Coupling Effect on Thermal Comfort in a Typical Cubicle–based Office with Personalized Floor Diffuser Control

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***Abstract*—A typical office layout with cubicles, in which occupants have their own control of the micro–environment by adjusting supply air flow rate of the floor diffuser, is numerically investigated for the impact of the discrepancy in personal thermal sensation preference on thermal comfort. The comparison among different scenarios indicates that whether the local thermal comfort is significantly affected by the neighboring cubicle (coupling effect) depends on whether the doorway is closed or not whereas the "openness", of upper space has no influence on such coupling effect but observably on the thermal comfort. The effect of thermoregulation is also presented and compared with conventional constant heat flux assumption for the occupants.**

I. Introduction

Thermal comfort has been reported not merely related to sensation of satisfactory, but more vitally to health due to that human thermoregulatory system is only effective for a small range of ambient temperature range, beyond which many previous study has shown the deterministically resultant morbidity or mortality for either too hot [1][2] or too cold [3][4] conditions. Nevertheless, other author [5] also suggests that the human thermoregulatory system need "exercise" by intentionally and regularly exposing human body into thermally uncomfortable surroundings for a limited period of time which is analogous to a well–established relationship between regular physical exercise and cardiovascular health. This leads a further step to only thermal sensation preference that people's choice of thermal environment might be based on their different understanding that minimization of thermal discomfort is either or not necessarily beneficial to health. Plus nowadays people spend a considerable portion (about 90 % [6]) of their daily time indoors while in which, each thermal zone comprised of several office spaces, has its own temperature set point, or thermostat. The resultant "One–Size–Fits–All: OSFA" thermal environment may leave as many as 20% of the occupants dissatisfied and less productive. In order to satisfy every person's thermal comfort requirement, individual control of the thermal micro–environment of each occupant is required, i.e., adopting a "Have–It–Your–Way: HIYW" approach [7].

The present work is primarily supported by Forskningsradet Nærings-Ph.d (Project No.: 231952), Scientific Research Project (Project No. A2620110012), VRI bedriftsmidler (Project No.: 235042) and Regionale Forskningsfond Oslofjordfondet (Project No.: 235809).

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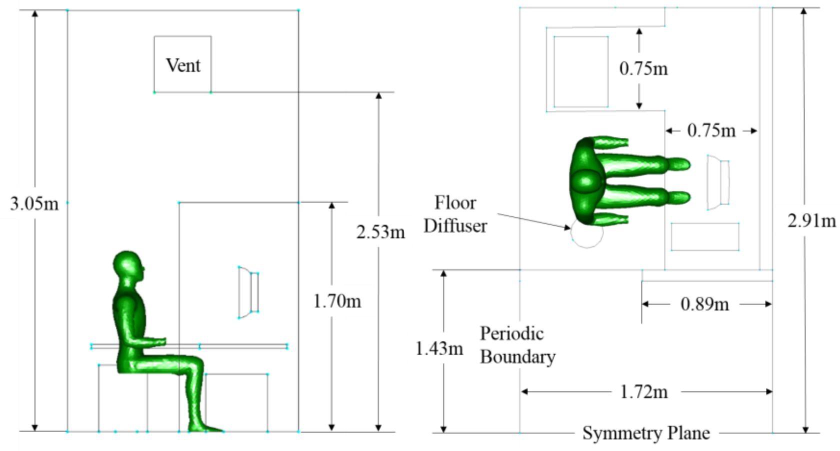
However, in this kind of HIYW approach, it is expected that flows of different temperature will get mixed in such a half–open space and sometimes even exceeds the original design purpose of thermal comfort. Simon[8] et al. analyzed the impact of varying supply air temperature on occupant in adjacent cubicles. From the result of CFD simulation by assuming that the occupants and computer cases are heat sources with constant surface fluxes, it is concluded that the effect of this cross flow on the personal micro–environment is relatively insensitive to diffuser swirl direction or to the arrangements of cubicles opening to the same corridor: side–by–side symmetrical, side–by–side periodic, opposite symmetrical, or opposite anti–symmetrical. Impeding this corridor flow would increase the effectiveness of individual thermal control via occupant–regulated supply air temperatures. A practical way recommended for impeding the corridor flow is to install pocket doors near the floor in the openings of the cubicle to the connecting corridor.

