New Convection Correlations for Cooled Ceiling Panels in Room with Mixed and Stratified Airflow

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Received June 8, 2005; accepted December 23, 2005

Models for buildings' surface convective boundary conditions are important components of building design and simulation programs. This is especially true for cooled ceiling (CC) panels, which extract major cooling loads from rooms. This paper introduces new convection correlations for rooms with CC panels, developed from laboratory measurements conducted in a full-scale experimental facility. The new convection correlations were developed for CC panels used with or without a ventilation system. When the ventilation system was used, the air was supplied through a high aspiration diffuser. High aspiration diffusers, recommended for use with CC systems, create a specific airflow pattern characterized by narrow and directional jets with a large entrainment of surrounding air. To account for the forced airflow and local buoyancy effects, new convection correlations for CC surfaces were developed as functions of temperature difference and normalized volume flow rate, i.e., the number of air changes per hour (ACH) in the room. Results show that the high aspiration diffuser, even though installed at the ceiling, produces enough air motion in the vicinity of the floor and vertical wall surfaces to affect the convective heat transfer at these surfaces as well. In fact, the effect of forced convection due to the presence of a high aspiration diffuser increases the total convection coefficient at CC surfaces by 4% to 17%.

INTRODUCTION

Cooled ceiling (CC) panels are hydronic cooling systems capable of removing large sensible cooling loads while providing good thermal comfort in rooms. For ventilation purposes, these systems can be combined with displacement ventilation or mixing ventilation systems (Novoselac and Srebric 2002). With ventilation systems that supply only fresh air, such as dedicated outdoor air system (DOASs), high aspiration diffusers can provide good ventilation effectiveness with a relatively small amount of supply air (Mumma 2004). The major roles of the ventilation system are to provide fresh air for the occupants and remove a latent cooling load. The major part of the sensible load is extracted by CC panels, and the room temperature is controlled primarily by regulating the heat flux at the surfaces of the CC panels. Therefore, it is very important to have accurate convection correlations for CC panels both for use in system design procedures and for application in energy simulation programs.

Most of the existing design and simulation models for CC panels use natural convection correlations that were developed based on experimental measurements at (1) isolated surfaces and free edge plates (Alamdari and Hammond 1983) or (2) heated floor surfaces in an enclosure (Min et al. 1956; Awbi and Hatton 1999). In addition, designers and simulation program developers can use mixed convection correlations developed by Chen et al. (1989), Awbi and Hatton

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(2000), and Beausoleil-Morrison (2000) to express the effect of forced air movement, created by ventilation systems, on convective heat transfer. The correlations for natural convection were developed based on measurements at heated floor surfaces. The similarity between boundary layers for a heated floor and a cooled ceiling was used to imply that the equation for heated floors is valid for cooled ceiling surfaces. The analogy between convective heat transfer for cooled ceiling surfaces and for heated floor surfaces is valid in the boundary layers. However, room temperature stratification, combined with the arbitrary reference temperature selection, limits the use of the same convection correlation for calculations of convective heat fluxes at cooled ceiling and heated floor surfaces. Figure 1 shows the effect of stratification on temperature profile in the vicinity of warm floor and cooled ceiling surfaces. In addition, high aspiration diffusers, recommended for use with CC systems (Mumma 2004), create a specific airflow pattern characterized by narrow and directional jets with a large entrainment of surrounding air. The previous research studies do not consider the effects of this type of diffuser on convective heat transfer in a room. The differences of the vertical temperature profiles in the vicinity of the heated floor surfaces and the specific airflow with high aspiration diffusers establish the need for new convection correlations specialized for airflow and temperature distribution characteristics for rooms with CC panels.

The present study has two main objectives. The first objective is to develop convection correlations for the CC surfaces in rooms with only natural convection. The second objective is to quantify the influence of the high aspiration diffuser on convection at cooled ceiling surfaces as well as at other wall and floor surfaces. To accomplish these two objectives, full-scale experiments were conducted in a test chamber with cooled ceiling panels when

1. no air was supplied and air movement was created by buoyant forces only and
2. air was supplied by a high aspiration diffuser that creates large air mixing in the room.

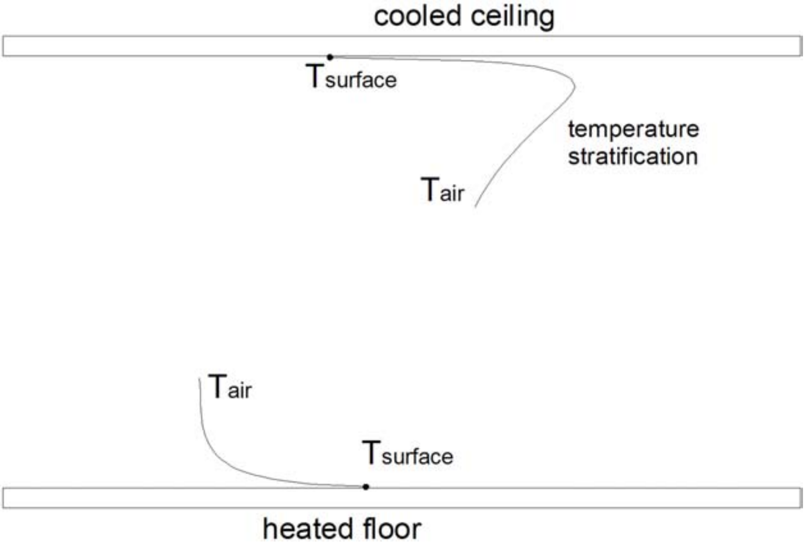

Figure 1. Difference in the vertical air temperature profiles for a cooled ceiling and a heated floor.

