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Development of new and validation of existing convection correlations for rooms with displacement ventilation systems

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Abstract

Building airflow, thermal, and contaminant simulation programs need accurate models for the surface convective boundary conditions. This is, especially, the case for displacement ventilation (DV) systems, where convective buoyancy forces at room surfaces significantly affect the airflow pattern and temperature and contaminant distributions. Nevertheless, for DV, as a relatively new ventilation system, the convective correlations are adopted from more traditional mixing ventilation correlations, or non-existent. In this study, the existing recommended correlations are validated in a full-scale experimental facility representing an office space. In addition, new correlations are developed for floor surfaces because the current literature does not provide necessary correlations, even though, the floor surface is responsible for >50% of the total convective heat transfer at the envelope. The convective correlations are typically functions of a surface-air temperature difference, airflow parameters, and characteristic room dimensions. Validation results show that the floor convection correlations expressed as a function of volume flow rate are much stronger than the correlations expressed as a function of a temperature difference between the surface and local air. Consequently, the new convection correlation for floor surfaces is a function of the number of hourly room air changes (ACH). This correlation also takes into account buoyant effects from local floor heat patches. Experimental data show that the existing correlation can be successfully applied to vertical and ceiling surfaces in spaces with DV diffuser(s). Overall, the new and the existing convection correlations are tabulated for use in building simulation programs, such as annual energy analyses or computational fluid dynamics.

Keywords: Convection correlation; Displacement ventilation; Commercial buildings; Laboratory experiment

1. Introduction

One of the most important factors in designing mechanical systems for buildings is defining accurate thermal boundary conditions. Convection at the internal room surfaces has a large impact on the total heat transfer and varies based on the ventilation system being used. With ventilation systems that utilize displacement diffusers, the temperature field is vertically stratified and the low-momentum supply jet is attached to the floor as shown in Fig. 1. This specific airflow pattern and temperature distribution has several potential advantages related to air quality when compared to distributions in traditional mixing ventilation systems [1]. The popularization of displacement ventilation (DV) systems creates an incentive to validate existing convection correlations or develop new correlations specifically for these systems.

In the past decade, several simplified models were developed for the temperature stratification calculations in rooms with DV [2,3]. Furthermore, Chen et al. [1] developed design guidelines for calculation of the temperature difference between the occupants' head and ankles. The design guidelines also give an equation for calculating ventilation effectiveness at the breathing level. Even though, temperature stratification and ventilation effectiveness models are very sensitive to wall convection coefficients, they do not include correlations for their calculations. Chen et al. [1] used average convection coefficients, such as 4 W/(m^2 K) for floor surfaces. Other researchers recommend similar average values or use of convection correlations developed for room surfaces with all natural convection in

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ACH	air changes per hour [h^{-1}]
c	coefficient for determining forced convective heat transfer coefficient
c_L	coefficient for determining convective heat transfer coefficient under laminar flow
C_L	coefficient for determining Nusselt number under a laminar flow
c_T	coefficient for determining convective heat transfer coefficient under turbulent flow
C_T	coefficient for determining Nusselt number under a turbulent flow
D_h	hydraulic diameter [m]
h	convective heat transfer coefficient [W/(m^2 K)]
k	conductivity of the air [W/(m K)]
m	exponent coefficient for forced convection
n	exponent coefficient in Churchill and Usagi equation
Nu	Nusselt number
Pr	Prandtl number
q	heat flux [W/m^2]
Re	Reynolds number
T	temperature [°C]
ΔT	characteristic temperature difference [°C]
v_{air}	viscosity of the air [m^2/s]
V_{room}	volume of the room [m^3]
V_{supply}	supply flow rate from the diffuser [m^3/s]

the room [4], isolated surfaces and free edge heated plates [5], or room surfaces with well mixed air and heated room surfaces [6]. Currently, these correlations have not been experimentally validated for use in rooms with displacement ventilation systems, rather they are recommended based on the surface flow similarities to the flow condition in the original experiments.

The use of constant convective coefficients creates inaccuracies, especially for floor surfaces where a major part of the convective heat flow at the room envelope occurs (>50%). The existing correlations for natural convection do not take into account effects from the DV diffuser jet. On the other hand, existing forced convection correlations are not suitable because they are developed for a standard ceiling diffuser [7], or for diffusers where the jet discharge velocity has a large impact on convection at the floor [8], which is not the case with DV diffusers. Accordingly, the first objective of the present study is to develop a new convection correlation for the floor surface in rooms with the side-positioned DV diffuser. The second objective is to validate the existing wall surface correlations for application with the DV system. To accomplish these two objectives, full-scale experiments were conducted in test chambers with displacement ventilation.

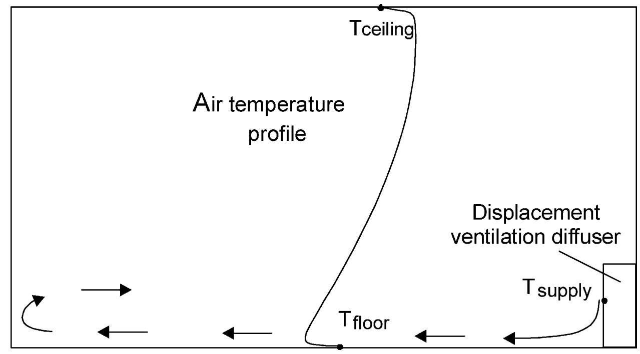

Fig. 1. A characteristic airflow pattern and temperature stratification in a room with displacement ventilation.

2. Methodology for deriving convection correlations

The experiments in the full-scale environmental chamber followed a methodology based on the conservation of energy at room surfaces. This methodology is similar to the methodology used by Khalifa and Marshall [9], Spitler et al. [10], and Awbi and Hatton [4]. All of these studies developed convection correlations for application with different heating and cooling systems that create relatively uniform room temperature distribution. Fig. 2 presents the conservation of energy at internal room surfaces, under steady-state heat flow, for determining the convective heat fluxes.

