and resultant temperatures generally follow the distribution of mean radiant temperature and the magnitude is, as expected, between the air and radiant temperatures. Overall, except for the area near the window where it is slightly cool (PMV < 0.0 and PPD > 10%), thermal comfort in the room is satisfactory as indicated by the comfort indices in figure 1g–j. The average value for PPD is below 10% and the predicted operative, and dry resultant temperatures are between 20.0 and 21.5 °C.

When the window is double glazed, it is seen by comparing figures 2 with figure 1 that the distributions of air, radiant, operative and resultant temperatures as well as other parameters are much more uniform than those predicted for single glazing. Although slight downdraught still exists in terms of air movement, all the thermal comfort indices are within comfort limits with PPD < 10% and radiant asymmetry being less than 10 K in the whole space. Thus, local discomfort is avoided that would occur near the single-glazed window. This clearly demonstrates that double-glazing improves the thermal environment in the office. Besides, to achieve the same comfort level, the quantity or the temperature of supply air which was determined for the single-glazed window conditions could be reduced since the average value for PMV in the office using double-glazing is higher than that using single glazing (both within the comfort limit). This would give rise to energy savings.

Effect of Boundary Heat/Mass Transfer on the Thermal Environment

The effect of heat and mass transfer through the room boundaries on the predicted room surface temperatures and thermal environment in the space is shown in table 4.

When both direct radiation heat transfer and moisture transfer are taken into consideration (cases I and IV), the window temperature is higher and less or no condensation occurs than when only one of the transport phenomena is considered (cases II, III and V). Without considering direct radiation heat transfer, condensation would take place on the window (i.e. $q_1 > 0$) even if it is double glazed (case V). This shows the importance of taking radiation heat transfer into consideration when predicting the potential of moisture condensation. This is because a cold room surface (i.e. window) would receive heat from other enclosing surfaces through radiation and consequently its temperature would rise compared with a surface without heat gain through radiation heat transfer (e.g. a surface with low emissivity). On the other hand, the average room surface temperature is decreased by 2 to 3 °C when the radiation model is used. This is due to heat transfer from warm surfaces, in particular the ceiling with heat production, to and subsequent loss through the window. Malalasekera and James [27] also observed that a 5 K increase in the temperature of a heated surface could result in a sig-

Table 4. Predicted thermal environment in the office

Parameter	Simulation regime				
	I	II	III	IV	V
T_{win}, °C	5.29	4.81	4.70	13.31	11.88
Ts, °C	19.67	22.52	19.25	21.94	24.12
q_l, W/m^2	12.29	15.25	–	0.0	1.58
V, m/s	0.067	0.083	0.070	0.066	0.072
T, °C	21.51	21.48	21.17	22.15	21.83
C, kg/kg	0.0078	0.0077	–	0.0078	0.0078
P_v, Pa	1,270	1,257	–	1,268	1,264
RH, %	49.49	49.09	–	47.51	48.31
T_{mr}, °C	19.84	22.49	19.43	22.0	23.86
PMV	−0.16	0.07	−0.23	0.11	0.25
PPD, %	5.88	5.60	6.53	5.51	6.72
T_o, °C	20.65	22.0	20.27	22.07	22.88
T_{res}, °C	20.70	21.97	20.33	22.08	22.82

nificantly different heat flux distribution on walls and that heat losses through a cold window were high in a room with heated ceiling.

The predicted air velocity is slightly lower with the use of the radiation model than without because less downdraught near the window occurs. The effect of radiation heat transfer on the average humidity level in the room is negligible. The radiant temperature and values for the relevant comfort indices are lower with radiation heat transfer than without because of the decrease of room surface temperature. The difference in radiant temperature due to radiation is similar to that for the room surface temperature (~2°C).

The effect of moisture transfer is such that when condensation occurs on the single-glazed window the surface temperature increases due to the release of latent heat. This leads to a small increase in all the temperature levels in the space. The overall impact is, however, small with an average temperature increment of less than 0.5°C. For the double-glazed window, the effect is negligible as there would be no condensation under this room environment provided that direct radiation heat transfer is taken into account. However, if condensation takes place when outdoor air is much colder than used for simulation for example, the effect of moisture transfer should be considered for the same reason as for the single-glazed window.

Conclusions

The numerical analysis of the indoor thermal environment has shown that radiation heat transfer and moisture condensation are important processes and should be incorporated in comprehensive evaluation of occupants' thermal comfort when these transport phenomena arise.

Ignoring direct radiation heat transfer would underpredict heat loss/gain in enclosures with substantial temperature difference between surfaces. This means overpredicting radiant and related temperatures in winter conditions and likewise underpredicting the parameters under hot summer conditions.

Ignoring moisture transfer would result in underpredicting the room surface temperature as well as air and radiant temperatures in the space when condensation took place in winter.

For accurately assessing window condensation potential both direct radiation heat transfer and moisture transfer must be taken into account.

Acknowledgements

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Appendix

C	mean concentration of moisture (kg/kg)
C_s	moisture generation rate (m^3/s-m^3)
H	mean specific enthalpy (J/kg)
g_i	gravitational acceleration in i direction (m/s^2)
h_c	convective heat transfer coefficient (W/m^2K)
h_r	radiative heat transfer coefficient (W/m^2K)
k	turbulent kinetic energy (m^2/s^2)
L	characteristic length of supply opening (m)
M_a	molecular weight of dry air (kg/mol)
M_v	molecular weight of water vapour (kg/mol)
n	normal distance from a solid surface (m)
PMV	predicted mean vote
PPD	predicted percentage of dissatisfied (%)
p	static pressure of air (Pa)
p_a	partial pressure of dry air (Pa)
p_v	partial pressure of water vapour (Pa)
p_{at}	atmospheric pressure (Pa) ($p_{at} = p_a + p_v$)
q	volumetric rate of heat production (W/m^3)
q_l	latent heat release due to condensation on a window (W/m^2)
R	universal gas constant (8.314 J/mol K)
RH	relative humidity (%)
T	mean air temperature (°C)
T_{mr}	mean radiant temperature (°C)
T_o	operative temperature (°C)
T_{res}	resultant temperature (°C)
T_{win}	temperature of inner surface of a window (°C)
T_s	mean temperature of room surfaces (°C)
U_i	mean velocity component in i direction (m/s)
V	resultant mean velocity (m/s)
V_g	mean velocity component in gravity direction (m/s)
V_h	mean velocity component in horizontal direction (m/s)
V_s	spatial mean velocity across the supply opening (m/s)

Greek Letters

δ_{ij}	Kronecker delta
ε	dissipation rate of turbulent kinetic energy (m^2/s^3)
κ	von Karman constant
μ	laminar dynamic viscosity (kg/m-s)
μ_t	turbulent dynamic viscosity (kg/m-s)
ρ	air density (kg/m^3)
ρ_r	air density at a reference point (kg/m^3)
σ	laminar Prandtl number
σ_c	turbulent Schmidt number
σ_s	laminar Schmidt number
σ_t	turbulent Prandtl number

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