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CORRELATIONS FOR CONVECTIVE HEAT TRANSFER FROM ROOM SURFACES*

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ABSTRACT

Correlations of the rate of heat transfer from room surfaces to the enclosed air, based on empirical and analytic examinations of convection in two-dimensional enclosures, have been developed. The heat-transfer data base from which the correlations were derived was generated by a validated numerical-analysis computer program. Simulations were performed for a variety of temperature and geometry boundary conditions. The correlations extracted from this data base express the heat-transfer rate in terms of boundary conditions relating to room geometry and surface temperatures. The correlations are applicable to a class of room configurations with cold and warm surfaces on opposite vertical walls.

In this paper, the numerical-analysis technique is presented, the generalized methodology for developing simplified correlations is described, and correlation results are presented and compared to results using standard ASHRAE techniques. It is shown that the correlation significantly improves the accuracy of heat-transfer rate calculations in buildings.

INTRODUCTION

Of the fundamental heat-transfer processes, convection is the least understood. In contrast to the equations covering conduction and radiation, those governing convective heat and mass transfer in fluids (the Navier-Stokes equations) typically do not have closed solutions even under steady-state conditions. As a result, convection research has usually been limited to experimental work, which characteristically is not amenable to generalizations. Thus, there is a large disparity in the understanding of convective processes in comparison to radiative and conductive phenomena.

The objective of the work reported here is to develop a simple, accurate, and highly generalized set of correlations for the convective heat-fluxes in enclosures typical of rooms in buildings. The technique used in this study consists of performing "numerical experiments" using a recently developed and validated numerical analysis computer program that solves the Navier-Stokes equations in two or three dimensions. The program is used to generate a data base of heat transfer from the surfaces of an enclosure for a variety of temperature and geometry boundary conditions. Although this project examined only a few of the possible combinations of boundary conditions, the results indicate that accurate correlations can be developed. The numerical experiments are far more rapidly performed than their physical counterparts and, assuming that carefully selected validation experiments are performed in conjunction with the analysis, provide a larger amount of accurate data on velocity, fluid temperature, and heat transfer than the best experiments reported in the literature.

This report will outline the significance of surface convection coefficients in the

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building sciences and briefly outline the current state of the art. Later sections will describe the numerical-analysis technique and the methodology used to develop the correlation. Results of the work will be compared with the standard results, using techniques recommended by ASHRAE. Recommendations for future research will also be given together with concluding remarks.

BACKGROUND

Convective processes affect the energy performance of buildings through three mechanisms: (1) coupling between room air and the surfaces to which it is exposed, (2) distribution of thermal energy within and between zones by air circulation, and (3) coupling of the interior air to the external environment through ventilation or infiltration. These processes, all significant, are the subject of ongoing research. The work reported here focuses on the first of these mechanisms.

Heat transfer between room air and the surfaces of the room can be characterized in terms of a surface convection coefficient, h, which describes the thermal conductance between the air and the surface. In contrast to the conductance of a normally insulated building envelope element, the convection coefficient is large and has little influence on the overall heat transfer through the wall. At high conductivity surfaces such as windows, however, the convection coefficient has significant influence* on the heat losses to the environment. Uncertainty in the magnitude of the coefficient leads to potentially large errors in the calculation of thermal gains and losses, i.e., in the thermal load calculation. For this reason, accurate values for the surface convection coefficients are necessary to properly determine the heating and cooling system size and the air-handling system capacity, as well as the thermal capacity of convectively coupled mass in the building.

Heat transfer between room air and exposed surfaces is normally modeled using convection coefficients recommended by ASHRAE (1981) and Lokmanhekim (1976). These coefficients are largely based on experimental research conducted 40 to 50 years ago (ASHRAE 1981). The temperature dependence of the data from these experiments (Wilkes and Peterson 1938, p. 513) disagrees with more recent experimental results (Holman 1976, p.255).

Recently, there has been renewed interest in convective heat-transfer processes in buildings. Though much of the recent convection research does not focus directly on the evaluation of convection coefficients or zone coupling, the research methodology and analytical tools are sufficiently developed to reconsider past estimates of the importance of convective heat-transfer processes in buildings. Bauman et al. ("Convective," 1983) contains a list of such investigations on convective heat-transfer within and between thermal zones in building configurations. In addition to the references cited there, recent work in Japan and Belgium is described below.

Nomura et al. (1981), and Sakàmoto and Matsuo (1980, p.67-72) have developed numerical models of three-dimensional turbulent flow in a mechanically ventilated room. They have compared the predictions of computer simulations based on this model with controlled experiments in full-scale rooms. The main focus of their work has been on the prediction of velocity fields, ventilation efficiency, and mass transport. Experimental research on heat transfer, thermal stratification, and passive and active heating systems in residential buildings has been carried out at Tohoku University; some recent experimental results are described by Hasegawa et al. (1981).

Of particular interest to the present study is the ongoing experimental research in Liege, Belgium, described by Lebrun and Marret (1977, p. 417) and Marret (1981). In these studies, heat transfer and temperature measurements have been made in a full-scale test room to investigate the convective and radiative exchange between interior surfaces of the room. Experiments have been performed for a number of configurations that often include the combination of one cold vertical surface (window) and a heat source (radiator or heated surface). No detailed data are available from these studies to allow a direct comparison of their experimentally determined surface convection coefficients with those predicted by the numerical convection program described here.

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*The convection coefficient constitutes about 80% of the thermal resistance of a single-pane window.

NUMERICAL ANALYSIS TECHNIQUE

Computer programs that solve the full Navier-Stokes equations of motion for airflow in buildings have been developed Gadgil 1979; Gosman et al. 1980, p.316-323; Markatos and Malin 1982, p.63-75). The program used for the research reported in this paper, described in detail by Gadgil 1979, is based on the finite-difference method. Time-dependent differential equations are integrated over the finite number of subvolumes and over each time step to obtain a large number of simultaneous algebraic equations, which are solved for successive time steps until steady-state flow fields are obtained.

The numerical-analysis program is suitable for modeling both natural and forced convection in two and three dimensions, for internal and external flows (Gadgil 1979). In addition, the program can model any combination of obstacles (internal partitions, furniture, building exteriors), heat sources and sinks (space heating and cooling), and velocity sources and sinks (fans, windows). The program presently is limited to simulation of steady laminar flow; a turbulent model is currently being incorporated into the program. The laminar flow calculations have been verified against analysis and detailed experiments performed at Lawrence Berkeley Laboratory and elsewhere (Bauman et al., "Buoyancy," 1980; Shiralkar et al. 1981, p. 1621-1629; Tichy and Gadgil 1982, p.103-110; Gadgil et al. 1983).

Using this program requires specification of the geometric configuration, thermal and velocity boundary conditions, and fluid properties. The air velocity (or its derivatives) must also be specified on all bounding surfaces of the enclosure volume. Internal obstacles, such as partition walls, can be included by specifying zero velocity at appropriate regions within the enclosure. Forced ventilation is dealt with by setting velocity boundary conditions dictated by fan parameters at appropriate locations. The computer simulation predicts the velocities and temperatures throughout the volume of interest and also predicts convective heat-fluxes on all surfaces. This information allows calculation of heat-transfer coefficients as a function of position on all surfaces of the room.

CORRELATION METHODOLOGY

Convective heat-transfer coefficients on the interior surfaces of rooms have been investigated (Bauman et al., "Convective," 1983; Gadgil, Bauman, and Kammerud 1982). It was found that the convection coefficients recommended by ASHRAE (1981) are inconsistent and are in disagreement with the results of recent numerical and experimental research. The current level of uncertainty in convection coefficients may lead to unacceptably large errors in calculating the thermal load in buildings; improved correlations for convective heat-fluxes need to be developed. Because of the complexity of the recirculating natural convection flow of room air, which is influenced by the temperature distribution on all the room surfaces, it is not expected a priori that a single set of correlations, which will accurately predict the convective heat-fluxes on all surfaces for all possible configurations of surface temperatures and air temperatures, can be obtained. It is hoped, however, that by examining a rather small number of generic configurations, and developing appropriate correlations for each, that the vast majority of the combinations of geometric and thermal boundary conditions encountered in buildings will be covered. Furthermore, it is hoped that the correlations that are developed are systematically related to one another and have forms that are tractable to the practitioner.