The conservation of energy results in the convective heat flux ($q_{convective}$) as a function of the radiative ($q_{radiation}$) and conductive fluxes ($q_{conduction}$):

$$q_{convective} = -q_{radiation} - q_{conduction} \quad (1)$$

The conductive heat flux $q_{conduction}$ is calculated based on the thermal resistance of the wall and the temperature difference between the internal wall surface and the outdoor air. For

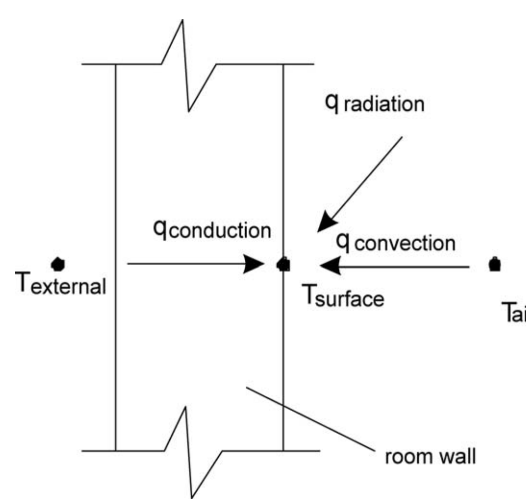

Fig. 2. The energy balance at an internal wall surface used to develop convection correlations.

surfaces with small thermal resistances, the conductive heat flux was directly measured using a system of heat flux meters. Furthermore, the radiative heat flux $q_{\mathrm{radiation}}$ is calculated based on the surrounding wall surface temperatures and view factors using a computer program for building energy and airflow (BEAF) simulations [11]. For this calculation, all of the enclosure surfaces need the long-wave emissivity (ε) and temperatures (T) as input data. To precisely calculate radiative heat fluxes, the enclosure was divided into a large number of smaller sub-surfaces where the temperature of each sub-surface was precisely measured. Knowing the convective heat flux (Eq. (1)), surface (T_{surface}), and air (T_{air}) temperatures, the convection coefficient (h) is calculated as:

$$h = \left| \frac{q_{\mathrm{convective}}}{T_{\mathrm{surface}} - T_{\mathrm{air}}} \right|. \tag{2}$$

Based on Eq. (2), calculations of h need measured temperatures and heat flux data. This way calculated convection coefficient often combines both natural and forced convection effects. For example, the jet velocity at the floor surface is very low, but it still produces effects of forced convection on a large portion of the floor surface, which is combined with natural convection effects created by surface-air temperature difference.

In our study, the correlation for floor surfaces is developed as a function of supply volume airflow rate, normalized by room volume. Spitler et al. [12] provided justification for this approach. Their study has shown that the heat transfer coefficient is relatively insensitive to supply jet velocity and supply jet momentum. Furthermore, Fisher and Pedersen [7] suggested that convection correlations require a physical understanding in terms of the room control volume, rather than in terms of the surface boundary layer. Therefore, for the floor surface, h is given as a function of the room number of air changes per hour (ACH).

A simplified relation between h and ACH can be obtained by considering general relationships between Nusselt (Nu), Prandtl (Pr), and Reynolds numbers (Re). These relations define forced convection along a plate [13]:

$$\text{for laminar flow}: \; Nu = C_{\mathrm{L}}\, Re^{1/2}\, Pr^{1/3}, \tag{3}$$

$$\text{for turbulent flow}: \; Nu \approx C_{\mathrm{T}}\, Re^{4/5}\, Pr^{0.43}, \tag{4}$$

Nusselt and Reynolds numbers can be expressed as functions of room volume (V_{room}) and supply volume airflow rate (V_{supply}):

$$Nu = \frac{h V_{\mathrm{room}}^{1/3}}{k_{\mathrm{air}}} \tag{5}$$

$$Re = \frac{V_{\mathrm{supply}}}{\upsilon_{\mathrm{air}}\, V_{\mathrm{room}}^{1/3}} \tag{6}$$

Substituting Eqs. (5) and (6) into expressions (3) and (4) and substituting values for constant Prandtl number (Pr), air conductivity (k_{air}), and dynamic viscosity (υ_{air}), the following expressions are obtained for laminar and turbulent flows:

$$h_{\mathrm{forced\_laminar}} = c_{\mathrm{L}} \cdot \mathrm{ACH}^{1/2} \tag{7}$$

$$h_{\mathrm{forced\_turbulent}} = c_{\mathrm{T}} \cdot V_{\mathrm{room}}^{1/5} \cdot \mathrm{ACH}^{4/5} \tag{8}$$

The room volume term ($V_{\mathrm{room}}^{1/5}$) in Eq. (8) is usually neglected, so the forced convection at a flat room surface is a function of volume flow rate [7]:

$$h_{\mathrm{forced}} = c \cdot \mathrm{ACH}^{m} \tag{9}$$

There are a large number of previously developed convection correlations for natural convection in a room. Therefore, the intention of this study is to identify an appropriate existing correlation for wall surfaces in a room with displacement ventilation. Alamdari and Hammond [5], Awbi and Hatton [4], and Min et al. [6] developed natural convection correlations typically used in building design and research practice. All these correlations express natural convection as a function of temperature difference between the wall surface and air ($\Delta T = T_{\mathrm{surface}} - T_{\mathrm{air}}$). In addition, some correlations use a characteristic length scale, such as height of the vertical surfaces or hydraulic diameter for horizontal surfaces.

In this study, the local air temperature ($T_{\mathrm{air\_local}}$) is used. It is defined as the average temperature of the air layer close to the surface. Karman's correlation for the ratio between boundary layer thickness and characteristic length [14] shows that for typical room dimensions and temperature differences (ΔT) boundary layer thickness is almost always <0.1 m. Even with a very small temperature difference, such as $\Delta T = 0.3$ °C, and a large characteristic length, 5 m, the thickness is <0.1 m. Therefore the distance of 0.1 m ensures that local air is defined as air that is close to the surface but outside of the boundary layer.

To combine the effects of natural and forced convection at floor surfaces in the room with displacement ventilation, the present study used the Churchill and Usagi [15] method originally proposed for interpolation between limiting solutions of two independent variables. With this method, the convective coefficient combines forced (h_{forced}) and natural (h_{natural}) convection in the following way:

$$h_{\mathrm{combined}} = (h_{\mathrm{natural}}^{n} + h_{\mathrm{forced}}^{n})^{1/n}. \tag{10}$$

Eq. (10) enables the larger term to take over the final value of h_{combined} and in this way to represent the dominant convection phenomenon. The coefficient n is an arbitrary constant that defines the degree at which the final value of h_{combined} reflects the dominant term. For example, Fig. 3 shows how natural and forced convection can be combined for $n = 2$, 3, and 6. The appropriate value of n varies based on the phenomena that are combined and can be obtained from experimental results.

