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Effect of furniture and partitions on the surface temperatures in an office under forced flow

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ABSTRACT

Computational Fluid Dynamics (CFD) simulations were performed to study the effects of surfaces typically found in an office, such as partitions and desks, on the temperatures of the surfaces surrounding the occupants. An accurate estimation of these temperatures is needed to assess the thermal comfort conditions in a space, thus ensuring that the occupants are satisfied with their thermal environment. A parameter quantifying the area available to convect heat to the air was proposed to account for the effect of surfaces of different sizes and positions. A higher amount of available convective area was found to decrease the temperatures of the surfaces in the room, and vice versa. Although the air temperature also increased with additional available convective area, this change was found to be small (typically below 15%). Moreover, the shape of the air temperature profile far from the surfaces remained unchanged when changing the amount of available convective area. Therefore, the operative temperature that the occupants experience is lower in spaces with a large amount of available convective area. Expressions to estimate the temperatures of the surfaces, generally within ±10% of the air temperature rise in the space, were proposed. These expressions can be useful during the early stages of the design process, when CFD simulations are impractical and the exact location and type of furniture in a space are not well known. Together with a model to predict the air temperature profile, these correlations can improve the accuracy of thermal comfort assessment.

1. Introduction

The thermal environment conditions that ensure that occupants are comfortable in a space refer not only to the air temperature or velocity near them, but also to the temperatures of the surfaces that surround the occupants. These surfaces exchange heat with the occupants via radiation, so surfaces that are too hot or cold can result in unacceptable thermal conditions in the space, even if the air temperature is by itself acceptable [1]. Asymmetry in the temperatures of these surfaces can also result in thermal dissatisfaction [1]. The Graphical Comfort Zone Method for Typical Indoor Environments and the Optional Method for Determining Acceptable Thermal Conditions in Naturally Ventilated Spaces – the adaptive comfort method – of the ASHRAE Standard 55-2010 take into account the radiation exchange between people and surfaces by means of the operative temperature [1].

Models commonly used to assess the thermal conditions in offices, including CFD and multi-zone models, can be used to compute the operative temperature experienced by the occupants. However, the geometries of these models rarely include surfaces found in a real office such as partitions and desks (e.g. [2]). These surfaces can have an important effect on the temperature of all the surfaces in the room, including the ceiling, floor and walls, because they facilitate the transfer of heat from the sources (e.g. lighting, equipment and occupants) to the air.

Ultimately, the temperatures of the surfaces in an office depend on several factors, including convective heat transfer coefficients associated with the surfaces and the view factors between them, among others. Taking into account all these parameters is time-consuming and might not be appropriate during the early stages of the design process, when the exact position of the furniture and partitions in an office is not well defined. Hence, this work presents a simple method to estimate the temperatures of the surfaces in a room that does not require exact knowledge of the position of the furniture or the partitions.

1.1. Heat transfer paths in an office

In a general sense the heat transfer processes in a space transport thermal energy from the source(s) to the heat sink(s). Heat is trans-

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Nomenclature

β	volumetric thermal expansion coefficient of air
ΔT_r	air temperature rise across room
θ	dimensionless temperature

Roman symbols

A	fitting constant
A_{conv}	unheated area available for convection heat transfer
A^*_{conv}	dimensionless unheated convective area
Ar_{h_w}	Archimedes number based on inlet height
A^*_d	dimensionless desk area
A^*_p	dimensionless partition area
A^*_w	dimensionless inlet area
B	fitting constant
C	fitting constant
c_p	air heat capacity at constant pressure
D	fitting constant
E	fitting constant
F	fitting constant
G	fitting constant
H	room height
h_c	convective heat transfer coefficient
h_r	radiative heat transfer coefficient
h^*	dimensionless height coordinate
h_w	inlet height
h^*_w	dimensionless inlet height
I	fitting constant
J	fitting constant
K	fitting constant
L	room length
l^*	dimensionless length coordinate
$\dot{m}$	airflow rate
$\dot{Q}$	total heat gains
R^c_f	floor convective resistance
R^c_p	partition convective resistance
$R^r_{c,f}$	radiative resistance from the ceiling to the floor
$R^r_{c,p}$	radiative resistance from the ceiling to a partition
$R^r_{p,f}$	radiative resistance from the partition to the floor
$\bar{T}_a$	mean air temperature
T_{air}	air well-mixed temperature
T_{ceiling}	ceiling temperature
T_{floor}	floor temperature
T_{inlet}	inlet/supply temperature
T_o	operative temperature
T_{outlet}	outlet temperature
$T_{\text{partition}}$	partition temperature
$\bar{T}_{\text{rad}}$	mean radiant
u	inlet velocity
x	x coordinate
y	y coordinate
z	z coordinate