In the present study, since most commercial centralized air–conditioning systems, like the widely used variable air volume (VAV) system[9], tend to have more flexibility in controlling flow rate instead of temperature of the supply air, simulation by setting different supply air flow rates is carried out. On the other hand, as a comparison a thermoregulatory model is incorporated in the simulation for thermal boundary setting of the occupant (manikin) for further discussion.

II. Computational Model

Since it is expected that the occupants' thermal comfort is most likely to be affected by the supply air condition of its adjacent cubicles, a bi–cube model has been set up to investigate this coupling effect. The bi–cubicle computational domain is created by duplicating the single cubicle in a side–by–side periodic [8] way as shown in Fig. 1. There are three surface heat sources, the monitor, the computer case and the manikin. The monitor is assumed to be of uniform heat flux, generating 37 W of sensible heat. Again with uniform heat flux, an assumption of 200W total heat generation is applied to the back side (facing the partition wall in front of the manikin) of the computer case located to the right side of the manikin's right leg under the desk. The rest four surfaces of the computer case is assumed to be adiabatic. Two different types of thermal boundary conditions are applied to the manikin with a total surface area of 1.8 m^2 [10] in different cases which will be discussed later in this paper. Interior partition walls, external boundary walls, surfaces of the desk and cabinets to the left of the manikin under the desk are assumed to be adiabatic. The floor diffuser with a swirl magnitude (the ratio of tangential velocity to axial velocity which is normal to the boundary) of 1 is assumed to supply the



(a) Front view (b) Top view

Figure 1. Cubicle layout with corridor

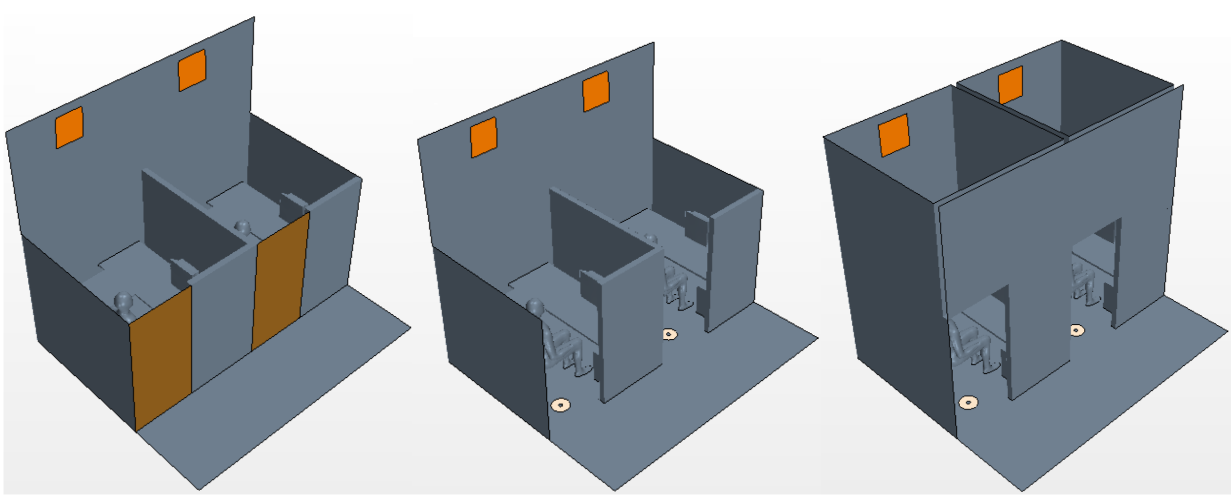
conditioned space with a constant air temperature of 16°C and various air change rates (ACH) to up 6, a typical office ventilation system [11]. The vent located on the upper wall to the left of the manikin is set as an outlet with constant pressure of 1 atm.