To develop correlations for convective heat-fluxes in a single room, the convection computer program described earlier was used to examine the sensitivity of surface heat-flux to variations in different parameters governing the flow. These parameters include the bounding surface temperature distributions, the surface areas, and the enclosure aspect ratio (H/L). For simplicity, a two-dimensional room, similar to that described by Bauman et al., "Convective," 1983, was chosen for simulation by the convection program. Experiments (Bohn et al. 1983) have demonstrated that the boundary layers on walls are almost two-dimensional at high Rayleigh number values, even for cubical enclosures. The room has a height, H, of 2.44 m and a length, L, of 3.66 m; each of its four surfaces has been divided into three subsurfaces. The room is shown schematically in figure 1; the individual subsurfaces are identified numerically in this figure and the subsurface areas (per unit depth) are shown. The enclosure volume was discretized with an unevenly spaced 20 x 30 grid with a high density of grid lines near the surfaces for good resolution of the boundary layers. It was verified that a simulation using this 20 x 30 grid gave the same thermal and velocity profiles as obtained with simulations using a 31 x 37 high resolution grid. The 20 x 30 grid was used in all simulations in the interest of computational efficiency.

For the first configuration selected, the hottest surface is located on one vertical wall and the coldest on the opposite vertical wall; this configuration is characteristic of situations often encountered in buildings. Nine simulations were initially performed to study natural convection under the different temperature boundary conditions displayed in table 1. For these nine analyses, only two parameters were varied: the coldest subsurface temperature (T_2) was varied from -6.67°C (20°F) to 15.56°C (60°F), and the hottest subsurface temperature (T_8) was varied from 21.11°C (70°F) to 37.78°C (100°F)*. The temperatures of all other subsurfaces were held constant at 20°C (68°F), as were the subsurface areas and the enclosure aspect ratio. The results from these nine simulations were used to develop a methodology for producing correlations for the convective heat-fluxes from the surface to the air, for this particular configuration.

TABLE 1

Parametric Simulation Description

	Parameter Varied			
Simulation #	Temperature of Cold Wall (T_2) (°C)	(°F)	Temperature of Hot Wall (T_8) (°C)	(°F)
1	-6.67	20	21.11	70
2	-6.67	20	29.45	85
3	-6.67	20	37.78	100
4	4.45	40	21.11	70
5	4.45	40	29.45	85
6	4.45	40	37.78	100
7	15.56	60	21.11	70
8	15.56	60	29.45	85
9	15.56	60	37.78	100

T_i: Temperature of subsurface #i

For all simulations:

$$T_1 = T_3 = T_4 = T_5 = T_6 = T_7 = T_8$$
$$= T_{10} = T_{11} = T_{12} = 20°C\ (68°F)$$

The convective heat-fluxes on all subsurfaces, as predicted by the computer program for the nine initial simulations, are shown in figure 2. The average room air temperatures (T_{air})** calculated by the program are also shown in the figure. It can be seen that subsurfaces 1, 2, 7, and 8 contribute a very large portion (typically on the order of 90%) of the total heat transfer into or out of the room air. For this reason, the data interpretation has focused on these four subsurfaces (henceforth also referred to as C', C, H', and H, respectively, for easy reference), which have been labeled "active" subsurfaces. Successful development of correlations for predicting the heat flux from the active subsurfaces would account for

*This range of temperatures represents the Rayleigh number range of $9 \times 10^9 \leq Ra \leq 7 \times 10^{10}$.

**T_{air} was obtained by taking an average of the air temperature at all the nodes of the grid, weighted with the control-volume surrounding each node.

about 90% of the total heat flux for this particular configuration. The remaining subsurfaces, labeled "inactive," have not been considered for correlation development.

Based on the difference between the temperature of the subsurface and the numerically calculated average room air temperature (ΔT_{sa}), convection coefficients, h, for four active subsurfaces in each of the nine parametric runs were calculated using the standard equation

$$h_j = \frac{q_j}{\Delta T_{sa}} \tag{1}$$

where

q_j = convective heat-flux from subsurface j to the room air.

The computed convection coefficients are shown in figure 3. The large variation in the convection coefficients for any given subsurface is apparent, particularly for subsurfaces C' and H'. Even for the primary heat-transfer subsurfaces C and H, the convection coefficients vary by nearly 50% to 60%.

It is interesting to note that for large values of wall to wall temperature differences (T_C = -6.7°C, T_H = 37.8°C), the convection coefficients for subsurfaces C and H are within a few percent of the constant value (3.08 W/m^2°C) documented in ASHRAE 1981. However, the extremely large value of wall to wall temperature difference (44.5°C) exceeds the temperature differences obtained even under peak load conditions for almost all buildings.

Since the magnitude of the heat flow into a subsurface is determined by the temperature difference between the subsurface and the adjacent air (ΔT_{adj}), convection coefficients calculated from equation 1, using ΔT_{sa} as defined, will often lack a consistent pattern and will, therefore, be difficult to predict using a simple equation. Whenever the sign of ΔT_{sa} is opposite to that of ΔT_{adj}, the convection coefficient calculated according to equation 1 will have a negative sign. For example, the updraft of warm air leaving subsurface H will deposit heat into the cooler subsurface H' despite the fact that ΔT_{sa} is positive. These conditions, which occur in most of the simulations, produce a negative convection coefficient on subsurface H', as seen in figure 3.

Convection coefficients can also be defined using ΔT_{adj} according to equation 2:

$$h_j = \frac{q_j}{\Delta T_{adj}} \tag{2}$$

The computer simulations predicted the air temperature distribution throughout the room for each parametric run. Thus, T_{adj} could be obtained for each subsurface.* The convection coefficients on subsurfaces C' and H', computed according to equation 2, show a much more consistent pattern than those in figure 3. Thus, it appears that a successful prediction of convective heat-fluxes on wall subsurfaces may be possible if an empirical correlation can be developed in terms of an estimated air temperature (T_j') adjacent to the subsurface j, which is calculated as a function of the known areas and temperatures of the subsurfaces in the enclosure. This approach would be useful in computations for building energy analysis since, in these computations, T_{adj} is not available, while surface temperatures and areas are commonly available. T_j' was defined for each active subsurface and the inactive subsurfaces using the following equations:

$$\left.\begin{aligned} T_H' = T_C' = T_I' &= (L_H T_H + L_C T_C + L_{H'} T_{H'} + L_{C'} T_{C'} + L_I T_I)/(L_H + L_C + L_{H'} + L_{C'} + L_I) \\ T_{H'}' &= (L_H T_H + L^u_{H'} T_I)/(L_H + L^u_{H'}) \\ T_{C'}' &= (L_C T_C + L^u_{C'} T_I)/(L_H + L^u_{H'}) \end{aligned}\right\} \tag{3}$$

*T_{adj} was calculated for each active subsurface as the average of the individual temperatures predicted by the program along the grid line immediately outside the boundary layer structure.

where

$$T_j = \text{temperature of subsurface } j$$

$$L_j = \text{area of subsurface } j$$

$$L^u_{H'} = L_9 + L_{10} + L_{11} + L_{12} + L_1$$

$$L^u_{C'} = L_3 + L_4 + L_5 + L_6 + L_7$$

For the various subsurfaces, these adjacent air temperatures, T'_j, have the following physical significance: for the two main heat-transfer subsurfaces, (H and C), and for the inactive subsurfaces (I), this air temperature is simply the area weighted mean of the temperatures of all the subsurfaces. For subsurfaces H' and C', which are immediately downstream of the two primary heat-transfer subsurfaces, the relevant air temperatures are given by an area weighted average of the temperatures of the upstream subsurfaces up to, but excluding, the primary heat-transfer surface farthest from H' or C'. This procedure ensures that the adjacent air temperature thus obtained is weighted towards the temperature of the main heat transfer surface immediately upstream.

The generality of the initial data base (table 1) was increased by performing additional numerical simulations (see table 2). For these simulations, the temperature boundary conditions were identical to those of simulation 5 of table 1. This particular simulation was chosen because it contains the median conditions of the nine initial simulations. The parameters varied were:

- Area of hot subsurface, L_H
- Area of cold subsurface, L_C
- Temperature of inactive subsurfaces, T_I
- Aspect Ratio of the room, (Height/Width)

The above simulations (table 2) were included with the original 9 simulations (table 1) to form an expanded data base of a total of 18 simulations. This expanded data base is thus based on a variation in six independent parameters for the configuration studied.