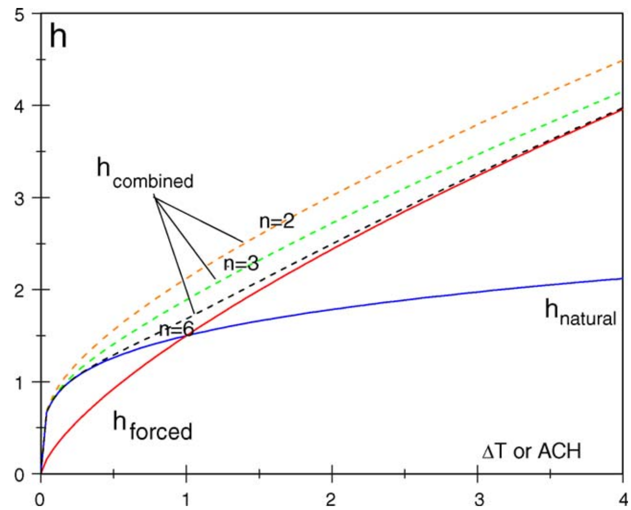

Fig. 3. Graphical interpretations for combined effects of forced and natural convection.

3. Experimental facility used for convection correlation development

The experiments were conducted at the building environmental simulation and testing facility at The Pennsylvania State University. This facility is a state-of-the-art installation for research related to energy, airflow, thermal comfort, and air quality in buildings. Fig. 4 represents this facility with two adjacent chambers. Each of the chambers has an individual heating, ventilating, and air-conditioning (HVAC) system for air handling, and the environmental chamber also has a hydronic surface cooling system. To insulate the facility from external thermal influences, the chamber walls are built from insulating material that provides a conduction resistance of $R = 5.3$ (m^2 K)/W. An important part of this facility is the sophisticated data acquisition system used for measurements of energy and airflow parameters, such as surface heat fluxes, surface and air temperatures, and air velocities in different parts of the facility.

Both the environmental and climate chamber had displacement ventilation diffusers (Fig. 4). The climate chamber tests provided data for the convection correlation development at floor surfaces, while the experiments in the environmental chamber enabled the validation of existing correlations for natural convection with and without DV diffusers. The size of the climate chamber is 2.5 m × 3.9 m × 2.7 m. In this chamber, the heat sources were low temperature heating panels positioned at the floor, and the DV system provided cooling as shown in Fig. 4(a). The dimensions of the environmental chamber are 6.0 m × 3.9 m × 2.4 m. The heat sources were also low temperature heating panels positioned at the floor and wall surfaces (Fig. 4(b)). In this chamber, cooling was delivered by DV or by cooling panels positioned at the ceiling.

To accurately calculate the radiative heat fluxes at different surfaces using conservation of energy, the envelope of the climate chamber was divided into 21 sub-surfaces. Each surface had attached thermistor sensors, which measure surface temperature with an accuracy of ±0.1 °C. The number of sensors positioned on a surface depended on the importance of the surface for the overall heat flow in the chamber. To account for the uneven floor surface temperature, the floor in the climate chamber had 8 sub-surfaces with 10 attached thermistors. An additional eight thermistors were propped 0.1 m above the floor surface sensors to measure local air temperatures. Aluminum tin foil shielded the thermistors from radiative heat exchange. Supply and exhaust air temperature measurements also used thermistors. A system of flow stations measured the supply volume airflow rate with an accuracy of ±5%. Electric heating panels, regulated by a transformer, covered the floor of the climate chamber. The overall accuracy of the total heat flux measurements at the electric panels was ±2.5%.

The environmental chamber had 38 characteristic sub-surfaces with the total of 48 surface thermistor sensors (accuracy of ±0.1 °C). An additional 38 thermistors obtained air temperatures 0.1 m from all of these surfaces. Further away from the surfaces, 28 air temperature sensors, 24 RTD and 4 thermistor sensors with accuracies of ±0.2

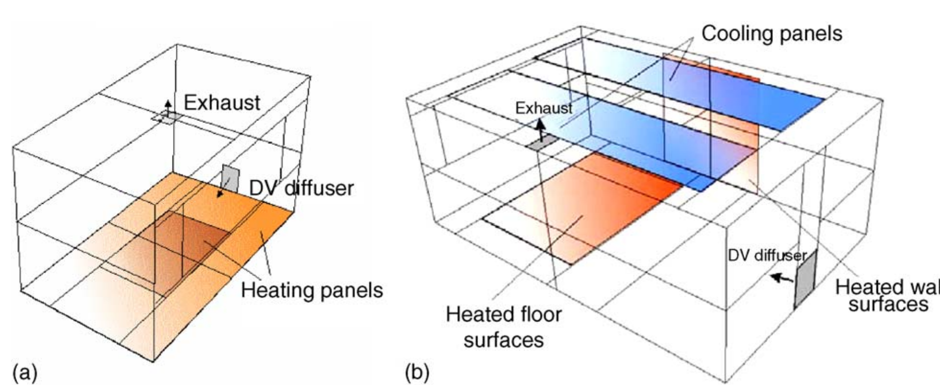

Fig. 4. Schematics of building environmental simulation and testing facility chambers: (a) climate chamber and (b) environmental chamber.

and ±0.1 °C, respectively, collected the room air temperatures. Similar to the climate chamber measurements, the number of sensors positioned in the vicinity of the surfaces depended on the importance of the surface for the overall heat flow in the chamber. Besides, surface and air temperature measurements in the environmental chamber, 24 air velocities in the vicinity of the surfaces were also measured. These velocities revealed whether the convective regime at a certain surface was predominantly forced or buoyant. Similar to measurements in the climate chamber, supply and exhaust temperatures, volume flow rate, and total heat flux at the heating and cooling panels were monitored.