The *k*–ε model is selected with enhanced wall treatment for the viscous sub–layer near the wall in the present work for turbulence, the y^+ for the next–to–wall cell layer below 5. To account for the radiation heat transfer which is 38 – 58% of the total sensible heat transfer rate, a discrete ordinate (DO) model is applied for all simulations presented in this article for higher accuracy compared to the surface–to–surface (S2S) model which is although computationally less expensive [12].

III. Result and Discussion

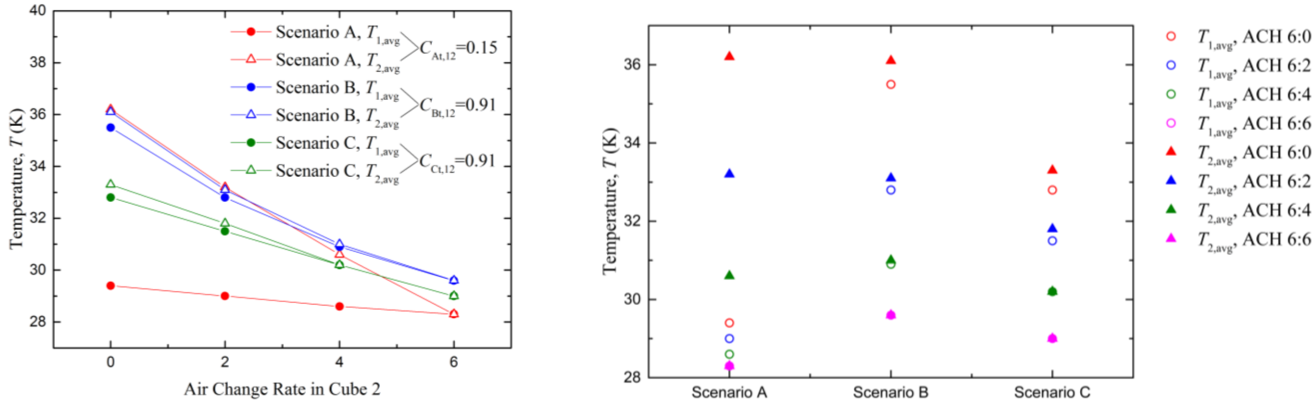
Fig. 2 Different layouts of the computational domain, Scenario A, doorway closed, upper space open; Scenario B, doorway open, upper space open; Scenario C, doorway closed, upper space closed.

A constant–heat–flux (i.e. 1 met or 58.1 W/m^2 for a seated person with regular office work [10]) assumption is made for the manikin, while the variation of the area–weighted surface temperature of the manikin T_{avg} is used to evaluate the thermal comfort impact from one cubicle to its adjacent. In this analysis, the same concept of coupling parameter, as is used in Simon's work, is applied to define the magnitude of such impact.



Scenario A Scenario B Scenario C

Figure 2. Different layouts of the computational domain, Scenario A, doorway closed, upper space open; Scenario B, doorway open, upper space open; Scenario C, doorway closed, upper space closed.



(a) (b)

Figure 3. Area–Weighted Surface Temperature of Manikin vs. Air Change Rate in Cube 2 (Air Change Rate is Held Constant at 6 ACH) for different scenarios.

$$C_{\mathrm{ij}} = \frac{S_{\mathrm{ij}}}{S_{\mathrm{jj}}} \tag{1}$$

where, S is the mean slope of the linear regression curve depicting the variation of T_{avg} (or heat flux q, which is distinguished by the subscript "t" or "q" later in this paper) of the occupant versus the supply air flow rate and the subscript stands for the relationship between cubicles. In another word, i is one of the cubicle being investigated, j is another one adjacent to it and S_{ij} therefore represents the effect of air flow change in cubicle j on the thermal comfort index, i.e. area–weighted surface temperature, of the occupant in cubicle i. Since the change of air flow rate may impact the local occupant more significantly than those somewhere else, it is expected that the impact indicator denoted by S_{ij} will fall between 0 and 1 and can be used as an index for quantifying this coupling effect after being normalized by the local impact indicator S_{jj}.