EXPERIMENTAL FACILITY USED FOR CONVECTION CORRELATION DEVELOPMENT

The experiments were conducted at the Building Environmental Simulation and Testing (BEST) facility at The Pennsylvania State University. This facility is an installation for research related to energy and airflow, thermal comfort, and air quality in buildings. The facility has two adjacent chambers, environmental and climate, separated by a partition wall with a window. The purpose of the climate chamber was to simulate external weather cooling and heating loads, while the environmental chamber produced typical indoor airflow and temperature distribution in a room with CC panels. 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 system for CC panels. To insulate the facility from external thermal influences, the chamber walls have a conduction resistance of $R = 5.3\ m^2{\cdot}K/W$ ($R = 30\ ft^2{\cdot}h{\cdot}°F/Btu$). All measurements related to the correlation development were conducted in the environmental chamber (Figure 2), which has dimensions of $6.0 \times 3.9 \times 2.4\ m^3$ ($20 \times 13 \times 8\ ft^3$). The sophisticated data acquisition system with its system of probes measured energy and airflow parameters such as surface heat fluxes and surface and air temperatures, as well as air velocities at different locations.

To accurately calculate the radiative heat fluxes at different surfaces in accordance with the conservation of energy, the envelope of the environmental chamber was divided into 38 characteristic subsurfaces. Each surface had attached thermistor sensors, which measured surface temperature with an accuracy of ±0.1°C (±0.18°F). The total number of surface thermistors was 48, and the number of sensors positioned on a surface depended on the importance of the surface for the overall heat flow in the chamber. An additional 38 thermistors collected air temperatures 0.1 m from these surfaces. A total of 28 air temperature sensors, 24 RTD and 4 thermistor sensors, with accuracies of ±0.2°C (±0.36°F) and ±0.1°C (±0.18°F), respectively, collected the room air temperatures farther way from the surfaces. Thermistors also measured the supply and

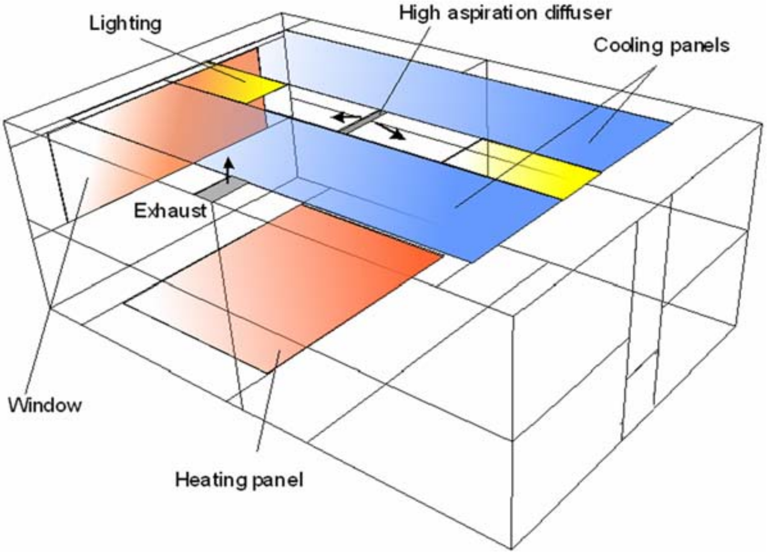

Figure 2. Experimental setup in the environmental chamber used for the development of convection correlations.

exhaust air temperatures and the temperatures around the chambers. Aluminum foil shielded the air thermistors from radiative heat exchange. Overall, the temperature data points were collected at a total of 124 locations. 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 environmental chamber. The overall accuracy of the total heat flux measurements at the electric panels was ±2.5%.

The experimental setup created environmental conditions typical to rooms of modern buildings. Cooling loads were created by (1) heat conduction through the window and the partition wall and (2) internal heat sources (Figure 2). The internal heat sources were lighting, low-temperature heating panels that simulated the heat gains from solar radiation, and convective heat sources that simulated the occupants and indoor equipment. Radiant panels and/or the ventilation system extracted the cooling load, depending on the convection correlations under consideration. A high aspiration diffuser was positioned in the central part of the room between the two radiant panels, as shown in Figure 2. The diffuser supplied the air along the panel's longer sides.

For the development of convective correlations, it is very important either to know the radiative heat flux or to make it negligible. In this study, the radiation from internal objects to the surrounding surfaces was decreased to the negligible level to reduce the error due to imprecise calculation of the convective/radiative heat flux ratio for these objects. The internal objects have nonuniform surface temperatures, and the calculation of radiative heat fluxes from object surfaces to the chamber enclosure would introduce an error. Therefore, in this study, the radiative heat flux from heat sources positioned in the environmental chamber was minimized by use of a device that released energy only by convection. Figure 3a shows the position and design of this convective heat source device. The device contains two light bulbs in a perforated aluminum case placed in a semi-closed box built from thermal insulating material covered by aluminum foil, as shown in Figure 3b. Use of the double casing with insulation and a low-emissivity aluminum cover resulted in the external surface temperatures of this device being similar to the enclosure surface temperatures. The total radiative heat flux from the device was negligible compared to the total convective flux because of the relatively small temperature differences, the small surface area of the device, and the low emissivity of aluminum foil on external surfaces.

