Convective Heat Transfer in Building Energy and Thermal Load Calculations

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ABSTRACT

Valid enclosure film coefficients, required by hourly energy and thermal load programs, were experimentally determined for ventilative flow rates below 12 air changes per hour (ACH). Forty-eight experiments were performed in a full-scale room in order to determine film coefficients at low ventilative flow rates. The experiments, which were performed over a range of conditions from 3 to 12 ACH, showed that for most room configurations, natural convection film coefficients significantly underpredict the rate of surface convective heat transfer. The new film coefficients were implemented in the BLAST program, an hourly heat-balance-based building energy simulation. The significance of the error incurred by using natural convection film coefficients in a ventilated space was estimated by comparing BLAST results obtained with both natural convection and the new film coefficients. For the case of space cooling, where surface-to-air temperature differences are relatively small, errors in calculated space cooling loads were typically on the order of 10%.

INTRODUCTION

Calculation of room surface-to-air heat transmission is dependent on an accurate estimate of the film coefficient. Forty-eight experiments were performed in a full-scale room in order to determine the significance of the error incurred by the common practice of using natural convection heat transfer coefficients in a mechanically ventilated room.

The experimental work, which was sponsored by ASHRAE as RP-664, was performed over a range of realistic room configurations. Ventilative flow rates were set between 3 and 12 air changes per hour (ACH), and inlet air temperature setpoints were specified between 10°C (50°F) and 25°C (77°F). The experimental room was configured with both isothermal and nonisothermal interiors and with both ceiling and sidewall diffusers.

The research showed that actual room film coefficients may be up to 20 times higher than natural convection based coefficients. BLAST simulations showed that the error introduced by the natural convection assumption typically results in an annual calculated cooling load error on the order of 10%.

Significantly, the research also showed that rooms with radial ceiling diffusers are relatively well stirred, even at ventilative flow rates as low as 3 ACH. This unexpected fact led to the simplified engineering correlations that are presented in this paper: heat transfer coefficients as a function only of ACH. The correlations can be easily implemented in room thermal load programs, building energy analysis programs, and other engineering calculations.

The paper also demonstrates the importance of selecting the correct reference temperature in defining the film coefficient. The uncertainty analysis associated with the experimental method argues in favor of a room inlet reference temperature.

The temperature and flow fields of recirculating cavities and enclosures have enjoyed considerable attention in the literature during the last 20 years. Hundreds of papers present the results of computational studies, and dozens more present experimental results. Although many researchers have studied the buoyancy-driven enclosure (Bauman et al. 1980; Nansteel and Greif 1981, 1983; Bohn and Anderson 1984, 1986; Allard et al. 1990; Chen et al. 1990) and a significant number have investigated forced convection at high ventilative flow rates (Spitler et al. 1991b; Nielsen et al. 1978, 1979; van der Kooi and Bedeke 1983; van der Kooi and Forch 1985; Murakami et al. 1987; Neiswanger et al. 1987), relatively few have investigated enclosure heat transfer at low ventilative flow rates (Neiswanger et al. 1987; Chen et al. 1989; Kapoor and Jaluria 1991; Pavlovic and Penot 1991). Of these, none has developed convective heat transfer correlations for realistic room configurations.

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This study used the experimental building enclosure constructed by Spitler et al. (1991a) and designed originally for convective heat transfer experiments at high ventilative flow rates. The facility, which was modified to accommodate research in the mixed and natural convection flow regimes, is unique in several respects. First, the enclosure is relatively large—the size of a small office. Second, all interior surfaces can be temperature controlled.

For many of the experiments presented in this paper, all room surfaces were controlled at the same temperature. The isothermal room configuration was an important factor in minimizing the uncertainty associated with the experimentally determined film coefficients. For this configuration, the radiation component of the surface heat transfer was small compared to the magnitude of the convective flux; even a relatively large uncertainty in the radiation heat transfer calculation had an insignificant effect on the uncertainty associated with the convective heat transfer coefficient.

