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CONVECTION COEFFICIENTS AT BUILDING SURFACES

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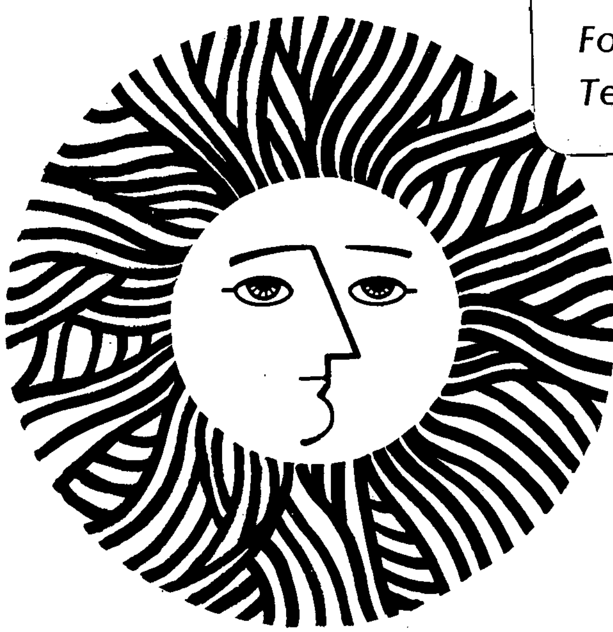

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CONVECTION COEFFICIENTS AT BUILDING SURFACES*

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ABSTRACT

Correlations relating the rate of heat transfer from the surfaces of rooms to the enclosed air are being developed, based on empirical and analytic examinations of convection in enclosures. The correlations express the heat transfer rate in terms of boundary conditions relating to room geometry and surface temperatures. Work to date indicates that simple convection coefficient calculation techniques can be developed, which significantly improve accuracy of heat transfer predictions in comparison with the standard calculations recommended by ASHRAE.

INTRODUCTION

Among the fundamental heat transfer processes, convection is the least understood. In contrast to conduction and radiation, the equations 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 been largely limited to experimental work which is difficult to generalize.

The influence of convective heat transfer processes on the energy performance of buildings can be described in terms of three mechanisms: (1) coupling between room air and the surfaces to which it is exposed, (2) distribution of thermal energy within and between zones due to air circulation, and (3) coupling of the interior air to the external environment through ventilation or infiltration. These processes are all significant and are the subjects of ongoing research. The work summarized here consists of analyses and experiments under way at Lawrence Berkeley Laboratory and the Solar Energy Research Institute; it is directed at understanding the coupling between room air and the room surfaces. The objective is to develop simple, accurate, and highly generalized correlations for the surface convection coefficients in enclosures typical of rooms in buildings.

The LBL research consists in developing numerical analysis capabilities for convection in enclosures, and execution of small-scale laboratory experiments which produce data used to validate and guide the development of the analysis. The numerical analysis is then used to perform "numerical experiments," i.e., to produce a numerically-generated data base of heat transfer information from the surfaces of an enclosure for a variety of temperature and geometry boundary conditions. The LBL experimental research has concentrated on two-dimensional heat transfer in enclosures, which are long enough that the end walls have no effect on the internal convection process. The SERI research consists of small-scale laboratory experiments where the effect of the third dimension of the enclosure is explicitly examined. Heat transfer data and flow visualizations are produced in order to develop a detailed data base from which heat transfer correlations can be developed, and which can be used to validate the analysis.

The experimental effort is limited in the number of configurations that can be examined, but provides a frame of reference for the analysis and is the ultimate test for the analysis. The "numerical experiments," on the other hand, are far more rapidly performed than their physical counterparts. Assuming that carefully selected validation experiments are performed in conjunction with the analysis, the "numerical experiments" provide a larger amount of accurate data on velocity, fluid temperature, and heat transfer than the best experiments reported in the literature. The two research efforts are highly complementary; the experimental work provides depth to the research, while the analysis provides breadth.