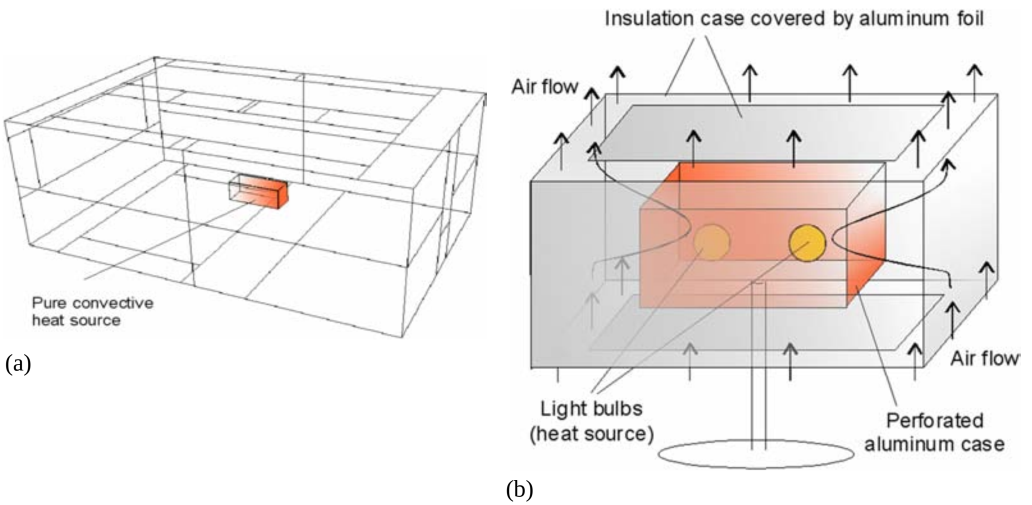

Figure 3. The position (a) and design (b) of the convective heat source device.

METHODOLOGY FOR CONVECTION CORRELATION DEVELOPMENT

The experiments in the full-scale environmental chamber used a methodology based on the conservation of energy at internal room surfaces under steady-state heat flow. This methodology is described in detail by Novoselac et al. (2005), and it is similar to the methodology used by Khalifa and Marshall (1990), Spitler et al. (1991a, 1991b), and Awbi and Hatton (1999). The major difference of methodology applied in this study compared to the one applied by Spitler et al. (1991a, 1991b) is in selection of a reference air temperature. Spitler et al. used air supply temperature as the reference temperature because they analyzed convective heat transfer in rooms with large volume flow rates of supply air. With CC systems the supply airflow rates are lower, and this study uses local air temperature as the reference one. The local air temperatures were obtained based on measurements in an air layer 0.1 m from the surfaces. Awbi and Hatton (1999) used the local temperatures too, and the difference in this study is only in the position of heat sinks in the testing room. Awbi and Hatton (1999) used a cooled side-wall as a heat sink, while in this study heat sinks were cooled ceiling surfaces. Khalifa and Marshall (1990) used different methodology for convective flux calculation. They used low-emissive surfaces and neglected the radiative heat exchange between enclosure surfaces.

This study uses radiative heat fluxes to calculate precise convective heat fluxes at surfaces. Based on the conservation of energy, the convective heat flux for a certain surface was calculated as the sum of measured conductive and calculated radiative heat fluxes ($q_{convective} = -q_{radiative} - q_{conductive}$). Surrounding wall surface temperatures and view factors provided data for radiative heat flux calculations. The enclosure division into a large number of smaller subsurfaces (Figure 3a) and the precise temperature measurements of these subsurfaces further enhanced the radiative heat flux calculations. Knowing the surface ($T_{surface}$) and local air ($T_{air\ local}$) temperatures as well as the convective heat flux ($q_{convective}$), the total convection coefficient (h_{total}) is calculated as

$$h_{total} = \left| \frac{q_{convective}}{T_{surface} - T_{air\ local}} \right| . \tag{1}$$

The experimental data provide the input for the right-hand side of the equation that gives the value of the convective coefficient. To generalize the results, the experiments were repeated under a series of different conditions, and the convective coefficient was expressed in the form of a correlation depending on the convection mechanism, natural or forced. Natural convection correlations use simplified forms as a function of temperature difference between the wall surface and air (ΔT) and a characteristic length scale such as height of the vertical surfaces or hydraulic diameter for the horizontal surfaces. Using the dimensionless analysis, a simplified relationship between h and ΔT can be obtained for different airflow regimes. The relationship between Nusselt numbers (Nu), Prandtl numbers (Pr), and Grashof numbers (Gr) is defined differently for laminar and turbulent flows (ASHRAE 2001):

$$\text{For laminar flow: } \mathrm{Nu} = C_L(\mathrm{Gr} \cdot \mathrm{Pr})^{1/4} \tag{2}$$

$$\text{For turbulent flow: } \mathrm{Nu} = C_T(\mathrm{Gr} \cdot \mathrm{Pr})^{1/3} \tag{3}$$

Substitution of values for the constant Pr number for air and other air constants in the expressions for Nu and Gr numbers provides simplified forms for convection correlations:

$$h_{natural\_laminar} = a(\Delta T/L)^{1/4} \tag{4}$$

$$h_{natural\_turbulent} = b(\Delta T)^{1/3} \tag{5}$$

For forced convection, the present study developed a correlation as a function of air supply volume flow rate, normalized by room volume ($\dot{V}/V$). Spitler et al. (1991b) 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. Also, Fisher and Pedersen (1997) 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, h is expressed as a function of the room number of air changes per hour (ACH = $\dot{V}/V$).

A simplified relation between h and ACH can be obtained by considering general relationships between Nusselt (Nu), Prandtl (Pr), and Reynolds (Re) numbers. Using the room volume as the characteristic length ($V^{1/3}$) and the volume flow rate as the characteristic velocity ($\dot{V}/V^{2/3}$) in the expressions for Nu and Pr numbers and knowing that ACH = $\dot{V}/V$, the forced convection can be presented in a simplified form:

$$h_{forced} \approx c \cdot \mathrm{ACH}^{m} \tag{6}$$

To combine the effects of natural and forced convection at floor surfaces in a room with displacement ventilation, the present study used the Churchill and Usagi (1972) method originally proposed for interpolation between limiting solutions of two independent variables. With this method, the convection coefficient combines forced (h_{forced}) and natural ($h_{natural}$) convection in the following expression:

$$h_{total} = (h_{natural}^{n} + h_{forced}^{n})^{1/n} \tag{7}$$

Equation 7 enables the larger term to take over the final value of h_{total} 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. The appropriate value of n varies based on the phenomena that are combined and can be obtained from experimental results. Incropera and Dewitt (1985) recommended a value of n = 3 for mixed convection and internal surfaces.