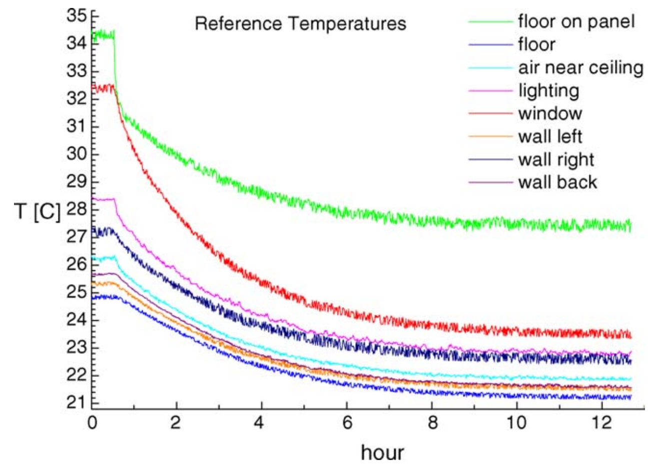

Fig. 5. An example of temperature recording for determination of steady-state conditions in the environmental chamber for natural convection validation.

4. Experimental procedure for convection correlation development

The climate and environmental chambers provided data for two main tasks:

- development of convection correlation for floor surfaces in the climate chamber;
- validation of existing convection correlations in the environmental chamber.

The climate chamber was used for the correlation development because its smaller size enabled precise measurement of heat fluxes along the entire floor surface. The experiments in the environmental chamber enabled evaluation of natural convection correlations by way of the installed hydronic cooling panel system. Table 1 presents the total number of conducted experiments and additional specifications for the two experiment types based on the purpose of the collected data.

In the climate chamber experiments, the convective heat fluxes at the floor were measured at different supply volume airflow rates. The variation of volume flow rate was in the range of 2.5–9.9 ACH in the room. In addition, the experiments used three different power adjustments for heating panels at the floor surfaces, which provided an approximate floor convective heat flux of 7, 15, and 40 W/m^2. The floor heating panels were regulated to provide both, uniform total heat flux on the whole floor, and non-uniform total heat flux by using only the heating panels in the central part of the floor (see Fig. 4(a)).

For the environmental chamber experiments, validation of existing natural convection correlations was conducted with the DV system either ‘‘on’’ or ‘‘off’’. Heating panels created appropriate temperature differences and convective heat flu-xes at the wall surfaces. In the experimental cases, when the DV system was ‘‘on,’’ this system removed the heat gains. In the cases when the DV system was ‘‘off,’’ there was no supply air and the ceiling cooling panels worked as heat sinks (Fig. 4(b)). In experiments related to the validation of existing natural convection correlations at floor surfaces, the ventilation system was ‘‘off’’. In this case, the ceiling cooling panels extracted the entire cooling load. Power variation of the floor heating panels created different temperature gradients at the floor surface for different experiments. For all of the experiments conducted in the environmental chamber, local velocities were measured to calculate Gr/Re^2 ratio. This ratio determined the surfaces with dominant forced convection, where $Gr/Re^2 < 1$, or buoyant convection, where $Gr/Re^2 > 1$ [16].

To ensure the accuracy of the measured parameters for the calculation of convective heat fluxes, measurements were conducted for steady-state airflow and conductive heat flow in different elements of the chamber's structure. For each experiment:

- controlled parameters, such as the air supply temperature and volume flow rate and/or water flow and temperature for radiant panels were adjusted to a set point;
- surface and air temperatures for 32 reference points were recorded every 50 s (as shown in Fig. 5) until a steady-state temperature distribution was attained;

Table 1
Types, number, and specification of experiments used in data analysis

Type of experiments	Experiment specifications	Total number	Chamber	Area of heat source	Heat sink(s)
Development of convection correlation for floor surfaces	Forced convection correlation development	10	Climate	Entire floor	DV system
	New correlation testing for floor heat patches	3	Climate	Local floor	DV system
Validation of existing convection correlations	Measurements of convection at walls	3	Environmental	Wall	DV system
	Measurements of natural convection at walls	5	Environmental	Wall	Cooling panels
	Measurements of natural convection at floor	7	Environmental	Local floor	Cooling panels

- values for steady-state temperature and velocity at all installed sensors were recorded for 2 min and then averaged.

To test the validity of measurements, an energy balance check was conducted for each experiment by comparing the heat gains with the energy extracted by the DV ventilation system or ceiling cooling panels. The energy balance showed that in all of the experiments, the difference between the heat gain and extraction was <6%. This small difference proves that steady-state was reached and that the experiments were conducted under well controlled conditions.

5. Results and discussions

The experimental part of this study took place over a course of several months. The total number of conducted experiments is 28, as presented in Table 1. The experimental results are the base for the development of new floor convection correlations and validation of the existing correlations for natural convection at the vertical and floor surfaces.

5.1. Convection correlation development for floor surfaces

Experiments with displacement ventilation show that the major convective heat transfer occurs at the floor surfaces. In those experiments, the measured convective heat flux at the floor surface was from 51 to 82% of the total convective surface heat flux. The smaller percentage values are for experimental cases where the heating panels were at wall surfaces, while the larger values are for cases where the heating panels were at the floor surface (Table 1). This large portion of convective heat transfer at the floor suggests that the cool air supplied at the floor level by DV diffusers significantly cools down the floor surface, which represents a radiative sink for the other room surfaces. Considering this phenomenon, special attention is dedicated to the development of accurate convection correlations for the floor surface with DV.

Table 2
Variation of local values for *Gr, Re*, and *Nu* numbers with floor position and airflow

	Volume flow rate			
	3.4 ACH		6.1 ACH	
	1.4 m[a]	3.2 m[a]	1.4 m[a]	3.2 m[a]
Gr	1.71×10^{10}	1.59×10^{10}	1.95×10^{10}	1.83×10^{10}
Re	1.79×10^{4}	1.39×10^{4}	3.38×10^{4}	2.79×10^{4}
Nu	428	271	477	367

[a] Distance from diffuser.