The analysis determined that in the interest of generality, the heat fluxes from each of the active surfaces should be first represented in terms of a (dimensionless) Nusselt number (equation 4):

$$Nu_i = q_i\, H/[(T_H - T_C)k] \tag{4}$$

where

$$i = \text{subsurface number}$$

$$k = 0.0258 \text{ W/m°C for air at NTP}$$

The Nusselt numbers, Nu_i, were then defined as a function of five (dimensionless) Rayleigh numbers, Ra_j, according to the following equation:

$$Nu_i = K_{iH} \frac{H}{L_H} Ra_H^{\frac{1}{4}} + K_{iC} \frac{H}{L_C} Ra_C^{\frac{1}{4}} + K_{iH'} \frac{H}{L_{H'}} Ra_{H'}^{\frac{1}{4}} + K_{iC'} \frac{H}{L_{C'}} Ra_{C'}^{\frac{1}{4}} + K_{iI} \frac{H}{L_I} Ra_I^{\frac{1}{4}} \tag{5}$$

where the twenty coefficients K_{ij} (i = H, C, H' and C'; j = H, C, H', C' and I) were obtained by using the method of least squares to fit equation 5 to the data base and the Rayleigh numbers are defined according to equation 6 below:

$$Ra_j = P|T_j - T_j'|L_j^3 \tag{6}$$

<u>where</u>

T_j' is defined in equation 3

$P = g\beta Pr/\nu^2 = 1.02 \cdot 10^8\ °C^{-1}m^{-3}$ for air at NTP

j = H, C, H', C' or I

TABLE 2

EXPANDED DATA BASE SIMULATION DESCRIPTION$

Base Case#	Parameter Varied
	Area of Hot Subsurface, A_H
T_C A_H = 1.01 m² T_H	T_C A_H = 1.72 m² T_H (simulation #10); T_C A_H = 2.44 m² T_H (simulation #11)
	Area of Cold Subsurface, A_C
T_C A_C = 1.26 m² T_H	T_C A_C = 1.72 m² T_H (simulation #12); T_C A_C = 2.44 m² T_H (simulation #13)
	Temperature of Inactive Subsurface, T_I
T_C T_I = 20.0 °C T_H	T_C T_I = 17.2 °C T_H (simulation #14); T_C T_I = 22.8 °C T_H (simulation #15)
	Enclosure Aspect Ratio, H/L
T_C H/L = 0.67 T_H	T_C H/L = 1.0 T_H (simulation #16); T_C H/L = 0.33 T_H (simulation #17); T_C H/L = 0.25 T_H (simulation #18)

$For all simulations: T_H = 29.45 °C, T_C = 4.45 °C

#Base case = simulation #5, Table 1.

In equation 5 the heat transfer from any active surface is given by a linear superposition of the contributions from all the subsurfaces. Each contribution is proportional to the 1/4th power of the Rayleigh number for each subsurface and to the ratio of the room height to the subsurface length. The 1/4 power was used since this is the Ra dependence of Nu ($Nu \propto Ra^{1/4}$)

for laminar natural convection along a vertical flat plate (Holman 1976). The length ratio (H/L_j) was included in each term of equation 5 to account for the length of each particular subsurface. Equation 6 shows that the Rayleigh number for each subsurface is proportional to the various fluid properties (lumped into a constant, $P = 1.02 \times 10^8$), the cube of the length of each subsurface, and the temperature difference between the surface and the adjacent air with which it thermally interacts.

RESULTS

The correlation constants obtained from the least squares fit are shown in table 3 together with equation 5. The coefficients K_{ij} in the above correlation demonstrate the symmetry demanded by the problem. For example, the primary heat-transfer surfaces, H and C, show an equal but opposite dependence on their respective Rayleigh numbers, as well as on the Rayleigh numbers of the opposing heat-transfer surface (i.e., the coefficients K_{HH} and K_{CC} are equal in magnitude and opposite in sign to each other, as are the pairs of coefficients K_{HC} and K_{CH}, $K_{HC'}$ and $K_{CH'}$, etc.). Similar symmetry is seen in the correlation coefficients for the surfaces H' and C' as well.

The method of using the correlation is described in detail in Appendix A. A sample calculation is presented in Appendix B.

TABLE 3

Correlation Form

$$Nu_i = K_{iH} \frac{H}{LH} Ra_H^{\frac{1}{4}} + K_{iC} \frac{H}{L_C} Ra_C^{\frac{1}{4}} + K_{iH'} \frac{H}{L_{H'}} Ra_{H'}^{\frac{1}{4}} + K_{iC'} \frac{H}{L_{C'}} Ra_{C'}^{\frac{1}{4}} + K_{iI} \frac{H}{L_i} Ra_I^{\frac{1}{4}} \qquad (5)$$

i	K_{iH}	K_{iC}	$K_{iH'}$	$K_{iC'}$	K_{iI}
H	0.7253	-0.4062	-0.0650	0.0347	-0.1017
C	0.4062	-0.7253	-0.0347	-0.0650	-0.1017
H'	-0.4049	0.3997	-0.1256	-0.0918	0.1427
C'	-0.3997	0.4049	0.0918	0.1256	0.1427

The heat-flux predictions for all four active subsurfaces calculated from equations 4, 5 and 6 are compared against the numerically obtained results of the 18 numerical simulations in figure 4. The solid 45°-line represents the line of perfect agreement between the two calculation techniques. This figure shows that calculations based on the present correlation compare well with the numerical predictions of the convection program for all four active subsurfaces. The points plotted in figure 4 have a root mean square (RMS) deviation from the 45° line of only 3.16 W/m^2.

The relative and absolute errors in heat-flux predictions from equation 5 (in comparison with the numerically obtained results) are shown in table 4. The relative errors are small (typically < 5%) even when the magnitudes of the heat fluxes are small (cf. simulation 7), showing the applicability of the correlation for a wide range of heat-flux magnitudes. Similarly, the correlation is seen to be insensitive to each of the six parameters varied. The range of variation of these parameters is indicative of the wide range of validity of the correlation.

Furthermore, if one considers simulations 1, 4 and 7 from table 1, in which the temperature of the hot surface is 21.1°C, the magnitudes of the heat fluxes on subsurface H' are quite

TABLE 4

Errors in the Prediction of Convective Heat Flows at the "Active" Surfaces Using LBL Correlation

	Errors in Total Convective Heat Flow From Wall to Air		Errors in Total Convective Heat Flow from Air to Wall	
Simulation #	Absolute (W)	Relative (%)	Absolute (W)	Relative (%)
1	2.96	5.6	2.95	3.3
2	0.62	0.9	1.79	1.8
3	3.29	3.1	1.05	0.9
4	0.41	1.6	0.83	1.9
5	1.54	3.2	0.72	1.2
6	1.76	2.4	1.81	2.4
7	0.08	1.2	0.35	3.4
8	0.23	0.9	0.79	3.5
9	2.12	4.1	0.52	1.4
10	1.36	3.1	2.36	3.7
11	1.09	2.8	2.06	3.5
12	0.93	1.8	4.73	9.8
13	1.11	3.0	3.21	7.3
14	1.33	3.2	0.45	0.8
15	0.27	0.6	1.53	2.4
16	0.08	0.2	1.25	2.1
17	0.10	0.2	0.77	1.3
18	1.66	3.4	0.71	1.2

small (see figure 2). Under these conditions, one is on the outer limits of the definition of this configuration as "one hot and one cold surface on opposite vertical walls with the remainder of the surfaces at an intermediate temperature." The present correlation predicts the convective heat-fluxes even under these conditions reasonably well.

Two methods* of calculating convective heat-flux recommended by ASHRAE are similarly compared with the numerically obtained results in figure 5. The convective heat-fluxes in figure 5a calculated using the ASHRAE constant convection coefficients according to equation 7 below:

$$q_{ASHRAE1} = h \cdot T_{sa} \tag{7}$$

where

$h = 3.08\ (W/m^2\,°C)$ for vertical surfaces,

$= 4.04\ (W/m^2\,°C)$ for horizontal surfaces with heat flow upward.

$= 0.95\ (W/m^2\,°C)$ for horizontal surfaces with heat flow downward.

The convective heat-fluxes in figure 5b are calculated using the ASHRAE correlations for turbulent convection coefficients according to the equation below:

$$q_{ASHRAE2} = h \cdot \Delta T_{sa} \tag{8}$$

where

$h = (1.31)(\Delta T_{sa})^{0.33}$ for vertical surfaces

$= (1.52)(\Delta T_{sa})^{0.33}$ for horizontal surfaces with heat flow upward.