EXPERIMENTAL PROCEDURES FOR CONVECTION CORRELATION DEVELOPMENT

The measurements in the environmental chamber provided data for two main tasks:

1. the development of natural convection correlations for CC surfaces and
2. the development of forced convection correlations for rooms with high aspiration diffusers that are combined with the expressions for natural convection.

Table 1 presents the description and number of experiments conducted for the two main tasks. Measurements were conducted for steady-state airflow and conductive heat flow in different elements of the chamber's structure to ensure the accuracy of the measured parameters in both types of experiments. For each measurement, the energy balance was calculated to prove well-controlled conditions.

Measurements of Natural Convection at CC Surfaces

Temperature stratification in the room can significantly affect convective heat transfer (Novoselac et al. 2005). Therefore, in experiments related to natural convection at CC surfaces, the stratification created by different heat sources was analyzed. The temperature gradient near the cooled ceiling as shown in Figure 1 is always present in rooms with natural convection. The level of stratification depends on the types of heat sources in the room. In the case where heat sources are primarily convective and a large portion of the energy from the source is directly transferred to the air in the occupied zone, the temperature gradient in the vicinity of the ceiling is rather larger (1–3°C/m or 0.5–1.6°F/ft). In the case where heat sources are wall surfaces, the room temperature profile is more uniform as indicated in Figure 4. Computers and other office electric equipment are primarily convective heat sources, while solar radiation and lighting systems are primarily surface heat sources.

In experiments with the convective heat source device, the source power was varied to obtain different cooling loads and corresponding temperature differences between the local air and the CC surface (ΔT_{local}). In experiments where surface heat sources were used, the power of the floor panel was varied. In experiments where large cooling loads were needed to create large T_{local}, floor heating panels were combined with the lighting and conductive heat gains through the window and the partition wall (refer to Figure 2). Experiments conducted in this manner created large cooling loads without raising the surface temperature of the floor panels above temperatures typical in buildings. For two experiments, the convective device was combined with surface heat sources to evaluate the combined effects of different heat sources on convective heat transfer at CC panels. The specific cooling load extracted by radiation and convection on the panels' surfaces was in the range of 11–93 W/m^2 (3.5–29.5 Btu/h·ft^2). The convective portions were 51% for experiments with the convective heat source device, 33% for surface heat sources, and 40% for combined heat sources.

Table 1. Types, Number, and Description of Experiments Used in Data Analyses

Type of Experiment	Experiment Description	Heat Source(s)	Number of Experiments	Varied Parameters
Development of natural convection correlation for CC surfaces	– Ventilation OFF – Measurements focused on CC panels	Convective Surface Combination	5 5 2	– Cooling load for convective or surface heat source – CC surface temperature
Development of forced convection correlations for high aspiration diffusers	– Ventilation ON – Measurements at ceiling, walls, and floor at same time	Convective	12	– Cooling load for convective heat source – Volume flow rate –ACH – CC surface temperature

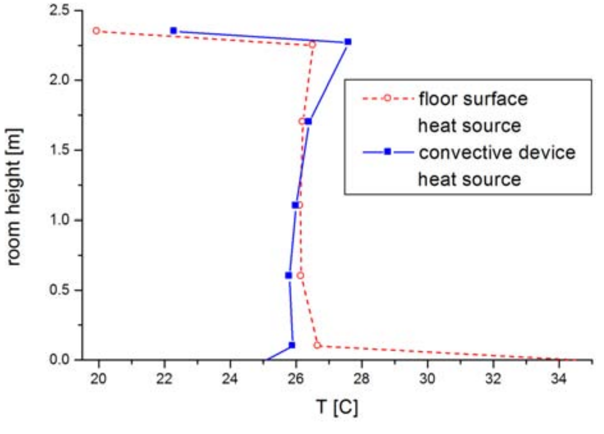

Figure 4. Measured air temperature profiles for convective device and surface heat sources.

Measurements of Forced Convection at Room Surfaces

Experiments with the high aspiration diffuser and the cooled ceiling system were conducted in the environmental chamber using the convective device as the heat source. The supply temperature was kept constant and approximately equal to the room temperature because this temperature had a very small effect on forced convection. This is due to the fact that the jet from a high aspiration diffuser has a large entrainment rate of room air and thus behaves like an isothermal jet. Therefore, the radiation from the ceiling cooling panels cooled the rest of the chamber's envelope, and as a consequence, the room air temperature was higher than the envelope surface temperatures. To develop the convection coefficient correlations, the volume flow rate was varied from 1 to 5 ACH. The supply velocity at diffuser nozzles was approximately 15m/s (3000 fpm). The variation of volume flow rate was achieved by changing the number of diffuser nozzles and keeping the same discharge velocity at nozzles (Figure 5). For each analyzed flow rate, several measurements were conducted varying cooling loads that affected temperature difference between the surface and local air (ΔT_{local}). The local air temperature is defined as the temperature of the air layer 0.1 m from the surface because this distance ensures that the local air is defined as the air close to the surface but outside the boundary layer.

Besides temperatures, local air velocities were measured to calculate Gr/Re^2 ratios in experiments with the high aspiration diffuser. This ratio determined the surfaces with dominant forced convection, where $Gr/Re^2 << 1$, or buoyant convection, where $Gr/Re^2 >> 1$ (Schlichting 1968). For surfaces with very low ΔT_{local}, forced convection was dominant. For these surfaces, forced convective heat flux was equal to the total convective heat flux ($h_{forced} = h_{total}$). At the surfaces where ΔT_{local} was large, such as at CC surfaces, natural and forced convection were present and Equation 7 calculated h_{forced} using the measured h_{total} (Equation 1) and the calculated $h_{natural}$ (Equation 5). These values for h_{forced}, with corresponding ACH, provided data for development of forced correlations in the form of Equation 6 for wall, floor, and ceiling surfaces.