A number of nonisothermal experiments were also performed in order to validate the application of correlations to rooms with nonuniform surface temperatures. The nonisothermal room was configured with three "hot" walls, a "hot" ceiling, a "hot" floor, and one "cold" wall.

THE EXPERIMENTAL FACILITY

The experimental facility, shown in Figure 1, consisted of an office-sized chamber located inside a larger structure. The space between the outer shell and the experimental enclosure was a temperature-controlled "guard space" that maintained a temperature difference of -0°C (32°F) across the walls of the experimental enclosure.

The enclosure, which consisted of 53 individually controlled heated panels, was ventilated through either of two inlets located in the ceiling and side wall. The layout of the heated panels and the air inlets and outlets is illustrated in Figure 2. Construction and validation of the facility are discussed by Fisher (1989).

One wall of the experimental room (S. Wall) was constructed with plate heat exchangers behind the heated panels. All of the nonisothermal room experiments were performed by circulating chilled water through these heat exchangers. The construction and instrumentation of the cold wall are discussed by Fisher (1995) and Mansfield (1993).

Each heated panel was instrumented with two surface thermocouples. Local heat transfer coefficients were calculated using the average of the two panel surface temperatures. The calculation of the interior radiant energy exchange was based on the assumption that the panel surface was uniform at the average of the two measured temperatures. At the low airflow rates typical of the mixed convection regime, temperature differences between any two points on a single panel were less than 0.5°C (0.9°F) (Fisher 1989). The uncertainty introduced by this error was incorporated in the calculation of the surface film coefficient uncertainty.

A chilled-water coil with electric reheat maintained the desired room air inlet temperature. Thermocouple grids located at the room air inlets and outlet measured entering and leaving air temperatures. A flow measurement box constructed according to ANSI/ASHRAE Standard 51-1985 (ASHRAE 1985) provided pressure and temperature information required to calculate the air mass flow rate to the room.

The room ventilation rate was calculated from the measured pressure drop across one or two elliptic flow nozzles of known characteristics. Various nozzle combinations were used to obtain ventilative flow rates ranging from 0.05 to 2 m³/s.

![Figure 1: Schematic of experimental facility (Fisher 1989).](Image)

![Figure 2: Heated panel layout (Fisher 1989).](Image)
Air speeds and temperatures in the experimental room were measured by 16 omnidirectional air speed transducers and 16 type-T thermocouples attached to a computer-controlled trolley (Cantillo 1990). The trolley, moving horizontally and vertically, typically collected data at 1,000 locations in the half-room.

Air temperature measurements were particularly important in evaluating possible reference temperatures. A number of bulk air and planar temperatures were investigated as possible references for defining the heat transfer coefficient.

The convective flux was explicitly calculated from measured surface temperatures and the thermal resistance of the guard space:

\[ q_{\text{conv}} = \frac{V^2}{R_i A_i}(W/m^2). \]  

(2)

The net radiant heat transfer from the ith room surface to all the other surfaces in the enclosure is given by Hottel and Sarofim (1967) as

\[ q_i - s = \frac{1}{A_i} \sum_{j=1}^{\infty} \left[ F_{ij} \cdot \sigma \cdot (T_i^4 - T_j^4) \right](W/m^2). \]  

(3)

Finally, the rate of heat transfer from the inside surface to the guard space (the “back loss”), \( q_{i-hl} \), is calculated from the measured surface temperatures and the thermal resistance of the surfaces:

\[ q_{i-hl} = \left( \frac{T_i - T_{ref}}{R_i} \right)(W/m^2). \]  

(4)

Thus, the convective flux was explicitly calculated from experimental measurements for each surface in the room. Since the guard space was controlled to the inside surface temperature, for the isothermal room configuration the only significant term on the right-hand side of the equation is \( q_{i-pwr} \).