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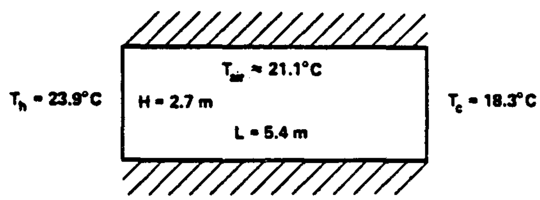
BACKGROUND

Heat transfer between the surfaces of a room and the enclosed air can be characterized in terms of a surface convection coefficient which describes the thermal conductance between the air and the surface. In comparison to the conductance of a normally insulated building envelope element, the convection coefficient is large and has little influence on the overall thermal resistance of the wall. At windows and other highly conductive envelope elements, however, the convection coefficient can represent a significant portion of the total resistance; uncertainty in the coefficient leads to potentially large errors in the calculation of conductive gains and losses, i.e., in the thermal load calculation [1]. For this reason, accurate values for the surface convection coefficients are necessary, for example, to properly determine the size of the passive heating or cooling system and the capacity of the mechanical system.

Heat transfer between room air and exposed surfaces is normally modeled using the convection coefficients documented by ASHRAE [2,3]. These coefficients are based on experimental research conducted 40 to 50 years ago [2,3 and references cited therein] using vertical free-standing flat-plate geometries rather than boundaries of enclosures which have configurations typical of buildings. As a result, the applicability of the reported convection coefficients to building heat transfer calculations is questionable. While these pioneering experiments appear to have been carefully conducted, the temperature dependence of the reported data (e.g., [4]) disagrees with more recent experimental results [5].

More recently there has been extensive research in natural convection in enclosures. Although much of this work has dealt with geometries atypical of rooms in buildings [6,7], there has also been renewed interest in convective heat transfer processes in buildings (see [1] for a recent bibliography). Though much of the recent research does not focus directly on the evaluation of convection coefficients, the research methodology and analysis tools are sufficiently well developed to reconsider past estimates of the importance of convective heat transfer processes in buildings. In one of the recent studies [1] by the present authors, it was shown that the convection coefficients typically used in building energy calculations [2] are in substantial disagreement with the coefficients calculated by existing numerical techniques. It was further shown that the dynamic variability of convection coefficients can lead to substantial errors in the calculation of thermal loads in buildings. Finally, for an example enclosure configuration, it was shown that the most commonly used convection coefficients (ASHRAE constants) are in disagreement by at least 50% with both the laminar and turbulent temperature dependent correlations also recommended by ASHRAE, and differed by about 80% from a preliminary correlation developed at LBL from experimental results (Table 1).

TABLE 1. Convection Coefficient Research Background



CALCULATION METHOD	HOT WALL, W/m²
ASHRAE – CONSTANT	23.3
– TURBULENT	14.0
– LAMINAR	10.8
EXPERIMENTAL CORRELATION	13.0

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The examples cited above clearly demonstrate the need for improved correlations for convection coefficients. 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, we do not expect that a single set of correlations can be obtained which will accurately predict the convection coefficients for all possible configurations of surface and air temperatures. The experimental and numerical simulation techniques used at LBL and SERI to investigate convection coefficients are described in the following section.

TECHNICAL APPROACH

Experiment

The LBL and SERI natural convection experiments utilize apparatuses scaled down by factors of approximately ten and twenty, respectively, from typical rooms. The experiments utilize water as the working fluid in order to suppress radiative exchange within the enclosure and obtain Rayleigh numbers* typical of buildings under enclosure surface temperature boundary conditions that are not so far from ambient that significant heat losses occur from the apparatus. Because the Prandtl** number for air is significantly different from that for water at the temperatures of interest, these are not similitude experiments and the measured heat transfer rates at a given Rayleigh number are not identical to those obtained in a full-scale, air-filled enclosure. The measured heat transfer rates in the LBL experiment are approximately 10% higher than for the full-scale room [1]. For the data from the two experiments used to validate the numerical analysis, the lack of perfect similitude is not a serious disadvantage, because the analysis includes the Prandtl number as a variable. However, the experimental data itself cannot be applied directly to building situations without corrections. The magnitude of the correction is not well understood and will be examined in detail in the near future.