RESULTS AND DISCUSSIONS

The following sections present the analysis of experimental results. The new natural convection correlation for CC panels is presented followed by the analysis of forced convection in

Figure 5. The high aspiration diffuser attached to the ceiling.

rooms with high aspiration diffusers. These results are the base for the development of the combined forced and natural convection correlations in rooms with CC and high aspiration diffusers.

Natural Convection at Cooled Ceiling Surfaces

The airflow in the vicinity of cooled ceiling panels is primarily turbulent. For typical dimensions of cooling panels (hydraulic diameter larger than 1.0 m [3.3 ft]) and temperature differences between the air and panel surface (larger than 2°C [3.6°F]), the Rayleigh number is $Ra > 2 \cdot 10^8$, which indicates turbulent airflow for horizontal surfaces (Kakac 1987). Therefore, the newly developed convection correlations have the form of Equation 5. The experimental results and function-fitting provided coefficient *b* in this equation.

Figure 6 shows the analysis of the measured data used to develop the convection correlations for CC surfaces. Each point in the figure represents a separate experiment. Experiments with the convective device and surface heat sources, conducted to evaluate the influence of the type of heat sources on convective heat transfer at the CC panels, show larger convective heat transfer with the primarily convective heat sources. With these heat sources, there is larger temperature stratification in the vicinity of the cooled ceiling panels created by the buoyant plume from the convective heat sources. This plume hits the ceiling and disperses heated air around the ceiling. This phenomenon creates a larger air movement and intensifies convective heat transfer. The effect of thermal plumes from large surfaces is not so intensive, and, therefore, the convective heat coefficient has smaller values when compared to the coefficient with the convective device. Figure 6 shows the results of the measurements with both the convective device and the surface heat sources. The difference in convective coefficients is around 10%. As expected, the convection coefficients based on experiments that were conducted with both heat sources were between coefficients obtained only with the convective device or the surface heat source. In real buildings, heat sources are always a combination of the convective and surface heat sources. Therefore, the convection correlation for cooled ceiling panels was obtained using all measured data. The final form for the natural convection at CC surfaces is:

$$h_c = 2.12 \cdot \Delta T^{0.33} \quad (\mathrm{W/m^2K}) \qquad [h_c = 0.308 \cdot \Delta T^{0.33} \quad (\mathrm{Btu/h \cdot ft^2 \cdot F})] \tag{8}$$

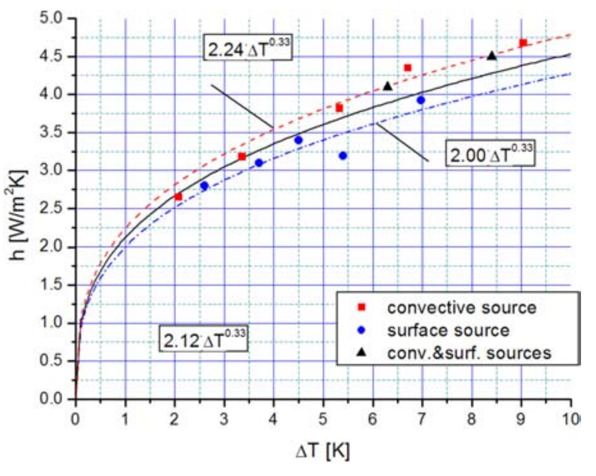

Figure 6. Experimental results and corresponding convection correlations for the CC surface.

The effect of temperature stratification depends on the room airflow. Therefore, this stratification was not detected in experiments where the convection correlations were developed based on the measurements with isolated surfaces and free edge cooled plates (Alamdari and Hammond 1983). Also, this stratification was not taken into account when the analogy between the cooled ceilings and heated floors was used (Awbi and Hatton 1999; Min et al. 1956). Figure 7 shows the comparison of previously developed correlations with the newly developed convection correlation for cooled ceiling panels (Equation 8). Results are compared for the hydraulic diameter of cooling surfaces in our experiments of D_h = 2.0 m (D_h = 6.7 ft). The newly developed correlation predicts considerably larger heat flux than the Alamdari and Hammond (1983) correlation and a slightly larger heat flux than the correlation developed by Awbi and Hatton (1999) or the correlation developed by Min et al. (1956). The newly developed correlation is closer to the correlations of Awbi and Hatton (1999) and Min et al. (1956) than to the Alamdari and Hammond (1983) correlation because the first two correlations are based on measurements in experimental chambers, while the Alamdari and Hammond (1983) correlation is based on the measurements with free isolated surfaces without the influence of confinement. The new correlation calculates 9% larger convection coefficients than the Awbi and Hatton (1999) correlation and 5% larger coefficients than the correlation developed by Min et al. (1956). The most probable reason for this increase is the concentrated convective heat sources that are used for the new correlation development; the two previously developed correlations used only surface sources.

Forced Convection in Rooms with High Aspiration Diffusers

Airflow with a high aspiration diffuser has many local effects. In conducted experiments, the diffuser positioned between the radiant panels supplied air along the panel's longer sides, as shown in Figure 8. The large supply velocity created a narrow jet attached to the ceiling. As a consequence, only a small area of the room ceiling was directly affected by the high jet velocity.