The convective heat transfer coefficient was calculated from the rate of convective heat transfer and the temperature difference between the surface and an arbitrarily selected reference temperature.

\[ h_i = \frac{q_{i-conv}}{(T_i - T_{ref})}. \]  

(5)

The selection of the reference temperature was arbitrary in the sense that for enclosure heat transfer, a clear and obvious choice for a temperature reference does not exist. An important part of the investigation was to examine the impact of various reference temperatures on the proposed correlations and on the experimental uncertainty associated with the heat transfer coefficients. The room inlet temperature, the room outlet temperature, and spatially averaged planar and bulk air temperatures were examined as possible references.

**Summary of Experiments**

Of the 48 experiments performed during the course of the investigation, 16 were performed with radial ceiling diffusers in the isothermal room and 20 with radial ceiling diffusers in the nonisothermal room. In addition, several heat gain experiments were performed with reduced room volume in order to estimate the effect of furniture. The balance of the experiments studied the free, horizontal jet and are not discussed in this paper.
The radial ceiling diffuser, used in many commercial applications, is of great practical importance. Several radial diffusers of different design and open area were used in the ceiling jet experiments in order to vary the outlet velocity of the jet and study the effect of diffuser type on surface convection. Table 2 shows the experiment code, the ventilative flow rate, the air inlet temperature, and the diffuser open area for each of the isothermal room ceiling experiments.

**TABLE 2**

<table>
<thead>
<tr>
<th>File</th>
<th>Flow Rate (ACH)</th>
<th>Inlet Temp.</th>
<th>Diffuser Area (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>e0112933</td>
<td>3.00</td>
<td>10.83</td>
<td>0.022</td>
</tr>
<tr>
<td>e0905921</td>
<td>3.00</td>
<td>19.98</td>
<td>0.0085</td>
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<tr>
<td>e1211921</td>
<td>3.04</td>
<td>20.04</td>
<td>0.0051</td>
</tr>
<tr>
<td>e1204922</td>
<td>5.95</td>
<td>9.98</td>
<td>0.0116</td>
</tr>
<tr>
<td>e0828921</td>
<td>5.99</td>
<td>15.04</td>
<td>0.0085</td>
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<td>e0624923</td>
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<td>e0299921</td>
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<td>15.03</td>
<td>0.022</td>
</tr>
<tr>
<td>e0618921</td>
<td>11.96</td>
<td>20.02</td>
<td>0.022</td>
</tr>
</tbody>
</table>

Each data set was initially screened by calculating an overall room heat balance from the data. It was required that the total energy transferred to the room air during the course of the experiment, \(E_{air}\), be within 10% of the total electrical energy dissipated by all room panels, \(E_{pwr}\):

\[
\frac{(E_{air} - E_{pwr})}{E_{pwr}} \leq 0.1
\]  

where

\[
E_{air} = \sum_{t=1}^{n} \dot{m}_t \cdot C_{p,air} (T_{out,t} - T_{in,t}) \cdot \text{etime}
\]

\[
E_{pwr} = \sum_{t=1}^{n} \text{etime} \sum_{i=1}^{53} \frac{V_i^2}{R_i}
\]

\[\dot{m}_t = \text{mass flow rate at time } t,\]

\[C_{p,air} = \text{specific heat of air,}\]

\[etime = \text{time step,}\]

\[T_{out,t} = \text{air outlet temperature,}\]

\[T_{in,t} = \text{air inlet temperature,}\]

\[V_i = \text{AC voltage, and}\]

\[R_i = \text{panel electrical resistance.}\]

This preliminary screening ensured that the facility was indeed at quasi-steady state and that the data-acquisition system was functioning properly. Calculation of the standard deviation of all measurements over the course of the experiment served as an additional check on both instrumentation and the data-acquisition system. Finally, uncertainties in both measured values and calculated results were computed for each data set.

**EVALUATION OF THE FLOW AND TEMPERATURE FIELDS**

Three methods were used to examine the validated data sets. First, raw air speed and air temperature data were rendered as color or gray scale maps in order to visualize the flow field. Second, a large data analysis program, originally written by Spitler (1990), calculated heat transfer coefficients based on various reference temperatures as well as dimensionless groups and experimental uncertainties, all of which were required for the development of correlations. Finally, special attention was given to the inlet jet. For each diffuser type the ceiling jet was characterized by additional air speed measurements near the ceiling.