For horizontal surfaces with heat flow downward, no expression for h is given for turbulent convection. The laminar correlation given by ASHRAE was used:

$$h = (0.51)(\ T_{sa}/L)^{0.25}$$

The ASHRAE correlations provide a fairly good prediction of the heat flux to the room air from the cold and hot subsurfaces (C and H), but they have large errors for the other two subsurfaces (C' and H'). The ASHRAE correlations, in fact, often predict the wrong direction of heat flow for subsurface H'. The RMS deviations of the points in figure 5 from the 45°-line are 16.1 W/m^2 and 16.8 W/m^2 respectively. It should be noted that these numbers are of the same order of magnitude as the heat fluxes on most subsurfaces. However, most of the aggregated deviation is contributed by the inability of the standard ASHRAE correlations to meaningfully represent the heat transfer for the subsurfaces (H' and C') immediately downstream of the heated and cooled subsurfaces (H and C). A comparison of figures 4 and 5 shows that the predictions from equations 4, 5 and 6 give substantially better agreement with the data base than do the ASHRAE calculations, particularly for subsurfaces H' and C'.

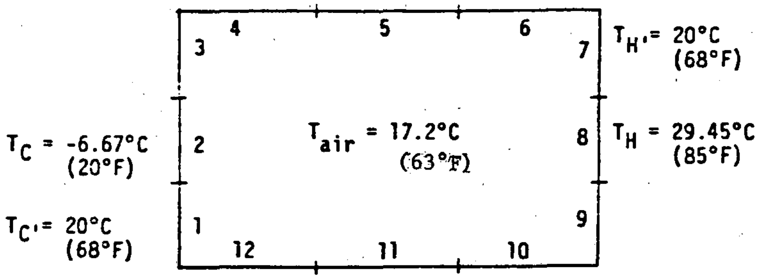
In table 5, the convective heat-flows from the active subsurfaces to the room air, as calculated by the present correlations, are compared in detail with the two ASHRAE calculations. The comparison has been made for one particular set of boundary conditions (simulation 2), shown in the accompanying diagram. The average room air temperature (T_{air} = 17.2°C) computed by the convection program has been used to calculate the surface-to-air temperature difference (ΔT_{sa}), which in turn has been used in the ASHRAE correlations. A measure of the imbalance in the total heat flow in the room, Q_{net}, can be obtained by calculating the total of convective heat-flows from each of the 12 subsurfaces to the room air. This quantity, although limited to the convective component of heat transfer within the room, has important implications in terms

*The ASHRAE constant convection coefficient for a vertical surface is derived from table 1, page 23.12, ASHRAE Handbook -- 1981 Fundamentals Volume by subtracting out the radiative component of the total surface heat-transfer coefficient. The radiative component is based on a 5.6°C (10°F) surface-to-surroundings temperature difference, which is not always typical for real buildings. In addition, it is noticed that the constant convection coefficient is representative of a 13°C (23°F) surface-to-air temperature difference in order to be consistent with the temperature-dependent convection coefficient for turbulent flow. The ASHRAE temperature dependent convection coefficient for turbulent flow is recommended for use with large plates (typical for buildings) and is taken from table 5, page 2.12, ASHRAE Handbook -- 1981 Fundamentals Volume.

of the accuracy of building load calculations. For the steady-state conditions of this configuration, the convection program properly computes Q_{net} to be very nearly zero, as it should be. As seen in table 5, the present correlation, combined with two convection coefficients for the inactive subsurfaces,* produces a room energy balance to within 3.5% of the desired result. Both ASHRAE calculation methods, however, demonstrate large errors in the energy balance calculation and are not even mutually consistent regarding the direction of the excess energy flow.

TABLE 5

Comparison of Natural Convection Heat Transfer Calculations (Simulation 2)



$T_1 = T_3 = T_4 = T_5 = T_6 = 20°C$

$T_7 = T_9 = T_{10} = T_{11} = T_{12} = 20°C$

$T_2 = -6.67°C, \; T_8 = 29.45°C$

Method of Calculation ($Q_j = q_j \cdot L_j$)	Convective Heat Flow (W) From Wall to Air for Subsurfaces C', C, H', H				Imbalance in Total Heat Flow, Q_{net}	
	$Q_{C'}$	Q_C	$Q_{H'}$	Q_H	(W)**	(%)†
LBL Convection Program	37.8	-94.8	-6.9	34.1	0.3	0.3
LBL Correlation	35.9	-92.6	-8.4	33.6	3.5	3.5
ASHRAE Const. Conv. Coeff. $q_j' = 3.08 \cdot \Delta T$	6.2	-92.6	6.2	29.4	-10.7	10.9
ASHRAE Turb. Conv. Coeff. $q_j = 1.31 \cdot \Delta T_{sa}^{1.33}$	3.7	-112.3	3.7	26.3	-44.8	-49.8

** $\left(\sum_{j=1}^{12} Q_j \right)$ † $\left[\left(\sum_{j=1}^{12} Q_j \right) \Big/ \left(\frac{1}{2} \sum_{j=1}^{12} |Q_j| \right) \right]$

CONCLUDING REMARKS

This report summarizes the results of attempts to develop improved correlations for convective heat-fluxes at the internal surfaces of an enclosure. Correlations have been obtained by

*The present correlations have only been developed for the active subsurfaces. In order to calculate a value for Q_{net}, two convection coefficients were estimated for the inactive subsurfaces along the ceiling and floor, respectively. The four convection coefficients for subsurfaces 3, 4, 5, and 6, as computed by the convection program were averaged together to yield a single value of 0.1 $W/m^2°C$, supporting the classification of these subsurfaces as being inactive. Similarly, for subsurfaces 9, 10, 11, and 12 a single average convection coefficient was computed to be 2.7 $W/m^2°C$.

analyzing results from numerical simulations of the natural convective process in an enclosure described by a two-dimensional configuration having the hottest and coldest surfaces on opposite vertical walls--a situation commonly encountered in buildings. The correlations allow prediction of the major convective heat-fluxes in any situation that can be approximated by the schematic shown in figure 1. The correlations have been shown to provide accurate calculation of convective fluxes for a range of enclosure geometries and surface temperatures. In particular, the correlations reproduce the variability of convection coefficients on the primary heat-transfer surfaces and on surfaces downstream of the primary surfaces. In order to achieve this generality the correlations have necessarily incorporated information specific to the fluid mechanics of the problem being characterized and have therefore become considerably more complex than alternate calculation techniques.

While currently available calculation techniques may be more accessible, they do not contain enough information to properly represent heat transfer in the enclosure, especially for surfaces in proximity to those that are driving the convection process.

The validity of the present correlation is limited to the particular configuration for which the numerical simulations were performed. However, the methodology in this report is quite general and can be appligd to other configurations of hot and cold surface positions, as well as to three-dimensional situations. The correlations obtained for other configurations may not be identical in form to those presented here.

An area needing careful consideration in the future is the prescription for identifying a given real situation with the single most appropriate configuration from the complete set of configurations for which correlations will be published. It is evident that in each and every case, identifying the one hottest and the one coldest subsurface may not be possible; all the subsurfaces may have temperatures close to one another. In this case, all the different applicable correlations must predict similar results. In other words, the correlations, if properly formulated, must give predictions that continuously and smoothly vary over all the possible subsurface temperatures, spanning a number of configurations.

The numerical-analysis technique used in this study has been compared with experimentally obtained flow patterns, temperature fields, and wall heat-fluxes for the configuration considered in this report. These comparisons have shown agreement usually within a few percent, providing one confidence in the computer program as a device for quickly performing "numerical experiments" for a variety of conditions within the domain of this configuration. It is necessary, in proceeding to obtain correlations for significantly different configurations, to reconfirm the validity of the computer program in each case, using experimental data for that configuration.

One caution is in order relating to the use of the preliminary correlation reported here: experimental evidence exists showing steady, laminar flow in enclosures for the present configuration, even at the high Rayleigh number (Ra) values that have been considered here. However, almost all the experiments in this regard have used fluids in the Prandtl number (Pr) range of 2.5 and higher. Since it is known that the Ra number for the onset of turbulence is decreased for lower Pr number fluids (Pr of air at room temperature is 0.71), the assumption of steady laminar flow requires careful experimental examination. Similarly, the Ra number for the onset of turbulence is influenced by the configuration, i.e., the temperature distribution on the subsurfaces, and this, again, needs careful experimentation. The computer program is unable to determine the Ra number for the onset of turbulence, and this information must be experimentally obtained for each configuration.