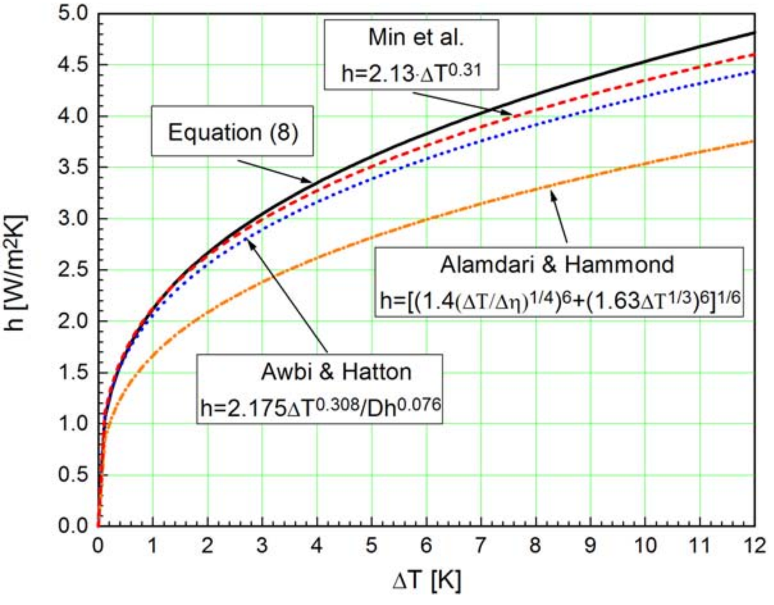

Figure 7. Comparison of newly developed correlation (Equation 8) with other correlations used for the convective heat flux calculations with cooled ceiling surfaces.

The air velocity in the vicinity of the cooling panel surfaces increased with an increase of ACH because the larger jet momentum created a larger jet entrainment. However, the increase of air velocity over the rest of the ceiling, not directly influenced by the jet, was not large. For example, with an increase of ACH from 0 to 5, the measured increase of velocity in the vicinity of the cooling panels was from 0.2 to 0.4 m/s (40 to 80 fpm). For the floor surface, the measured increase of local air velocity was from 0.05 to 0.2 m/s (for ACH from 0 to 5). At zero ACH in the room, the velocity in the vicinity of the cooling panels (0.2 m/s [40 fpm]) was larger than at the floor surfaces (0.05 m/s [10 fpm]) because of the larger heat flux and air movement caused by natural convection at these panels.

At the CC panels, the temperature difference between the surface and local air created a buoyant effect, which was the predominant heat transfer mechanism. This was the case for measurements with ACH = 1, 2, 3, where the temperature stratification ΔT = 5–8°C (ΔT = 9–14.4°F) created buoyant (natural) convection. Therefore, in the ceiling area, the buoyant effect was taken into account using Equation 7 for the combined forced and natural convection. The newly developed natural convection correlation for cooled ceiling surfaces (Equation 8) quantified the effect of natural convection. Knowing the measured h_{total} and ΔT, h_{forced} was calculated by:

$$\begin{aligned} &(h_{forced\_CC} = [h_{total}^3 - (2.12 \cdot \Delta T^{0.33})^3]^{1/3}) \quad (\mathrm{W/m^2K}) \\ &[(h_{forced\_CC} = [h_{total}^3 - (0.308 \cdot \Delta T^{0.33})^3]^{1/3}) \quad (\mathrm{Btu/h \cdot ft^2 \cdot F})] \end{aligned} \tag{9}$$

The convection coefficients were calculated for each experiment. Then, using h_{total} and the corresponding ACH for each experiment, the function fitting by the least squares method provided the coefficients c = 2.0 and m = 0.39. Figure 9 presents the resulting forced convection correlation at ceiling panels.

For vertical surfaces, the ratio Gr/Re^2 ≈ 1 indicated the presence of both forced and natural convection. In the experiments with low volume flow rates (1 and 2 ACH), natural convection created by low temperature differences (ΔT_{local} = 0.5–1.5°C or ΔT_{local} = 0.9–2.7°F) had a significant influence on the total convective heat transfer. For the experiments with larger volume

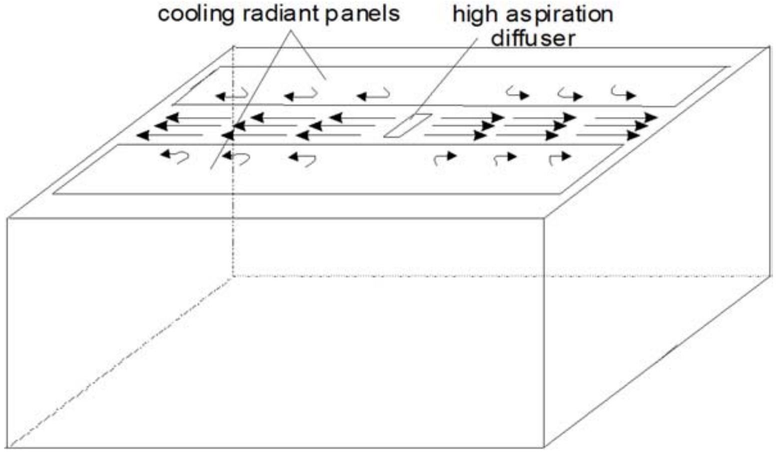

Figure 8. The smoke visualization results of the airflow pattern for the high aspiration diffuser providing a narrow jet with a high entrainment of the room air.