**The Effect of Buoyancy and Coanda Forces on Room Convective Heat Transfer**

Visualization of the flow field led to the important conclusion that over the range of experimental measurements, the rate of convective heat transfer was relatively independent of the density of the inlet jet. The gravity force that pulls a dense, cold ceiling jet toward the floor is opposed by the viscous Coanda force, which pulls the jet in the opposite direction—toward a low-pressure region near the ceiling. For jet inlet temperatures as low as 10°C (50°F) and jet inlet velocities as low as 1.3 m/s (4.2 ft/s), the Coanda effect dominated the flow field and the jet did not detach from the ceiling.

The flow and temperature fields were qualitatively evaluated using air speeds and temperatures measured at quasi-steady-state conditions. The measurements were taken at 1,000 locations in the half-room by the thermocouples and air speed probes attached to the trolley system. Figures 3 through 5 show a single plane of measured air speeds for a given set of inlet conditions. The outline of the “data cube” (measurement region) with the air inlet and outlet is shown to illustrate the orientation and location of the measurement plane. The measurement plane itself shows only air speeds above the specified threshold velocity. “Low-speed” areas in the measurement plane are “transparent.”

Figures 3a and 3b show air speeds of more than 20 fpm and 15 fpm for a volumetric flow rate of 9 ACH and an inlet air temperature of 10°C (50°F). Since the air speed probes did not penetrate the attached ceiling and wall jets, the “high-speed”
Similar air speed patterns were observed at both higher and lower volumetric flow rates. The flow fields shown in Figure 3 are typical of the Spitler (1990) data and the higher volumetric flow rate experiments performed as part of this investigation. The characteristic flow is also observed at lower volumetric flow rates. Figure 4 shows the same plane of air speed measurements for inlet conditions of 6 ACH at 10°C (50°F). Even at this relatively low volumetric flow rate and low inlet air temperature, the buoyancy of the jet is not sufficient to overcome the Coanda effect.

At the lowest volumetric flow rate and lowest air inlet temperature, the effect of buoyancy begins to show. In Figure 5a the ceiling jet is not visible. This is not because it does not exist, but rather because the jet-affected zone is not penetrated by the air speed probes. In contrast, the flow field near the floor, fed by the downward flow of cold air, is relatively active. The "still air core" is discernible in Figure 5b but is less well defined than at higher flow rates.

The data presented in Figures 3 through 5 indicate that the density differential between the cold, mechanically driven jet and the room air has a negligible effect on the flow field in comparison to the Coanda effect. The jet remained tenaciously attached over the entire range of conditions such that even the flow along the floor appears to be driven by the attached ceiling and wall jets.

Figure 6 shows measured temperatures for points coincident with the air speed measurements shown in Figures 3 through 5. Isotherms at 0.5°C (0.9°F) intervals are shown on the plots.
Figure 5  Measured air speeds for 6 ACH, 10°C: above 15 fpm (a) and above 10 fpm (b).

Figure 6  Temperature maps at 9 ACH (a), 6 ACH (b), and 3 ACH for 10°C inlet air.

At 9 ACH (Figure 6a), the room appears to be relatively "well stirred" with respect to temperature. There is no sign of stratification, and the temperature in the measurement plane varies by less than 1°C (1.8°F). At 6 ACH (Figure 6b), the room is still relatively well stirred with respect to temperature. The "cold corners" (also clearly visible at 9ACH) show the jet leaving the ceiling and reattaching to the sidewalls but do not indicate a buoyancy effect. The unistrut trolley rails mounted in the upper right and left corners of the room tend to exaggerate the effect of the upper corners.

Although the room air temperature is still quite uniform at 3 ACH (Figure 6c), some temperature stratification is evident at this volumetric flow rate. Buoyant forces are no longer negligible, and one would theoretically expect to see their effect on the rate of convective heat transfer—especially on the floor. The momentum of the mechanically driven jet, however, still dominates the flow field, even at these extreme conditions.