Fig. 3(a) shows the variation of T_{avg} vs. ACH for all given scenarios in Fig. 2. Two manikins sit in their individual cubicles, of which the one in front of the other is hereinafter referred to as Cube 2, the other Cube 1. The supply air flow rate in Cube 1 is held at 6ACH while in Cube 2 it varies from 0ACH to 6ACH in a step length of 2ACH. For Scenario A, the resultant $T_{1,\mathrm{avg}}$ (subscript 1 or 2 hereinafter represents manikin in Cube 1 or Cube 2 respectively) decreases with the increase of air flow rate change in Cube 2 from 29.4°C to 28.3°C meanwhile $T_{2,\mathrm{avg}}$ decreases from 36.2°C to 28.3°C, which yields a coupling parameter $C_{\mathrm{At},12}$=0.15 meaning that the air flow rate change in Cube 2 has very limited impact on the thermal comfort of the occupant in Cube 1. Scenario B and C yield $C_{\mathrm{Bt},12}$= $C_{\mathrm{Ct},12}$=0.91 which is way higher than that of Scenario A. However in Fig. 3(b), Scenario B is always worse than Scenario C since either $T_{1,\mathrm{avg}}$ or $T_{2,\mathrm{avg}}$ in Scenario B is always higher than that in Scenario C provide the same air flow rate settings while compared to Scenario B, $T_{2,\mathrm{avg}}$ in Scenario A varies from a little higher to lower than that in Scenario B with the increasing ACH in Cube 2, a similar trend observed when comparing Scenario A and C. The variation of $T_{1,\mathrm{avg}}$ decreases sequentially from Scenario B, A and C for cases with the same ACH in both cubicles although the difference between any two of the three scenarios keeps being diminished.

On the other hand, if the heat flux of the manikin is used as an indicator of thermal comfort, similar coupling effect should be expected. However, the CFD result (Fig. 4) turned out not so satisfactory. For instance, one of the cases ended up with negative heat flux of the occupant which physically means that to keep the surface temperature of the manikin constant, which is typically 28 °C for a person with light office work and regular garment[10], a person will need to gain heat from the ambient, which is physiologically not possible. This inversely signified the importance of taking the thermoregulation into consideration while modeling the surface temperature in an isothermal way.

Based on energy balance and empirical correlation between skin temperature and heat generation from human body with thermal comfort, Fanger[13] recommended the following equation to calculate the clothing temperature (the manikin surface temperature in current model),

$$\begin{cases} t_{\mathrm{cl}} = 35.7 - 0.028\left(M - W\right) - R_{\mathrm{cl}}\left(R + C\right) \\ R = 39.6\times10^{-9} f_{\mathrm{cl}}\left[\left(t_{\mathrm{cl}} + 273\right)^4 - \left(\bar{t}_{\mathrm{r}} + 273\right)^4\right] \\ C = f_{\mathrm{cl}} h_c \left(t_{\mathrm{cl}} - t_{\mathrm{a}}\right) \end{cases} \tag{2}$$

where, M – metabolic rate, W/m^2

W – mechanical work output, W/m^2

R_{cl} – clothing insulation, m^{2}K/W

f_{cl} – clothing area factor

$\bar{t}_{\mathrm{r}}$ – mean radiant temperature, K

h_{c} – convective heat transfer coefficient, W/(m^{2}K)