The results presented in this report indicate that predicting convective heat-transfer in rooms using general correlations is possible. The success achieved for the one case considered is encouraging, and leads to the expectation that a similar approach will yield successful predictions in other cases as well. The difficulties and uncertainties discussed here certainly exist; however, they are amenable to careful experimentation and analysis.

ACKNOWLEDGMENTS

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NOMENCLATURE

g = acceleration due to gravity, (m/sec^2)

Gr = Grashof number, $g\beta(T_H - T_C)H^3/\nu^2$

h_j = convective heat transfer coefficient at subsurface j, ($W/m^2 °C$)

H = enclosure height, (m)

k = thermal conductivity, ($W/m^2 °C$)

K_{ij} = correlation coefficient

L = enclosure length, (m)

L_j = length,(m), or area, per unit depth (m^2) of subsurface j

L_j^u = length, (m), or area per unit depth,(m^2), of active subsurfaces upstream from subsurfaces j

Nu_j = Nusselt number, at subsurface j, $q_j H/[(T_H - T_C)k]$

Pr = Prandtl number, ν/α

$q_{ASHRAE1}$ = convective heat-flux as calculated by equation 7, (W/m^2)

$q_{ASHRAE2}$ = convective heat-flux as calculated by equation 8 (W/m^2)

$q_{correlation}$ = convective heat-flux calculated using equations 4 and 5, (W/m^2)

q_j = convective heat-flux from subsurface j to room air (W/m^2)

Q_j = convective heat-flow from subsurface j to room air, (W)

Ra = Rayleigh number, Gr Pr

Ra_j = Rayleigh number for subsurface j, defined in equation 6

T_{adj} = temperature of room air immediately adjacent to a subsurface, (°C)

T_{air} = average room air temperature, (°C)

T_j = temperature of subsurface j, (°C)

T_j' = empirically correlated temperature of room air immediately adjacent to subsurface j, (°C)

ΔT_{adj} = $T_j - T_{adj}$, (°C)

ΔT_{sa} = $T_j - T_{air}$, (°C)

α = thermal diffusivity, (m^2/sec)

β = coefficient of thermal expansion, ($°C^{-1}$)

ν = kinematic viscosity, (m^2/sec)

Subscripts:

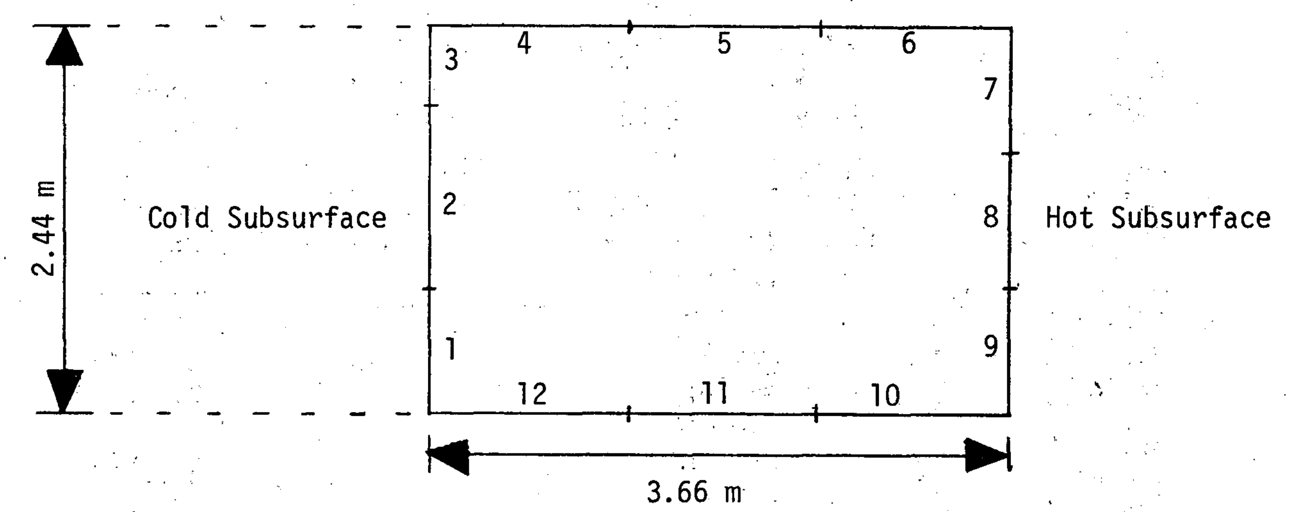
C' = subsurface 1

C = subsurface 2

H' = subsurface 7

H = subsurface 8

I = subsurfaces 3, 4, 5, 6, 9, 10, 11, and 12



Aspect Ratio (height/length) = 0.67

Subsurface #	1	2	3	4	5	6	7	8	9	10	11	12
Area* (m^2)	0.72	1.26	0.46	1.28	1.10	1.28	0.72	1.01	0.72	1.28	1.10	1.28

*Area per unit depth

FIGURE 1

Room description

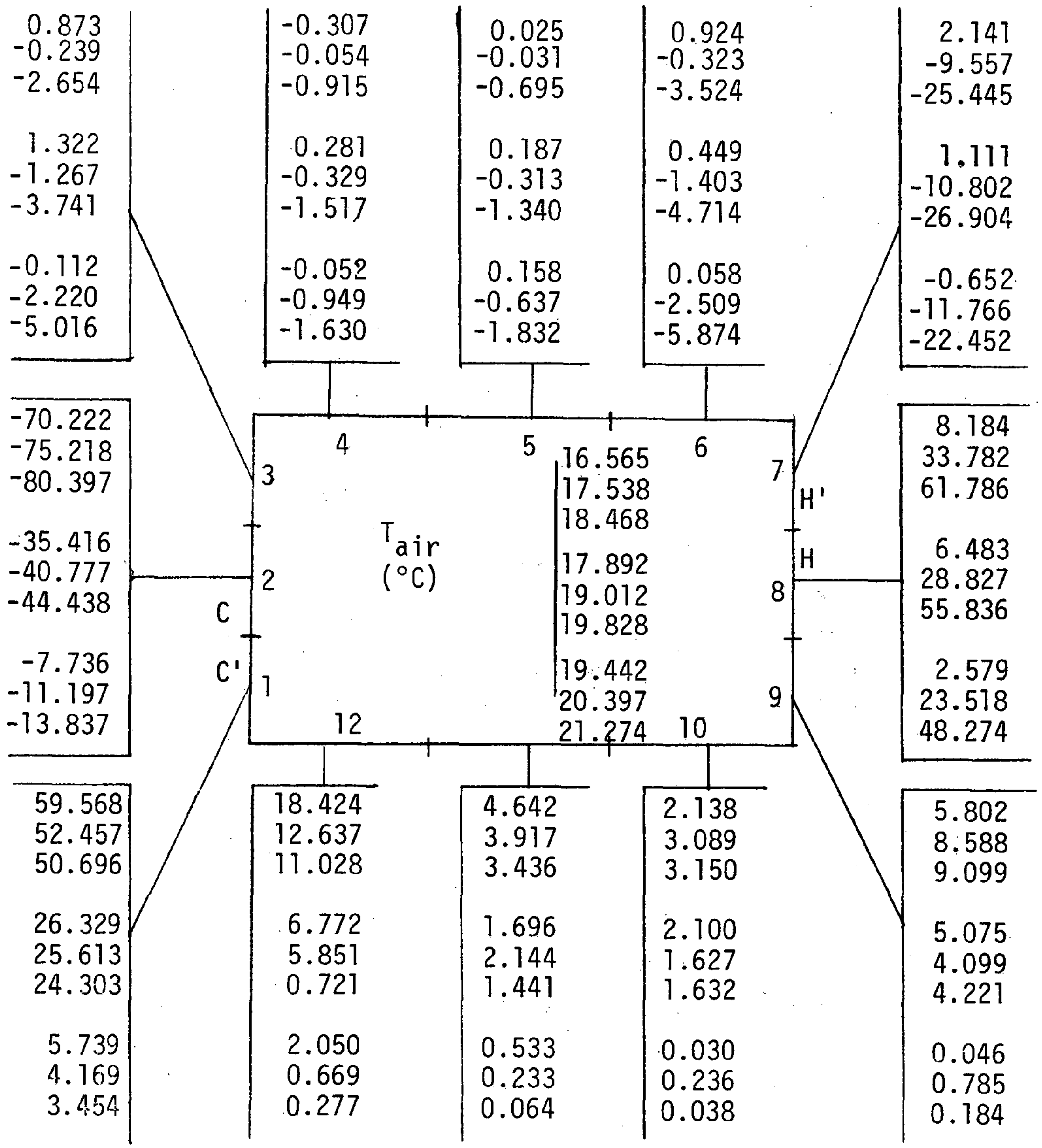


FIGURE 2

Numerically calculated convective heat fluxes (W/m^2) from the interior subsurfaces to the room air.