Selecting the Reference Temperature

Two criteria were applied in selecting the reference temperature for room convective heat transfer calculations. The first criterion required that the reference temperature be readily available both in energy simulation programs and to building designers in general. The second criterion required that the reference temperature minimize the experimental uncertainty associated with the calculation of the heat transfer coefficients. Based on these criteria, the room inlet temperature was the best reference for defining room convective heat transfer coefficients.

The first criterion eliminated all but three possibilities: the room inlet temperature, the room outlet temperature, and the "bulk" (or mean) air temperature. Other possible reference temperatures, such as near-wall air temperatures, can only be obtained in the laboratory. Of the three possibilities, the room inlet and outlet temperatures are most readily available both in practice and in energy simulation programs.

Although Spitler used the room outlet temperature as the reference for the convective heat transfer coefficient (Spitler et al. 1991b), at low ventilative flow rates the uncertainty in the film coefficient associated with an outlet reference temperature is unacceptably high. At low volumetric flow rates, the outlet temperature approaches the room surface temperature. This small ΔT leads directly to high uncertainties in the film coefficients.
Figure 7 Uncertainty in wall h for outlet reference (a) and inlet reference (b).

Figure 7 shows the calculated uncertainty in film coefficients using inlet and outlet reference temperatures. The figure shows average wall heat transfer coefficients for 15°C (59°F) inlet air experiments. The uncertainties shown in the figure are typical of room heat transfer coefficients over the range of experimental data.

Based on the experimental uncertainty, the inlet temperature is clearly the better choice for a reference temperature. The plots also indicate that the inlet reference temperature may correlate the data better. The heat transfer coefficients based on the inlet temperature showed good sensitivity to ACH over the entire range of conditions.

The Effect of the Radial Diffuser on Room Convective Heat Transfer

For a given volumetric flow rate, the “open area” of a ceiling diffuser determines the velocity of the jet at the room inlet. Experiments were designed to answer the question of whether or not the rate of convective heat transfer is dependent on the inlet jet velocity. The experimental evidence presented in the following paragraphs indicates that for a radial diffuser, surface convection is independent of the inlet jet velocity.

A number of tests at 12 and 6 ACH were performed in the isothermal room with 20°C (68°F) inlet air and 30°C (86°F) surfaces. For each test, the velocity of the jet at the inlet was varied for a constant volumetric flow rate by changing the open area of the radial ceiling diffuser.

Two types of diffusers were used for the tests, as shown in Figures 8 and 9. The first type was simply a cylindrical cover with evenly spaced slots or holes on the cylinder wall; the second type was an adjustable pan-type diffuser.

The measured rate of convective heat transfer showed no sensitivity to the inlet velocity regardless of the proximity to the inlet jet. Figure 10 shows the average rate of convective heat transfer for the three heated ceiling panels closest to the ceiling diffuser. The first case, with an approximate inlet jet velocity of 1.19 m/s (3.9 ft/s), is for the pan-type diffuser with a 2-in. (51-mm) gap between the cover plate and the diffuser pan. The second case, with an inlet jet velocity of 2.37 m/s (7.7 ft/s), is for a 1-in. (25.4-mm) gap with the pan-type diffuser. The third case, with an inlet jet velocity of 5.11 m/s (16.7 ft/s), is for the cylindrical-type diffuser shown in Figure 8. As the inlet velocity is changed by greater than a factor of four with a constant volumetric flow rate of 12 ACH, the convective flux remains relatively constant. The variations in the convective flux are explained by slight variations in inlet temperature, volumetric flow rate, and perhaps also by the jet attachment point.

EXPERIMENTAL RESULTS

Development of Correlations

Several conclusions can be drawn from the evaluation of the flow and temperature fields. First, due to the dominance of the Coanda effect, the flow field is momentum driven and is relatively unaffected by buoyancy. Second, the temperature of the inlet jet is the best reference for the definition of the surface heat transfer coefficients. Finally, notwithstanding the importance of momentum in determining the flow field, the surface heat transfer is independent of the inlet jet velocity.