Vertical temperature stratification is a well-known phenomenon with DV. However, in addition to this stratification, there is also a non-uniform horizontal air temperature distribution in the vicinity of the floor. In our climate chamber experiments, this horizontal stratification created a considerable floor surface temperature variation. Fig. 6 shows the influence of a non-uniform vertical temperature distribution on local convection coefficients for one test set-up. Fig. 6(a) shows that the surface temperature in the vicinity of the diffuser was lower than the temperature further away. On the other hand, the variation of temperature difference between the surface and local air is relatively small (Fig. 6(b)). However, non-uniform surface temperatures created non-uniform convective heat fluxes due to non-uniform radiative heat exchange with other surfaces. As a result, the variation of local convection coefficients was considerably large (Fig. 6(c)). Other test cases provided similar results. Because of the non-uniform floor surface temperature and non-uniform temperature distribution of the air layer above the floor, the convection correlations were developed for average surface and average local air temperatures. These averaged temperatures enable practical use of the newly developed correlation.

Variation of the local values for temperature difference, velocity, and convection coefficients resulted in the variation of local values for *Gr, Re*, and *Nu* numbers. Table 2 presents the impact of local flow by presenting the variation of these

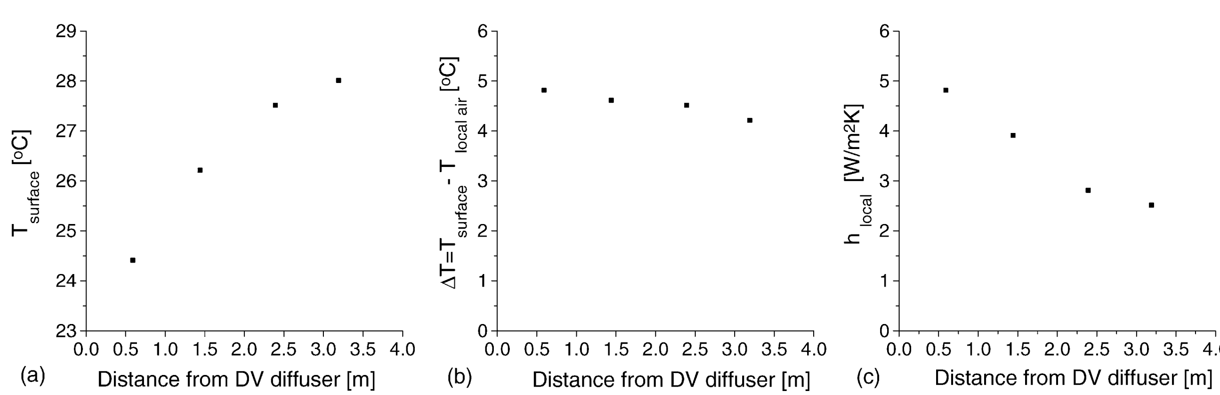

Fig. 6. Variation of temperatures and local convection coefficients with horizontal distance from the DV diffuser. The measurements were at the diffuser centerline and 0.1 m above the floor. The supply temperature was 17.9 °C and volume airflow rate provided 4.6 ACH in the climate chamber: (a) surface temperature distribution; (b) local temperature differences; (c) local convection coefficients.

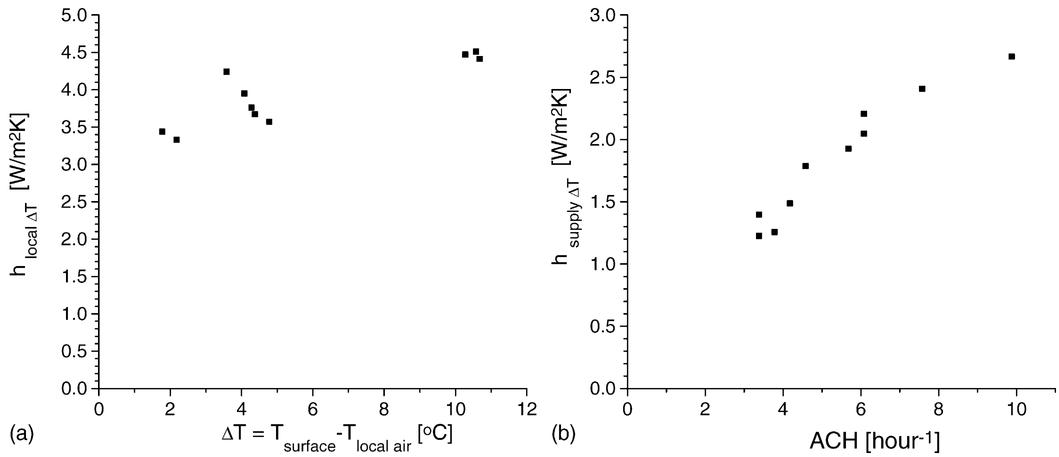

Fig. 7. Measured convection coefficients (h) for the floor with displacement ventilation as a function of a local temperature difference (ΔT) and the supply flow rate (ACH): (a) h as a function of ΔT and (b) h as a function of ACH.

numbers with floor position for two different volume flow rates. Variation of the local distribution of the Gr number with distance from the diffuser was not large, since the distribution of local temperature difference was relatively uniform (refer to Fig. 6(b)). On the other side, local distribution of local velocities resulted in considerable variation of locally defined Re numbers. The change of these locally defined Re numbers with supply volume flow rate was even larger. The change of the flow rate from 3.4 to 6.1 ACH doubled the values for local velocities and corresponding Re numbers (Table 2). The local distribution of h resulted in a large variation of locally defined Nu numbers with horizontal distances from DV. Also, the change of ACH affects the Nu number since velocity (Re number) increases and affect convective heat transfer.

Local air temperatures measured 0.1 m above the floor ($T_{\text{local\_air}}$) and the supply air temperature (T_{supply}) were the two reference temperatures used for the experimental data analysis. When the reference temperature is the supply air temperature, the convective heat flux is calculated as $q_{\text{surface}} = h_{\text{supply}}\ [T_{\text{surface}} - T_{\text{supply}}]$ and the convection coefficient (h_{supply}) is a function of the supply volume airflow rate expressed in ACH. Fig. 7 presents the measurement results as a function of a local temperature difference ($\Delta T = |T_{\text{surface}} - T_{\text{air}}|$) and ACH. These results indicate that the convection correlation expressed as a function of volume flow rate is stronger than the correlation given as a function of a temperature difference of the local air and floor surface. Therefore, the forced convection correlation at a floor surface with the displacement ventilation system has the form of Eq. (9).