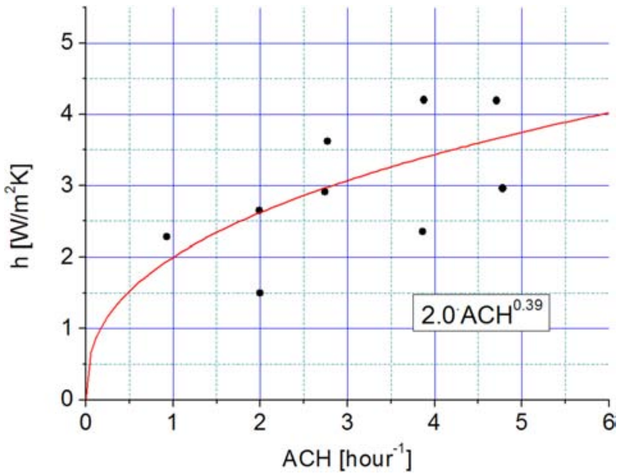

Figure 9. The forced convection correlation for a ceiling in a room with a high aspiration diffuser.

flow rates (ACH = 3, 4, and 5), forced convection was dominant. The effects of natural convection with low flow rates were taken into account in the same way as for the CC panels. The correlation developed by Awbi and Hatton (1999) for vertical surfaces is accurate for natural convection calculations when the local air temperatures are used as the reference temperatures (Novoselac et al. 2005). Using this correlation to express $h_{natural}$ in Equation 7, the forced convection at vertical surfaces was calculated by:

$$(h_{forced\_walls} = [h_{total}^3 - (1.823/D_h^{0.121} \cdot \Delta T^{0.293})^3]^{1/3}) \quad (\mathrm{W/m^2K})$$
$$[(h_{forced\_walls} = [h_{total}^3 - (0.3121/D_h^{0.121} \cdot \Delta T^{0.293})^3]^{1/3}) \quad (\mathrm{Btu/h \cdot ft^2 \cdot F})] \qquad (10)$$

The directional jet created a situation where the side walls that the jet impinged had a larger velocity and forced convection coefficients than the wall surfaces parallel to the jet. The difference in local convection coefficients at different walls varied up to 50%. In this way, forced convection coefficients for walls depend on room geometry and the orientation of the high aspiration diffuser's jets.

However, to create general correlations, which are applicable in practice where the orientation of the diffuser is not often available, forced convection coefficients for walls were averaged and presented as a function of ACH in Figure 10.

Figure 11 shows measurement results for floor surfaces. Unlike for vertical walls, the measured forced convection coefficients at floor surfaces were rather uniform. Also, temperature differences were very small, and, thus, the local air temperature was stably stratified. This resulted in a very low impact of the floor natural convection on the total convection. With all analyzed flow rates, the convection was primarily due to the dominant forced airflow in the vicinity of the floor surfaces. Therefore, the function form of Equation 6 was used to express the forced convection coefficient at floor surfaces (Figure 11).

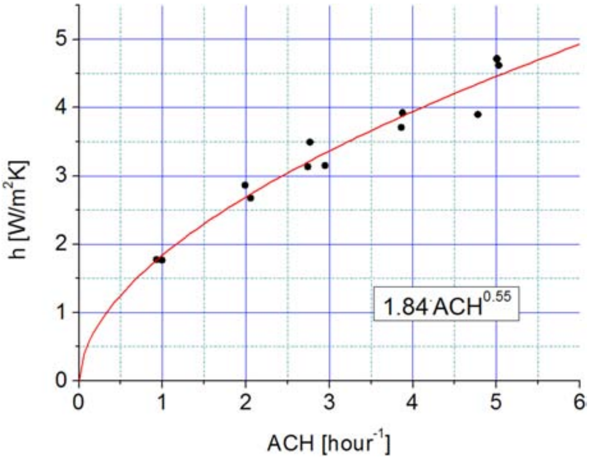

Figure 10. The forced convection correlation for vertical walls in a room with the high aspiration diffuser.

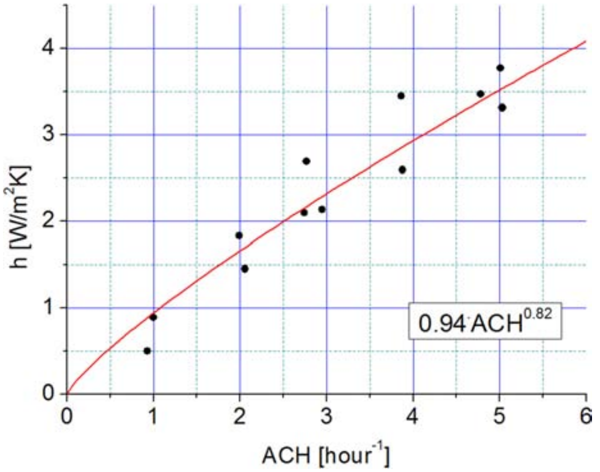

Figure 11. The forced convection correlation for a floor in a room with the high aspiration diffuser.

Combined Forced and Natural Convection

To create general surface convection correlations for rooms with CC panels where ventilation is provided by high aspiration diffusers, newly developed correlations for CC surfaces (Equation 8) were combined with the expressions for forced convection for the ceiling (Figure 9). For CC surfaces the general correlation is:

$$\left(h_{total} = [(2.12 \cdot \Delta T^{0.33})^3 + (2.0 \cdot \mathrm{ACH}^{0.39})^3]^{1/3}\right) \qquad (\mathrm{W/m^2K})$$
$$\left[\left(h_{total} = [(0.308 \cdot \Delta T^{0.33})^3 + (0.35 \cdot \mathrm{ACH}^{0.39})^3]^{1/3}\right) \qquad (\mathrm{Btu/h \cdot ft^2 \cdot F})\right] \tag{11}$$

Table 2. New Convection Correlations for a Room with CC Panels and a High Aspiration Diffuser