t_{a} – ambient air temperature, K

R – radiative heat transfer rate, W/m^2

C – convective heat transfer rate, W/m^2

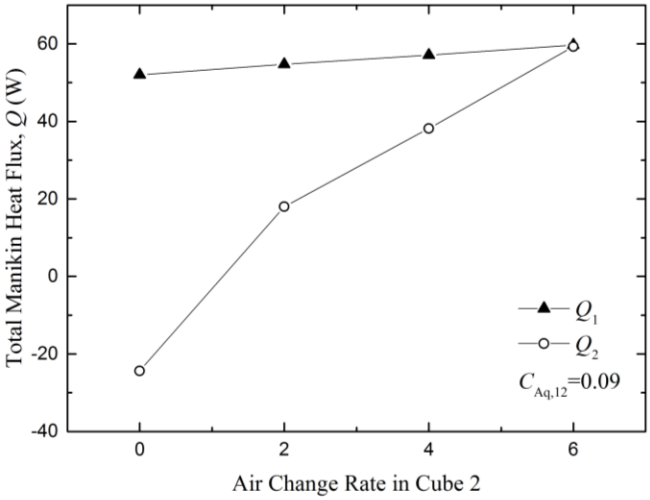


Figure 4. Manikin heat flux vs. air change rate in cube 2 (air change rate is held constant at 6ACH in cube 1) for Scenario A.

The convective and radiative heat transfer rate can be obtained from the built–in function that is available in current CFD package and the clothing temperature will be updated from the result of last iteration. In Fig. 5, $T_{1,avg}$ decreases from 28.3°C to 26.4°C quasi–linearly while $T_{2,avg}$ descends from 28.9°C to 26.4°C, which is comparatively a narrower band to the corresponding scenario in Fig. 3(a) where a variation of 35.5°C→29.6°C and 36.1°C→29.6°C for $T_{1,avg}$ and $T_{2,avg}$ is observed respectively. The coupling parameter $C_{Bt,12}$ also decreases from 0.91 to 0.77 indicating an attenuation of the coupling effect by thermoregulation.

IV. CONCLUSION

A numerical study on occupants' thermal comfort in a bi–cubicle computational domain has been carried out. Three different layout scenarios concerning the doorway and the upper space are compared with different supply air flow rate settings.

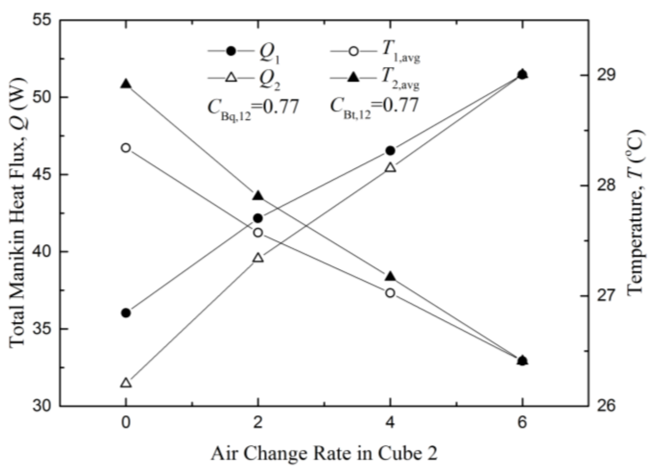


Figure 5. Manikin heat flux & area–weighted surface temperature vs. Air change rate in cube 2 (air change rate is held constant, 6ACH) for Scenario B using the thermoregulation model in (2).

Based on the coupling parameter defined in the literature, it is found that whether or not have the doorway blocked has great impact on the coupling effect of the occupants' thermal comfort in adjacent cubicles. For scenarios with the doorway open, an open upper space yields better thermal comfort for occupants in both cubicles. The thermoregulation model is applied and compared with the constant heat flux assumption for the occupants. The tendency to avoid a rapid change in clothing temperature with the variation of air change rate is obvious, which is consistent with the fact that thermoregulation is in nature an organism to keep the body temperature within a certain range.

ACKNOWLEDGMENT

RotoBoost AS which initiates and facilitates the present work is hereby acknowledged.

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