For measured velocities in the vicinity of the floor (0.07–0.25 m/s) and a room floor hydraulic diameter of $D_{\text{h}} = 3$ m, the local Re number for flow near to the floor surface was in the range of 10^4 to 5×10^4. This range of local Re numbers indicates a laminar flow regime [13]. Therefore, the exponent m, in general, Eq. (9) should be 0.5 as indicated in Eq. (8). However, experimental results and function fitting of general Eq. (9) show that the exponent m in Eq. (9) has a value close to 0.8 (Fig. 8). This value for coefficient m corresponds to the exponent coefficient for turbulent flow (Eq. (8)). The exponent value of $m = 0.8$ was also obtained in research studies conducted by Fisher [17], and Fisher and Pedersen [7]. They conducted experiments with ceiling diffusers and found that $m = 0.8$ fits the best for forced convection at all surfaces (ceiling, walls, and floor), even though, the Reynolds number at floor surfaces was rather small ($<5 \times 10^4$). A possible reason for this turbulent flow behavior for Re numbers $<5 \times 10^4$ is that the properties of room airflow are different from the flows on free surfaces due to the space confinement.

Using function-fitting and the experimental results presented in Fig. 8, the coefficient c for the forced convection correlation is $m = 0.8$, so Eq. (9) becomes:

$$h_{\text{force}} = 0.48 \cdot \text{ACH}^{0.8}. \tag{11}$$

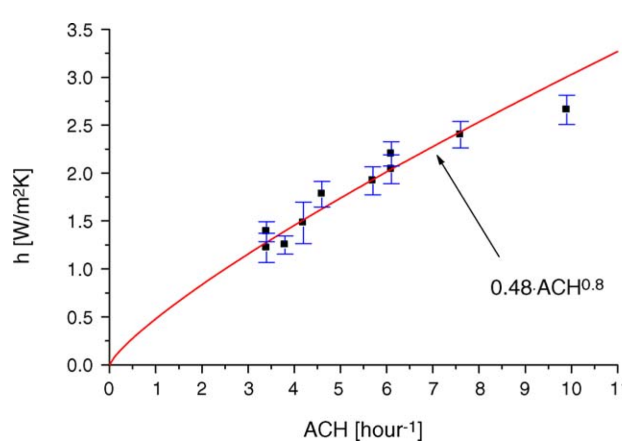

Fig. 8. Experimental data with uncertainties compared to the new equation for the convection correlation for floor surfaces with DV diffuser (exponent, $m = 0.8$).

This correlation is based on the supply air temperature and needs to be modified for use with the local air temperature (or the room average temperature):

$$h_{c\_local} = \frac{|T_{surface} - T_{supply}|}{\Delta T} 0.48 \cdot ACH^{0.8} \quad (12)$$

Eq. (12) should be used carefully because at some surfaces the local air and surface temperature difference ΔT is very small and close to zero. In these cases, h_{c_local} takes unrealistic values. This is, especially, the case, when this equation is used with automatic iterations, such as in energy simulation or computational fluid dynamics (CFD) programs [11,18]. To avoid this division problem resulting in unrealistically high h_{c_local}, the simulation programs should include restrictions for the denominator ΔT. When $\Delta T < \varepsilon$, the term $|T_{surface} - T_{supply}|/\varepsilon$ should substitute for the term $|T_{surface} - T_{supply}|/\Delta T$, where ε is any temperature difference at which the convective heat flux is small.

5.2. Convection correlation for floor surfaces with heat patches

The floor convection correlation, given by Eq. (11), is based on measurements, where the entire floor area has a uniform heat flux. In reality, some parts of the floor may have larger temperatures and convective heat fluxes than the rest of the floor. For example, surfaces heated by direct solar radiation (sun patches) or surfaces heated by local lighting system have a considerably higher temperature than the surfaces in the vicinity of the displacement diffuser. To test the new floor convection correlations, additional experiments with heat patches were conducted. In these experiments, a part of the floor was releasing a relatively larger heat flux (convection portion ~38 W/m^2), while the heat flux at other parts of floor was negligible. Fig. 9 presents the experimental results for floors with heat patches.

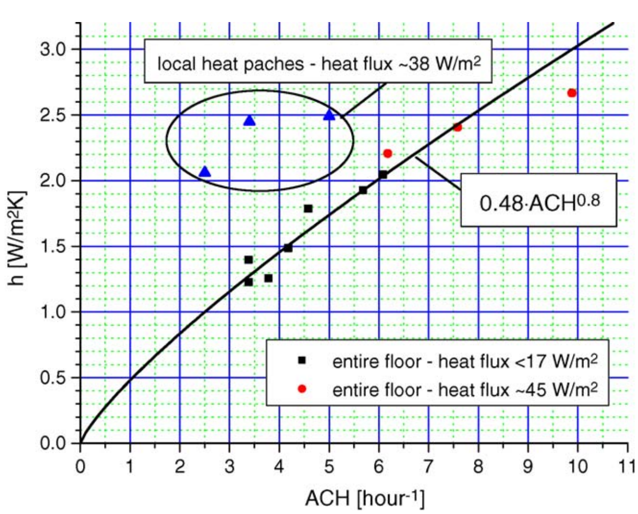

Fig. 9. Forced convection correlations for the floor with a DV diffuser including the convection coefficients measured at the heat patch floor area.

Based on Fig. 9, the new correlation for forced convection (Eq. (11)) cannot predict convective heat transfer for local heat patches, where buoyant airflow is predominant. The measurements show that Eq. (11) under-predicts the convective heat flux at heat patches from 30 to 50%. To account for the buoyant convective effect of local heat patches, Eq. (12) is combined with natural convection correlations using the Churchill and Usagi [15] method (Eq. (10)).

To find the appropriate natural convection correlations for floors with heat patches, existing convection correlations were tested. Several measurements of convection coefficients were conducted at floor surfaces in the environmental chamber with no air supply and $T_{floor} > T_{air}$. Fig. 10 shows the comparison of these measured convection coefficients with previously developed convection correlations [4–6]. The comparison results are for a characteristic length of 2.6 m, which is the hydraulic diameter of the heating panels at the floor (see Fig. 4(b)). As Fig. 10 shows, the correlation developed by Awbi and Hatton agrees well with the new experimental results. Consequently, this correlation is selected for the floor area with heat patches, where the natural convection is the dominant phenomenon.