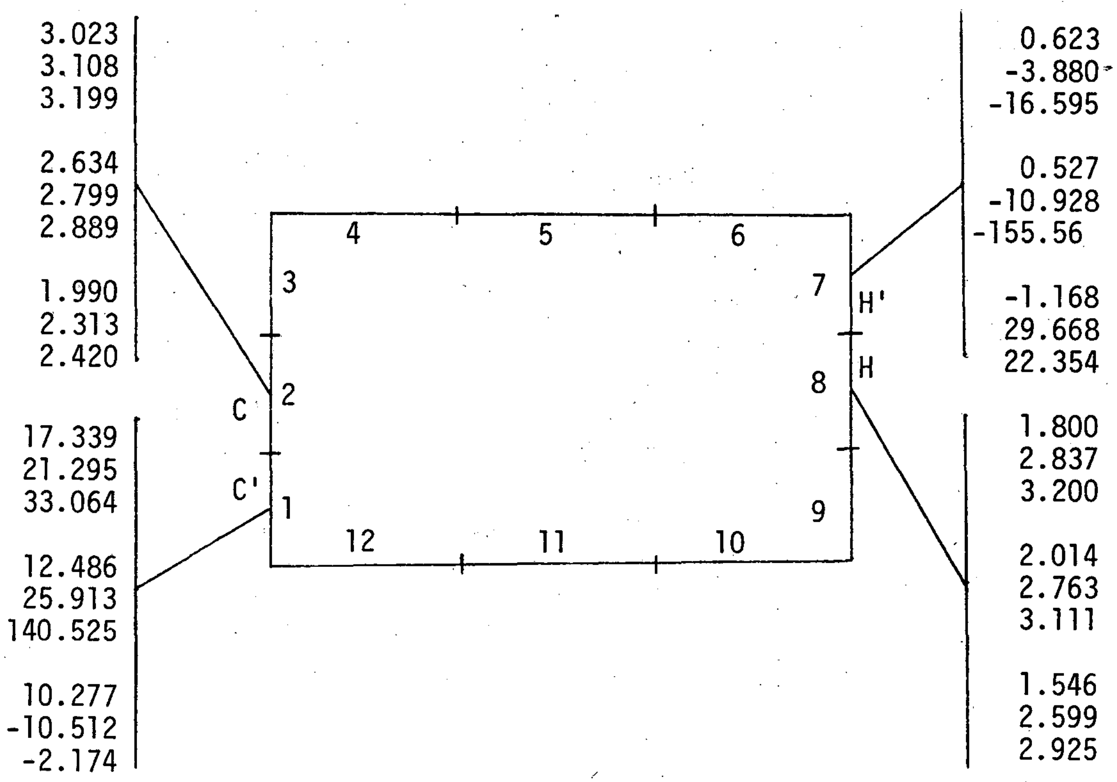


FIGURE 3

Calculated convection coefficients ($W/m^2 {}^\circ C$) from the interior subsurfaces to the room air