Surface	Regime	Convection Correlation	
Floor	$T_s > T_{air}$	$h_c = [(2.175 \cdot \Delta T^{0.308} / D_h^{0.076})^3 + (0.94 \cdot \text{ACH}^{0.82})^3]^{1/3}$	$(\text{W/m}^2\text{K})$
		$h_c = [(0.35 \cdot \Delta T^{0.308} / D_h^{0.076})^3 + (0.166 \cdot \text{ACH}^{0.82})^3]^{1/3}$	$(\text{Btu/h} \cdot \text{ft}^2 \cdot \text{F})$
	$T_s < T_{air}$	$h_c = [(0.704 \cdot \Delta T^{0.133} / D_h^{0.601})^3 + (0.94 \cdot \text{ACH}^{0.82})^3]^{1/3}$	$(\text{W/m}^2\text{K})$
		$h_c = [(0.234 \cdot \Delta T^{0.133} / D_h^{0.076})^3 + (0.166 \cdot \text{ACH}^{0.82})^3]^{1/3}$	$(\text{Btu/h} \cdot \text{ft}^2 \cdot \text{F})$
Ceiling	$T_s > T_{air}$	$h_c = [(0.704 \cdot \Delta T^{0.133} / D_h^{0.601})^3 + (2.0 \cdot \text{ACH}^{0.39})^3]^{1/3}$	$(\text{W/m}^2\text{K})$
		$h_c = [(0.234 \cdot \Delta T^{0.133} / D_h^{0.076})^3 + (0.35 \cdot \text{ACH}^{0.39})^3]^{1/3}$	$(\text{Btu/h} \cdot \text{ft}^2 \cdot \text{F})$
	$T_s < T_{air}$	$h_c = [(2.12 \cdot \Delta T^{0.33})^3 + (2.0 \cdot \text{ACH}^{0.39})^3]^{1/3}$	$(\text{W/m}^2\text{K})$
		$h_c = [(0.308 \cdot \Delta T^{0.33})^3 + (0.35 \cdot \text{ACH}^{0.39})^3]^{1/3}$	$(\text{Btu/h} \cdot \text{ft}^2 \cdot \text{F})$
Walls		$h_c = [(1.823 \cdot \Delta T^{0.293} / D_h^{0.121})^3 + (1.84 \cdot \text{ACH}^{0.55})^3]^{1/3}$	$(\text{W/m}^2\text{K})$
		$h_c = [(0.3121 \cdot \Delta T^{0.293} / D_h^{0.121})^3 + (0.324 \cdot \text{ACH}^{0.55})^3]^{1/3}$	$(\text{Btu/h} \cdot \text{ft}^2 \cdot \text{F})$

This equation shows that for the temperature difference of ΔT = 7°C (ΔT = 12.6°F) and a range of volume flow of 1–4 ACH, the increase of the surface convection coefficient, due to the presence of a high aspiration diffuser, is in the range of 4%–17%.

For wall and floor surfaces, the forced convection correlations presented in Figures 10 and 11 can be combined with correlations for natural convection to obtain general expressions. Natural convection correlations developed by Awbi and Hatton (1999) seem to be the most suitable for this general expression because these correlations were developed in a similar experimental environment using local air temperatures (0.1 m from wall) as the reference temperature. Table 2 contains the final general forms of convection correlations for rooms with CC panels combined with high aspiration diffusers.

CONCLUSIONS

This paper presented the development of new convection correlations for rooms with cooled ceiling (CC) panels. Based on measurements in a full-scale environmental chamber, a correlation for the natural convection at cooled ceiling surfaces was developed. Furthermore, measurements in experiments with a high aspiration diffuser, which is used with CC panels, quantified the effects of forced convection created by the diffuser's jet on ceiling surfaces as well as at wall and floor surfaces.

Measured data and their analyses showed that downward thermal plumes at cooled ceiling surfaces create primarily turbulent airflow. Furthermore, the upward thermal plumes from the room's internal heat sources intensified the air mixing in the vicinity of the CC and increased the convective heat transfer at the cooling panels. Therefore, the newly developed natural convection correlation for CC calculates 5%–9% larger convection coefficients than previously developed correlations, which are based on measurements at hot floor surfaces and do not account for the collision of upward and downward thermal plumes.

The results of experiments with a high aspiration diffuser showed that this diffuser can produce good air mixing even with very low volume flow rates that provide 1 to 2 ACH in the room, which is a common situation with dedicated outdoor air systems (DOAS). Measured data also show that a high aspiration diffuser creates very narrow directional jets, which very locally increase velocities at ceiling surfaces. However, the overall effect of forced convection on room surfaces' heat fluxes created by the high aspiration diffuser was not negligible. For CC surfaces, the increase was from 4% to 17% depending on the flow rate. The effect of forced convection on the convective coefficient is a function of normalized volume flow rate (ACH), and forced correlations for the floor, ceiling, and walls were developed as the functions of ACH. To obtain a general form of the correlation for CC combined with high aspiration diffusers, the newly developed forced convection correlations were combined with expressions for natural convection and presented in Table 2.

Correlations in Table 2 are based on local air temperatures and should be used in analyses of ventilating and air-conditioning systems that provide good air mixing and uniform temperature in a room. With systems that provide nonuniform air temperature distributions, such as displacement ventilation or air heating systems, the new correlation for CC (Equation 8) can be combined with models that calculate the temperature distribution or temperature gradient in a room.

ACKNOWLEDGMENT

This research was financially supported by the National Science Foundation (NSF Grant No. CTS-0134326).

NOMENCLATURE

a = coefficient for determining convective heat transfer coefficient under laminar flow
b = coefficient for determining convective heat transfer coefficient under turbulent flow
c = coefficient for determining forced convective heat transfer coefficient
C_L = coefficient for determining Nusselt number under a laminar flow
ACH = air changes per hour [hour^{-1}]
C_T = coefficient for determining Nusselt number under a turbulent flow
D_h = hydraulic diameter, m (ft)
h = convective heat transfer coefficient, W/m^{2}K (Btu/h·ft^2·F)
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 (Btu/h·ft^2)
Re = Reynolds number
ΔT = characteristic temperature difference; between surface and air layer 0.1 m from surface, °C (°F)
V_{room} = volume of the room, m^3 (ft^3)
V_{supply} = supply flow rate from the diffuser, m^3/h (ft^3/h)
L = characteristic length, m (ft)
T = temperature, °C (°F)

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