Eq. (10), with the exponent coefficient $n = 6$, combines effects of forced and natural convection. Neiswanger et al. [19] established that for Rayleigh numbers (Ra) $>10^{11}$, the ideal value of n is 3.2. However, in present experiments with displacement ventilation, Ra was below this threshold. The value of $n = 6$ was chosen based on good agreement with experimental data. The general form of the convection correlations for floor surfaces with $T_{floor} > T_{air}$ in rooms with DV diffusers is:

$$h_{combined} = \left[\left(\frac{2.175 \cdot \Delta T^{0.308}}{D_h^{0.076}}\right)^6 + \left(\frac{\Delta T}{|T_s - T_{supply}|} 0.48 \cdot ACH^{0.8}\right)^6\right]^{1/6} \quad (13)$$

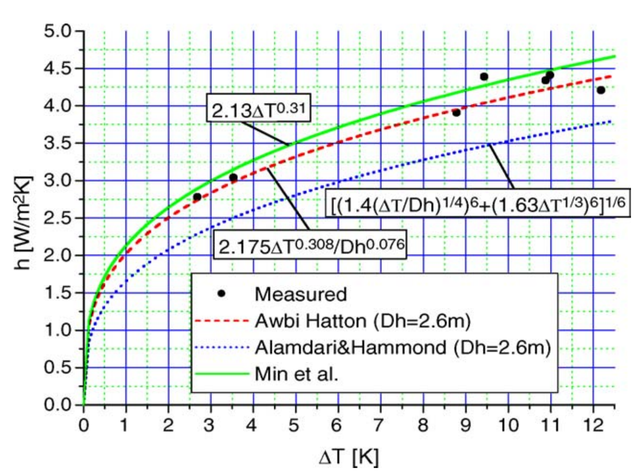

Fig. 10. Performance of the existing natural convection correlations and measured data for floor surfaces, where $T_{floor} > T_{air}$.

Analysis of the overall results shows that the largest difference between measured and predicted heat flux (by Eq. (13)) is <19%, including heat patches. Therefore, Eq. (13) is appropriate for the estimate of the convective heat transfer from warm floor surface ($T_{floor} > T_{air}$).

5.3. Validation of existing natural convection correlations

To account for the convective heat transfer at the other room surfaces besides the floor, existing convection correlations may be appropriate if a rigorous validation process justify their use. Due to the low air velocities in rooms with DV, the assumption is that the airflow at room surfaces is driven by the temperature difference between the local air and surface. To confirm this assumption, DV was "on" and "off" in tests conducted in environmental chamber. In both types of test, the Grashof (*Gr*) and Reynolds (*Re*) numbers were calculated in the vicinity of the surfaces to obtain the ratio Gr/Re^2. In all experiments, with and without DV, the ratio Gr/Re^2 for vertical surfaces was considerably above 1. This ratio indicated that natural convection is the dominant heat transfer phenomenon at vertical wall surfaces in rooms with a DV diffuser. Therefore, additional test measurements were conducted to determine which of the three previously developed natural convection correlations: (1) Alamdari and Hammond [5], (2) Awbi and Hatton [4], or (3) Min et al. [6], was the most appropriate for application at vertical surfaces in rooms with DV. Fig. 10 shows the result of these tests. In all of the correlation equations, the height of the chamber walls (2.35 m) was the characteristic length.

Fig. 11 show that correlations developed by Awbi and Hatton [4] have better agreement with measured results than correlations developed by Alamdari and Hammond [5] and Min et al. [6]. The most likely reasons for these results are the experimental set-ups used in the previous and current studies. Awbi and Hatton's natural convection correlations were developed in a similar environmental chamber to the one used in the present study, while the Alamdari and Hammond natural convection correlations were created based on a series of experiments that were primarily conducted with isolated surfaces. The correlations developed by Min et al. were also developed using a full-scale testing room, relatively similar to our environmental chamber. The slightly better performance of the Awbi and Hatton's correlations than Min et al.'s correlation is probably due to the definition of the air reference temperature. The Awbi and Hatton's correlations use local air temperature at 0.1 m from the wall surface as the reference temperature, which is the same reference temperature used in our experiments presented in Fig. 11. On the other hand, Min et al. defined the reference temperature as the temperature in the central part of the room (1.5 m above the floor). Considering the validation results, Awbi and Hatton's correlations are recommended for the calculation of natural convection in rooms with DV. Table 3 presents Awbi and Hatton's correlations for different surface types and flow regimes.

For wall surfaces, the temperature stratification in the air could lead to heat transfer from the lower part of the wall to the air and from the air to the upper part of the wall. Fig. 12 shows the result of experiments with this phenomenon on the left wall that represents an internal wall in a room. In situation like this, use of the equations from Table 3 with average values for surface and air temperatures (ΔT) can create a certain error in the convection coefficient (h) calculation. In the example presented in Fig. 12, the difference between the measured and calculated h was >40%. However, total heat flux at surfaces like this is small (19 W in the presented example) and has small effect on overall energy flow in rooms. It is much more important that

Table 3
Correlations for natural convection developed by Awbi and Hatton [4]

Surface and regime	Convection correlation
Floor when $T_s > T_{air}$ or ceiling when $T_s < T_{air}$	$2.175 \cdot \Delta T^{0.308}/D_h^{0.076}$
Ceiling when $T_s > T_{air}$ or floor when $T_s < T_{air}$	$0.704 \cdot \Delta T^{0.133}/D_h^{0.601}$
Walls	$1.823 \cdot \Delta T^{0.293}/D_h^{0.121}$

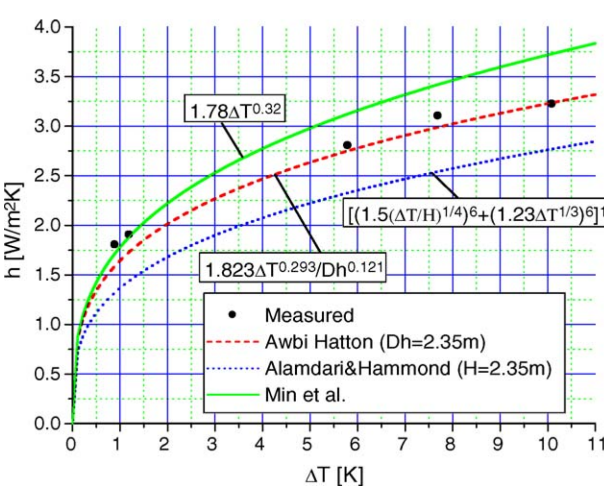

Fig. 11. Validation of the existing natural convection correlations with the measured data for vertical wall surfaces in a room with DV.