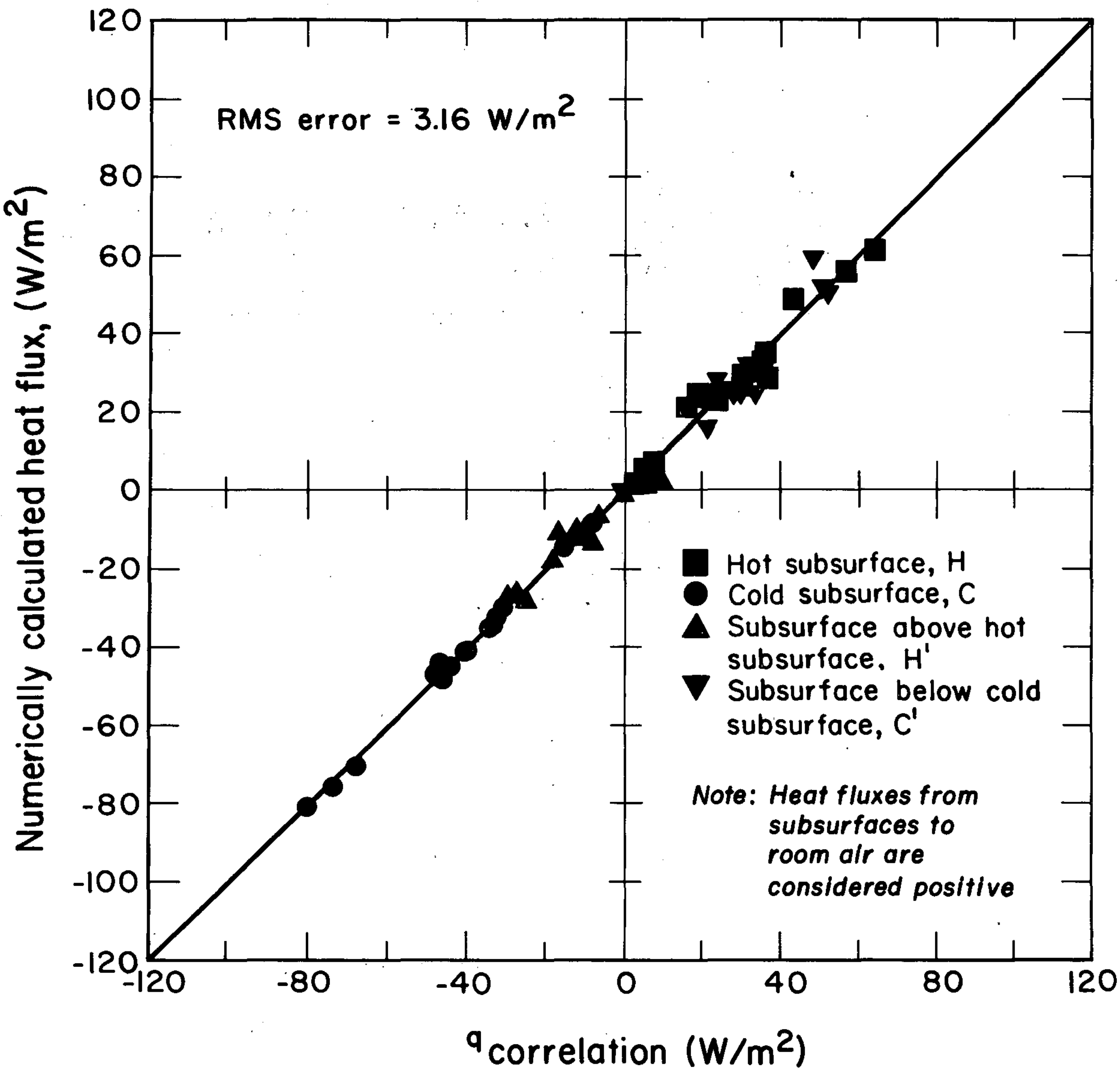


FIGURE 4

Comparison of present heat flux predictions

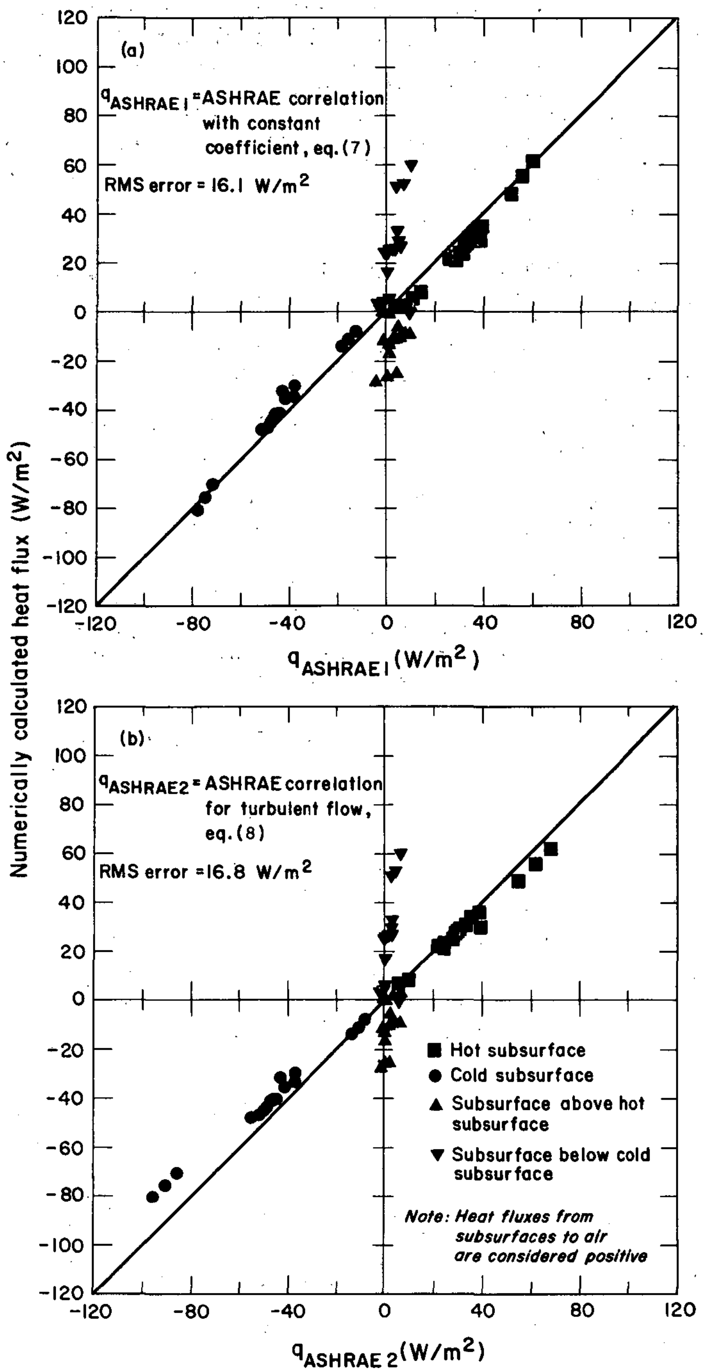


FIGURE 5

Comparison with ASHRAE heat flux predictions

APPENDIX A:

CORRELATION DESCRIPTION

To apply the correlation for convective heat transfer from room surfaces, an appropriate room must be selected. For the present correlation this is limited to rectangular rooms whose surface temperature distribution can be characterized by a warm surface (H) on one wall, a cooler surface (C) on the opposite wall, and the remaining surfaces at some intermediate temperature (Fig. A1). The correlation is capable of predicting the convective heat flux (W/m^2) to/from the room air for the two major heat transfer surfaces H and C, as well as for surfaces H' and C', the two vertical surfaces immediately downstream from surfaces H and C.

The correlation is suitable for use with:

(1) a warm surface (H) having a temperature in the range 21.1°C < T_H < 37.8°C and an area in the range 41% to 100% of the room height;

(2) a cool surface (C) having a temperature in the range -6.7°C < T_C < 15.6°C and an area in the range 52% to 100% of the room height;

(3) inactive surfaces (surfaces 3-6 and 9-12 in Fig. A1) having a temperature in the range 17.2°C < T_I < 22.8°C; and

(4) rectangular rooms having an aspect ratio in the range $0.25 \leq H/L \leq 1.0$ and whose convective heat transfer processes can be approximated as being largely two-dimensional in nature.

For room configurations which fall within the ranges of temperature and geometry specified above, the correlation will predict convective heat fluxes for individual surfaces to within about 3 W/m^2. For temperature and geometry conditions which fall outside of these ranges, the accuracy of the present correlation is not known.

Outlined below is a step-by-step procedure which should be followed when using the correlation.

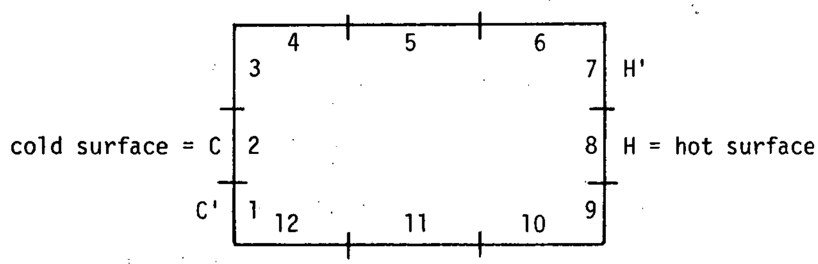


Figure A1. Room Description

Step 1a: Compute surface lengths (m) (surface length is equivalent to area per unit depth).

$$L_H = L_8$$

$$L_C = L_2$$

$$L_{H'} = L_7$$

$$L_{C'} = L_1$$

$$L_I = (L_3 + L_4 + L_5 + L_6) + (L_9 + L_{10} + L_{11} + L_{12})$$

Step 1b: Compute lengths (m) of surfaces upstream from H' and C' (used in equation 3). For a further explanation see page 6.

$$L^u_{H'} = L_1 + L_9 + L_{10} + L_{11} + L_{12}$$

$$L^u_{C'} = L_3 + L_4 + L_5 + L_6 + L_7$$

Step 2: Compute surface temperatures (°C).

$$T_H = T_8$$

$$T_C = T_2$$

$$T_{H'} = T_7$$

$$T_{C'} = T_1$$

$$T_I = T_3 = T_4 = T_5 = T_6 = T_9 = T_{10} = T_{11} = T_{12}$$

Step 3: Compute estimated air temperatures (°C) adjacent to the surfaces.

$$T'_H = T'_C = T'_I = \frac{(L_H \cdot T_H + L_C \cdot T_C + L_{H'} \cdot T_{H'} + L_{C'} \cdot T_{C'} + L_I \cdot T_I)}{(L_H + L_C + L_{H'} + L_{C'} + L_I)}$$

$$T'_{H'} = \frac{(L_H \cdot T_H + L^u_{H'} \cdot T_I)}{(L_H + L^u_{H'})} \tag{3}$$

$$T'_{C'} = \frac{(L_C \cdot T_C + L^u_{C'} \cdot T_I)}{(L_C + L^u_{C'})}$$

Step 4: Compute Rayleigh numbers for each surface ($P = g\beta Pr/\nu^2 = 1.02 \times 10^8$, $°C^{-1}m^{-3}$).

$$Ra_H = P|T_H - T'_H|L_H{}^3$$

$$Ra_I = P|T_C - T'_C|L_C$$

$$Ra_{H'} = P|T_{H'} - T'_{H'}|L_{H'}3 \tag{6}$$

$$Ra_{C'} = P|T_{C'} - T'_{C'}|L_{C'}{}^3$$

$$Ra_C = P|T_I - T'_I|L_I{}^3$$

Step 5: Compute Nusselt numbers for each surface (H = room height).

$$Nu_i = K_{iH} \frac{H}{L_H} Ra_H^{1/4} + K_{iC} \frac{H}{L_C} Ra_C^{1/4} + K_{iH'} \frac{H}{L_{H'}} Ra_{H'}^{1/4} + K_{iC'} \frac{H}{L_{C'}} Ra_{C'}^{1/4} + K_{iI} \frac{H}{L_I} Ra_I^{1/4} \qquad (5)$$

i	K_{iH}	K_{iC}	$K_{iH'}$	$K_{iC'}$	K_{iI}
H	0.7253	-0.4062	-0.0650	0.0347	-0.1017
C	0.4062	-0.7253	-0.0347	-0.0650	-0.1017
H'	-0.4049	0.3997	-0.1256	-0.0918	0.1427
C'	-0.3997	0.4049	0.0918	0.1256	0.1427

Step 6: Compute convective heat flux (W/m^2) for each active surface.

$$q_i = Nu_i (T_H - T_C) k/H$$

where

$$i = H, C, H', \text{and } C'$$

$$k = 0.0258 \text{ W/m}\cdot{}^\circ\text{C for air at NTP} .$$

APPENDIX B:

SAMPLE CALCULATION

Consider the room shown in Figure B1 having a warm 2.4 m high wall maintained at 30.0°C, and a cool 1.0 m high window maintained at 10.0°C on the opposite wall. The remaining surfaces have a temperature of 20.0°C. Other dimensions of the room surfaces are indicated in the figure.