ferred from active surfaces such as lights, equipment and people by convection to the air and by radiation to the walls, floor, partitions and furniture, which is then transferred to the surrounding air. In this work, it is assumed that lights, equipment and occupants are the only heat sources in the room, while the air in the space is the only heat sink in steady state. The temperature of the air in the space is not uniform, instead it is higher near the ceiling in spaces with either displacement or mixing ventilation systems [3]. However, multi-zone models cannot be used to compute the

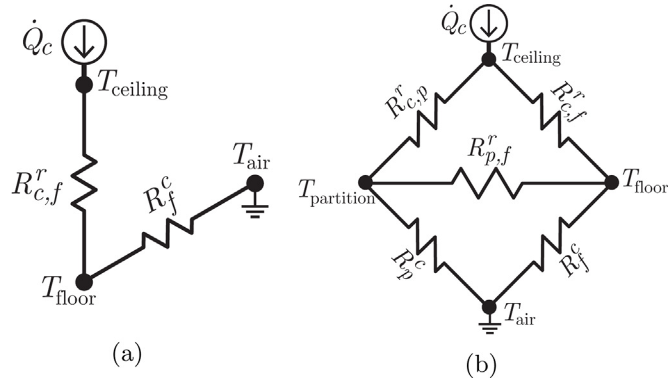

Fig. 1. Thermal circuit representation of a space with heat gains generated at ceiling level (a) without partition and (b) with a partition.

air temperature profile, instead they rely on the assumption of perfect mixing of the air, so the temperature in a zone is uniform and equal to the temperature of the outlet. This so-called "well-mixed" temperature, T_{air}, can be obtained using the steady-state energy conservation equation:

$$T_{\text{air}} = T_{\text{outlet}} = T_{\text{inlet}} + \frac{\dot{Q}}{\dot{m}c_p} \tag{1}$$

where T_{outlet} is the outlet temperature, T_{inlet} is the inlet temperature, $\dot{Q}$ are the total heat gains in the room, $\dot{m}$ is the mass flow rate of air through the space and c_p is the heat capacity of air at constant pressure. The well-mixed temperature serves as a convenient reference for the present work because its value is independent of the amount of furniture or the distribution of heat gains in a room under forced flow (wind-driven natural ventilation or mechanical ventilation). Moreover, this temperature can be calculated using multi-zone models according to interior and, in the case of natural ventilation, exterior conditions.

A thermal resistor network is a useful tool to represent the heat transfer paths in a room. As a simple example, consider a very long and wide space, where the view factors from the ceiling to the walls are small enough to be neglected. Assume also that the temperature of the surfaces are uniform and that the only heat source is due to lighting (or solar heat gains), which is located at the ceiling level, while the other surfaces are adiabatic. In a real room, except for other heat sources, the ceiling is generally the warmest surface in the space because hot air that rises due to buoyancy stagnates under it. As a consequence, the convective heat transfer coefficient associated with the ceiling is significantly lower than that of the other surfaces in the room. The only effective mechanism available to transfer heat from the ceiling is via radiation. The same holds for the simple example considered here, therefore it is possible to assume that the convective heat transfer from the ceiling to the air is negligibly small. This is a common assumption in ventilation studies (e.g. [4]). Fig. 1a shows the thermal resistor network of this example. In this figure, $R^r_{c,f}$ is the radiation thermal resistance between the ceiling and the floor, and R^c_f is the convective thermal resistance from the floor to the air. T_{ceiling}, T_{floor} and T_{air} are the temperatures of the ceiling, floor and the well-mixed temperature of the air, respectively. Fig. 1b shows the same space but with a thin partition in the room. In this figure, $R^r_{c,p}$ and $R^r_{p,f}$ are the radiation thermal resistances between the ceiling and the partition, and between the partition and the floor, respectively. R^c_f is the convective thermal resistance from the partition to the air and $T_{\text{partition}}$ is the temperature of the partition. The well-mixed temperature of the air does not change when adding the partition. Now some of the radiation from the ceiling is directed to the partition, reducing the direct radiation from the ceiling to the floor. The total surface area