The fact that the film coefficients are independent of the jet velocity precludes the possibility of correlating surface film coefficients to jet-velocity-based parameters such as the outlet Reynolds number:

\[ Re_o = \frac{U_o L}{v} \]  

where \( U_o \) is the jet velocity at the diffuser, \( L \) is a diffuser-based length scale, and \( v \) is the kinematic viscosity. Attempts
at correlating the heat transfer coefficients to the Reynolds number and other velocity-based parameters predictably failed.

Momentum dependence was tested by calculating the jet momentum number, \( J \), which had been previously used to correlate room heat transfer coefficients at high ventilative flow rates. The results of this test showed that room surface heat transfer was also independent of jet momentum for the ceiling inlet configuration.

The jet momentum number is defined as

\[
J = \frac{mU_o}{\rho g V_{room}} \tag{8}
\]

where
- \( m \) = mass flow rate of ventilation air,
- \( U_o \) = jet velocity at the diffuser,
- \( \rho \) = density of air,
- \( g \) = acceleration due to gravity, and
- \( V_{room} \) = volume of room.

For high ventilative flow rates in the enclosure, Spitler correlated the forced convection heat transfer coefficient to the jet momentum number (Spitler et al. 1991b). Variable diffuser area tests performed for the sidewall inlet at high ventilation rates showed that for the conditions under consideration, the rate of surface heat transfer was sensitive to changes in the inlet momentum. However, for the radial ceiling diffuser, the rate of surface heat transfer is completely insensitive to variations in the jet momentum number. Figure 11, which shows heat transfer coefficients based on an inlet air reference temperature, illustrates the independence of surface heat transfer and the momentum of the inlet jet.

The development of successful correlations was based on the observation that although the heat transfer coefficient was independent of both the inlet jet velocity and the inlet jet momentum, it was dependent on the jet mass flow rate. For incompressible flow over the temperature range studied, volumetric flow rate also resulted in reasonable correlations of the surface heat transfer. The implication of this observation is that in developing whole-room convective heat transfer coefficients, the physics must be understood in terms of the room control volume rather than in terms of the surface boundary layer.

The ventilative flow rate is proportional to the total energy delivered to the room at a given inlet air temperature. When normalized to the room volume, this parameter scales the room convective heat transfer to its fundamental driving potential—the difference between the surface temperature and the air inlet temperature. Preliminary experimental evidence indicates that the validity of the correlations depends only on a basic similarity of the flow field. Thus a single correlation for each surface orientation described the room convective heat transfer from 3 ACH to 100 ACH for a variety of diffusers, including the diffusers described in this paper and the commercial diffuser used by Spitler.

As shown in Figures 12 through 14, the volumetric flow rate dependence is of the form \( (h-\text{ACH}) \). The figures show room convective heat transfer coefficients as a function of volumetric flow rate for an inlet air reference temperature.

**Experimental Uncertainty and Range of Correlations**

The art of uncertainty analysis is well established in the thermal sciences; numerous texts and papers address the topic, including excellent texts by Taylor (1982) and Holman (1989). Of particular importance to this investigation is the fact that Spitler validated both the experimental procedure and the performance of the facility at high ventilative flow rates (Spitler 1990).

The classic paper on uncertainties in single-sample experiments was published by Kline and McClintock (1953), who showed that for the special case of a linear function with independent variables, each of which is normally distributed, the relation between the uncertainty interval for the variable and the uncertainty interval for the result is given by
\[
\epsilon_R = \left[ \left( \frac{\partial R}{\partial v_1} \epsilon_1 \right)^2 + \left( \frac{\partial R}{\partial v_2} \epsilon_2 \right)^2 + \ldots + \left( \frac{\partial R}{\partial v_n} \epsilon_n \right)^2 \right]^{1/2}
\]  \hspace{1cm} (9)

where

\[ R = \text{calculated result, } R = R(v_1, v_2, \ldots, v_n), \]
\[ \epsilon_R = \text{uncertainty interval in the result, and} \]
\[ \epsilon_i = \text{uncertainty interval in the } i\text{th variable.} \]

The partial derivative, \( \partial R/\partial v_i \), is a measure of the sensitivity of the result to a single variable. Multiplying the sensitivity or “influence” coefficient by the estimated uncertainty in the variable provides an estimate of the variable’s contribution to uncertainty in the result. The method of influence coefficients has traditionally been extended to nonlinear relationships by first-order expansion of the governing equation and evaluation of the partial derivatives at the base values (Holman 1989).