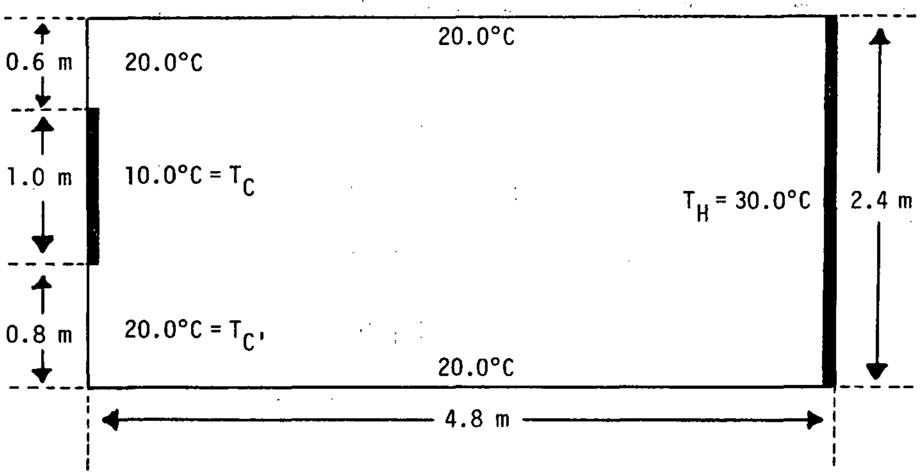


Figure B1. Sample Calculation Room Description

In the following calculations refer to Figure A1 for surface numbers.

Step 1a: Compute surface lengths.

$$L_H = L_8 = 2.4 \text{ m}$$

$$L_C = L_2 = 1.0 \text{ m}$$

$$L_{H'} = L_7 = 0.0 \text{ m}$$

$$L_{C'} = L_1 = 0.8 \text{ m}$$

$$L_I = (L_3 + L_4 + L_5 + L_6) + (L_9 + L_{10} + L_{11} + L_{12})$$

$$= (0.6 + 4.8) + (0 + 4.8)$$

$$= 10.2 \text{ m.}$$

Step 1b: Compute lengths of surfaces upstream from H' and C'.

$$L^u_{C'} = L_3 + L_4 + L_5 + L_6 + L_7$$

$$= 0.6 + 4.8 + 0.0$$

$$= 5.4 \text{ m.}$$

$L^u_{H'}$ is not needed because $L_{H'} = 0$.

Step 2: Compute surface temperatures.

$$T_H = T_8 = 30.0°C$$

$$T_C = T_2 = 10.0°C$$

$$T_{H'} = T_7 \quad \text{not applicable}$$

$$T_{C'} = T_1 = 20.0°C$$

$$T_I = T_3 = T_4 = T_5 = T_6 = T_9 = T_{10} = T_{11} = T_{12} = 20.0°C$$

Step 3: Compute estimated air temperatures adjacent to the surfaces.

$$T'_H = T'_C = T'_I = \frac{(L_H T_H + L_C T_C + L_{H'} T_{H'} + L_{C'} T_{C'} + L_I T_I)}{(L_H + L_C + L_{H'} + L_{C'} + L_I)}$$

$$= \frac{[(2.4)(30.0) + (1.0)(10.0) + 0 + (0.8)(20.0) + (10.2)(20.0)]}{(2.4 + 1.0 + 0 + 0.8 + 10.2)}$$

$$= 21.0°C$$

$$T'_{C'} = (L_C T_C + L^u_{C'} T_I)/(L_C + L^u_{C'})$$

$$= [(1.0)(10.0) + (5.4)(20.0)]/(1.0 + 5.4)$$

$$= 18.4°C$$

$T'_{H'}$ not applicable.

Step 4: Compute Rayleigh numbers.

$$Ra_H = P|T_H - T'_H|L_H{}^3$$
$$= 1.02 \times 10^8|30.0 - 21.0|(2.4)^3$$
$$= 1.27 \times 10^{10}$$

$$Ra_C = P|T_C - T'_C|L_C{}^3$$
$$= 1.02 \times 10^8|10.0 - 21.0|(1.0)^3$$
$$= 1.12 \times 10^9$$

$$Ra_{H'} = P|T_{H'} - T'_{H'}|L_{H'}{}^3$$
$$= 0$$

$$Ra_{C'} = P|T_{C'} - T'_{C'}|L_{C'}{}^3$$
$$= 1.02 \times 10^8|20.0 - 18.4|(0.8)^3$$
$$= 8.36 \times 10^7$$

$$Ra_I = P|T_I - T'_I|L_I{}^3$$
$$= 1.02 \times 10^8|20.0 - 21.0|(10.2)^3$$
$$= 1.08 \times 10^{11}$$

Step 5: Compute Nusselt numbers.

$$Nu_H = K_{HH}(H/L_H)Ra_H^{1/4} + K_{HC}(H/L_C)Ra_C^{1/4} + K_{HH'}(H/L_{H'})Ra_{H'}^{1/4} + K_{HC'}(H/L_{C'})Ra_{C'}^{1/4} + K_{HI}(H/L_I)Ra_I^{1/4}$$
$$= (.7253)(2.4/2.4)(1.27 \times 10^{10})^{1/4} + (-.4062)(2.4/1.0)(1.12 \times 10^9)^{1/4} + 0$$
$$+ (.0347)(2.4/0.8)(8.36 \times 10^7)^{1/4} + (-.1017)(2.4.10.2)(1.08 \times 10^{11})^{1/4}$$
$$= 243 - 178 + 0 + 10 - 14$$
$$= 61$$

$$Nu_C = K_{CH}(H/L_H)Ra_H^{1/4} + K_{CC}(H/L_C)Ra_C^{1/4} + K_{CH'}(H/L_{H'})Ra_{H'}^{1/4} + K_{CC'}(H/L_{C'})Ra_{C'}^{1/4} + K_{CI}(H/L_I)Ra_I^{1/4}$$
$$= (.4062)(2.4/2.4)(1.27 \times 10^{10})^{1/4} + (-.7253)(2.4/1.0)(1.12 \times 10^9)^{1/4} + 0$$
$$+ (.0650)(2.4/0.8)(8.36 \times 10^7)^{1/4} + (-.1017)(2.4/10.2)(1.08 \times 10^{11})^{1/4}$$
$$= 136 - 318 + 0 + 19 - 14$$
$$= -177$$

$$Nu_{H'} = 0$$

$$
\begin{aligned}
Nu_{C'} &= K_{C'H}(H/L_H)Ra_H^{1/4} + K_{C'C}(H/L_C)Ra_C^{1/4} + K_{C'H'}(H/L_{H'})Ra_{H'}^{1/4} + K_{C'C'}(H/L_{C'})Ra_{C'}^{1/4} + K_{C'I}(H/L_I)Ra_I^{1/4} \\
&= (-.3997)(2.4/2.4)(1.27 \times 10^{10})^{1/4} + (.4049)(2.4/1.0)(1.12 \times 10^9)^{1/4} + 0 \\
&\quad + (.1256)(2.4/0.8)(8.36 \times 10^7)^{1/4} + (.1427)(2.4/10.2)(1.08 \times 10^{11})^{1/4} \\
&= -134 + 178 + 0 + 36 + 19 \\
&= 99
\end{aligned}
$$

<u>Step 6</u>: Compute convective heat flux

$$
\begin{aligned}
q_H &= Nu_H(T_H - T_C)k/H \\
&= (61)(30.0 - 10.0)(.0258)/(2.4) \\
&= 13.1 \text{ W/m}^2 \qquad \text{(from surface to air)}
\end{aligned}
$$

$$
\begin{aligned}
q_C &= Nu_C(T_H - T_C)k/H \\
&= (-177)\ (30.0 - 10.0)(.0258)/(2.4) \\
&= -38.1 \text{ W/m}^2 \qquad (38.1 \text{ W/m}^2 \text{ from air to surface})
\end{aligned}
$$

$$
q_{H'} = 0
$$

$$
\begin{aligned}
q_{C'} &= Nu_{C'}(T_H - T_C)k/H \\
&= (99)(30.0 - 10.0)(.0258)/(2.4) \\
&= 21.3 \text{ W/m}^2 \qquad \text{(from surface to air)}
\end{aligned}
$$

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