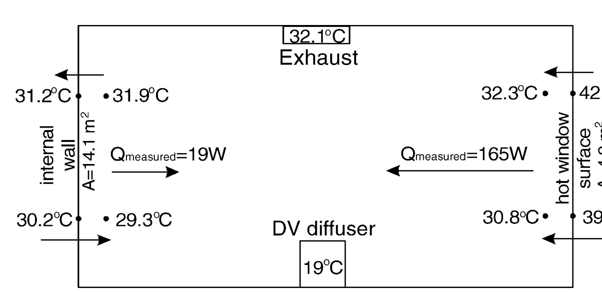

Fig. 12. Measured temperature and heat fluxes in the experimental facility with the hot window surface and the internal wall on the opposite side.

the convection correlation predicts precisely the heat flux at surfaces with large ΔT, which create large total heat fluxes. At surfaces with a large ΔT heat fluxes have the same direction in the lower and upper part of the surface and the equation from Table 3 is appropriate. In the experiment presented in Fig. 12, the right wall surface represents a hot external window surface. For this surface, the measured flux is 165 W while the equation from Table 3 calculates 150 W.

For ceiling surfaces, the temperature stratification with DV often causes higher local air temperatures than ceiling surface temperatures as shown in Fig. 1. At these surfaces, the convective heat transfer is similar to the one with cooled ceiling panels. Therefore, the correlation for cooling ceiling (CC) panels [11] is recommended at these surfaces when $T_s < T_{air}$. The correlation for cooled ceiling has the following form:

$$h_{\text{cooled\_ceiling}} = 2.12 \cdot \Delta T^{0.33}. \tag{14}$$

When the ceiling surface has only a slightly lower temperature than the local air, Eq. (14) gives similar h values as the Awbi and Hatton's correlation for ceilings, where $T_s < T_{air}$. Therefore, for these surfaces, both correlations are appropriate. In the case when $T_s > T_{air}$, such as surfaces close to lamps or ceiling surfaces in rooms with large solar heat gains, Table 1 gives the appropriate Awbi and Hatton's correlation.

5.4. Recommended convection correlations for all envelope surfaces in a room with DV

Table 4 summarizes the recommended convection correlations for all different surfaces in a room with displacement ventilation. The recommended correlations include newly developed correlations and the correlations developed by Awbi and Hatton [4] and Novoselac [11]. All of these correlations use the local air temperatures defined as the air temperature in the surface vicinity (0.1 m from the surface). Models developed by Mundt [2], Rees and Haves [3], and boundary condition models used in CFD programs [11,18] can calculate temperature distribution and local air temperatures, which then can be used for the calculations of h. These correlations are primarily for use in DV models that calculate temperature stratification in rooms for thermal comfort and air quality evaluations. In addition, models that present the room air temperature as a single node can also use the developed correlations. For example, the floor convection correlation presented in the form of Eq. (11) is appropriate for energy simulation models or standard design procedures that are based on the assumption of the uniform air temperature. For this purpose, Eq. (11) uses supply air temperature as the reference temperature and does not need an air temperature distribution.

Table 4
Recommended convection correlations h for a room with displacement ventilation

Surface	Regime	Convection correlation
Floor	$T_s > T_{air}$	$\left[\left(\frac{2.175\cdot\Delta T^{0.308}}{D_h^{0.076}}\right)^6 + \left(\frac{\lvert T_s - T_{supply}\rvert}{\Delta T}\cdot 0.48\cdot \mathrm{ACH}^{0.8}\right)^6\right]^{1/6}$
	$T_s < T_{air}$	$\left[\left(\frac{0.704\cdot\Delta T^{0.133}}{D_h^{0.601}}\right)^6 + \left(\frac{\lvert T_s - T_{supply}\rvert}{\Delta T}\cdot 0.48\cdot \mathrm{ACH}^{0.8}\right)^6\right]^{1/6}$
Ceiling	$T_s > T_{air}$	$0.704\cdot\Delta T^{0.133}/D_h^{0.601}$
	CC panel	$2.12\cdot\Delta T^{0.33}$
	$T < T$	$2.175\cdot\Delta T^{0.308}/D_h^{0.076}$
Walls		$1.823\cdot\Delta T^{0.293}/D_h^{0.121}$

6. Conclusions

This paper presented the development of new and validation of existing convection correlations for rooms with displacement ventilation. These correlations were developed and tested using a state-of-the-art experimental facility that enabled measurements in environments representing rooms in office buildings. Besides the recommended convection correlations presented in Table 4, the measured data and their analyses also pointed out the importance of convection correlation for floor surfaces.

The major heat transfer from room surfaces to the air appears at the floor surface. Consequently, a precise calculation of convective heat fluxes at the floor is crucial for accurate predictions of energy consumption, air quality, or thermal comfort in a room with displacement ventilation. The major parameters that affect heat flux at the floor surface are supply air temperature, volume flow rate, and local air temperature. Generally, the correlation based on normalized volume flow rate (ACH) that uses supply temperature as a reference temperature is stronger than the correlation based on a temperature difference between the surface and local air (ΔT_{local}). Modeling the forced convection at room surfaces as a function of ACH enables the development of general and practical convection correlations that are based on parameters readily available in design or simulation procedures. To take into account the buoyant effect of heat patches at the floor, the influence of ΔT_{local} should be considered.

The non-uniform horizontal temperature distribution created by displacement ventilation diffusers creates variations in floor surface temperatures. This horizontal temperature gradient causes a change in temperature differences between the local air and floor surfaces, which results in considerable variation of the local surface convective heat fluxes. Nevertheless, this variable heat flux can be successfully averaged and modeled based on supply air parameters, such as the air supply temperature and the supply volume flow rate. At the wall and ceiling surfaces, the convective heat flow is primarily driven by natural convection. The validation experiments of the three commonly applied convection correlation for natural convection [4–6] show that the correlations developed by Awbi and Hatton [4] are the most suitable for application in a standard office room with DV diffusers.

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