This method was used to calculate the uncertainty associated with the experimentally determined heat transfer coefficients. Figures 15 through 17 show the uncertainty in heat transfer coefficients for a room configured with a ceiling diffuser. The film coefficients are based on a jet inlet reference temperature.

The correlations shown in Table 3 are applicable to an isothermal room with a radial ceiling jet between 10°C (50°F) and 25°C (77°F) and an enclosure air change rate (ACH) within the range (3 < ACH < 100). The convective heat transfer coefficient, \( h \), is based on a reference temperature measured in the supply air duct.

**TABLE 3**

Heat Transfer Coefficients for Ceiling Inlet Configuration

<table>
<thead>
<tr>
<th>Surface Type</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Walls</td>
<td>( h = 0.19 \cdot \text{ACH}^{0.8} )</td>
</tr>
<tr>
<td>Floor</td>
<td>( h = 0.13 \cdot \text{ACH}^{0.8} )</td>
</tr>
<tr>
<td>Ceiling</td>
<td>( h = 0.49 \cdot \text{ACH}^{0.8} )</td>
</tr>
</tbody>
</table>

Although the correlations are subject to restrictions, they are suitable for many HVAC applications. The experimental range of both volumetric flow rate and inlet air temperature covers the normal operating range of most buildings. In spite of the fact that a number of variables (such as room aspect ratio, interaction of inlet jets from multiple diffusers, furnishings, and internal heat sources) were not checked by the parametric set of experiments, the fact that

**Figure 11** Convective heat transfer coefficient as a function of jet momentum number.
ceiling heat transfer dominates overall room convection is expected to mitigate the sensitivity of the correlations to these variables.

Twenty experiments were performed in the nonisothermal room to determine the applicability of the correlations (derived from isothermal room data) to realistic room surface temperature profiles. This set of experiments showed that the correlations can be applied to nonisothermal rooms with surface temperature differences of less than 20°C (68°F) without serious error. Although the room flow fields are significantly different, the maximum deviation in the heat transfer coefficient at 6 ACH with a 20°C (68°F) surface temperature differential was 15%, as shown in Figure 18.

APPLICATIONS

Implementing the Correlations

The expressions for room convective heat transfer coefficients presented in Table 3 are suitable for implementation in any heat balance based building energy or thermal load program with...
sufficient detail to support the calculations. A significant requirement is that the radiation and convection coefficients not be combined for computational purposes. The correlations cannot be easily incorporated in room weighting factors or in surface response factors that include the air nodes. Likewise, simplified fenestration models with combined radiation and convection coefficients will not be able to utilize the results of this study.

The new convective heat transfer coefficients were implemented in the BLAST (Building Loads and System Thermodynamics) program. BLAST implementation of time-varying film coefficients was essentially demonstrated in Appendix G of Spitler (1990). The correlations can also be implemented in computational fluid dynamic codes, where in most cases they will represent a significant improvement over typical wall functions.

**Significance of Results: A Case Study**

A BLAST model of a 1.4 million ft² office was used for this exercise. The facility, a seven-story building with masonry walls (24% glass) and a built-up asphalt roof with fiberglass and felt insulation, was modeled as seven occupied zones with an internal occupancy load of 1 person/100 ft², a lighting load of 2.0 W/ft², and an equipment load of 0.4 W/ft². The “dead air spaces” between the suspended ceilings (with recessed lighting) and the next floor (or roof) were modeled as separate, “uncontrolled” zones. “High internal loads” were simulated by increasing the occupancy load to 1 person/50 ft² and increasing the equipment load to 2.0 W/ft². High solar loads were simulated by increasing the glazing area to 54% of the total wall surface area. For the purposes of this exercise, both buildings were located in the central Midwest (St. Louis, Missouri).

Table 4 shows the error in the daily cooling load due to an error in the ceiling convective heat transfer coefficient. Column one shows the daily cooling load for each case with the currently used BLAST convective heat transfer coefficients. Column two shows the daily cooling load for each case calculated with the heat transfer coefficients that are associated with a moderate ventilation rate of 6 ACH with a 10°C (50°F) reference temperature. The dead air space heat transfer coefficients were set at the BLAST defaults (natural convection) for all 12 cases. Column three shows the percent difference between column one and column two. The example, though valid for only one building in one location, shows up to an 11% error in the daily calculated load on the system.

**TABLE 4**

BLAST Loads for Natural and Mixed Convection Heat Transfer Coefficients

<table>
<thead>
<tr>
<th>Case</th>
<th>Sens. Load (kBtu/h), Natural Convection</th>
<th>Sens. Load (kBtu/h), Low Momentum</th>
<th>Percent Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single story, low internal loads</td>
<td>3.520E + 03</td>
<td>3.897E + 03</td>
<td>10.7%</td>
</tr>
<tr>
<td>Multistory, low internal loads</td>
<td>2.534E + 04</td>
<td>2.662E + 04</td>
<td>5.05%</td>
</tr>
<tr>
<td>Single story, high internal loads</td>
<td>4.743E + 03</td>
<td>5.149E + 03</td>
<td>8.5%</td>
</tr>
<tr>
<td>Multistory, high internal loads</td>
<td>3.438E + 04</td>
<td>3.580E + 04</td>
<td>4.1%</td>
</tr>
<tr>
<td>Single story, high solar loads</td>
<td>4.57E + 03</td>
<td>5.092E + 03</td>
<td>11.4%</td>
</tr>
<tr>
<td>Multistory, high solar loads</td>
<td>3.372E + 04</td>
<td>3.598E + 04</td>
<td>6.7%</td>
</tr>
</tbody>
</table>

**SUMMARY AND CONCLUSIONS**

Heat balance based calculation of building energy or thermal loads requires either explicit or implicit estimation of the surface film coefficients. Simulation studies, which were performed to estimate the impact of film coefficients on overall room heating and cooling loads, indicated that commonly used film coefficients can result in a daily cooling load error on the order of 10%.

Equations relating room convective heat transfer coefficients to the air change rate of ceiling-ventilated rooms were developed from experimental data. The correlations that were originally developed for ventilative flow rates ranging from 3 to 12 ACH were extended to cover the Spitler data set (15 ACH to 100 ACH). The convective heat transfer coefficients were based on an inlet jet reference temperature, which not only resulted in the lowest experimental uncertainty but also significantly improved the correlations.

Several observations led to the selection of ACH as the correlating parameter for the convective heat transfer coefficient. First, it was observed that the ceiling jet did not detach from the room surfaces over the entire range of experimental parameters. Thus the flow field in the room was roughly similar from 3 to 100 ACH. Second, it was demonstrated that the rate of convective heat transfer was independent of both the velocity and the momentum of the inlet jet. The choice of ACH as the correlating parameter is supported by the physics of the control volume, which requires the overall room convective heat transfer to be proportional to the ventilative flow rate at a constant inlet air temperature.
The experimental investigation also left a number of questions unanswered. First, sensitivity of the correlations to the aspect ratio of the room was not investigated. A related problem involves the interaction of multiple diffusers in a large room. The effect of furnishings and heat sources in rooms also requires additional experimental investigation.

REFERENCES


Cantillo, J. 1990. Air velocity and temperature measurements in a full-scale ventilative cooling research facility. M.S. Thesis, Department of Mechanical and Industrial Engineering, University of Illinois at Urbana-Champaign.


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