Both experiments allow measurement of enclosure surface temperatures, surface heat fluxes, and temperature distributions within the working fluid for a variety of surface temperature boundary conditions. Both experiments also allow flow visualizations, which are critical to the understanding of the characteristics of the convection process, in particular to determining the conditions under which turbulence sets in.

The LBL apparatus is 15.2 cm x 30.5 cm in vertical cross section with a plan depth of 83.8 cm. The depth was selected to ensure that the end walls would not affect the convection process in the central region of the enclosure; the measurements therefore correspond to two-dimensional convection processes. One vertical wall of the enclosure can be heated and the opposite wall is cooled; all other surfaces are heavily insulated and approximately adiabatic.

The SERI experiment [8] is cubical with edge dimensions of 30.5 cm. The geometry was intentionally selected to investigate the effects of the third dimension of the enclosure on the heat transfer rates from individual surfaces. Furthermore, the apparatus allows the temperature of four of the six surfaces of the enclosure to be controlled so that the effect of the third dimension in real enclosures or rooms can be thoroughly explored for the cubical geometry. This is a largely unexplored area that will require considerable research before the understanding of convection in buildings is placed on equal footing with radiation and conduction.

Numerical Analysis

Computer programs which solve the full Navier-Stokes equations of motion for air flow in buildings have been developed [9-11]. These programs are based on the finite difference method in which the physical volume of interest is divided into a large number of subvolumes, and time is divided into discrete steps. The time-dependent differential equations are then integrated over the finite number of subvolumes and over each time step to obtain a large number of simultaneous algebraic equations, which are solved by matrix inversion. Solutions are obtained for a large number of successive time steps until steady-state flow fields are obtained. The program methodology is described in detail in [11].

The program developed at LBL [11] is suitable for modeling both natural and forced convection in two and three dimensions, for internal and external flows. In addition, the program can model obstacles (internal partitions, furniture), heat sources and sinks (space heating and cooling), and velocity sources and sinks (fans, windows). The program can, in principle, simulate both laminar and turbulent flow. The laminar flow calculations have been verified against analysis and detailed experiments performed at LBL and elsewhere [12-14]. The turbulence modeling capability has recently been added and is presently undergoing testing. This capability is particularly appropriate for the study of wind and fan-driven ventilation and other forced convection phenomena.

In order to use this program, it is necessary to define the problem by specifying the geometric configuration; thermal and velocity boundary conditions, and fluid properties. For example, to obtain the solution of natural convection of air driven by different wall temperatures in a room, one must specify the room geometry, the temperatures of all room surfaces, and the thermophysical properties of air. The air velocity

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*$Ra = GrPr = g\beta \cdot \Delta T H^3 Pr/\nu^2$, where Gr = Grashof number, g = acceleration due to gravity, β = coefficient of thermal expansion, ΔT = characteristic temperature difference, H = height of enclosure, Pr = Prandtl number, and ν = kinematic viscosity.

**Prandtl number: $Pr = \nu/\alpha$, where ν = kinematic viscosity and α = thermal diffusivity.

(or its derivatives) must also be specified on all bounding surfaces of the enclosure, such as at obstacles and internal surfaces where the velocity is zero, and at fan outlets where it is specified by the fan parameters. The computer simulation predicts the velocities and temperatures throughout the volume of interest and also predicts convective heat fluxes on all the surfaces, allowing the calculation of the heat transfer coefficients as a function of position on all the surfaces of the room.

CORRELATION METHODOLOGY

The goal of this activity is to develop technically sound and usable expressions for the convection coefficient or heat flux at each interior surface of an enclosure. There is not a known functional form for the convection coefficients or even assurances that an appropriate expression can be developed. Hence this method consists of (1) generation of a heat transfer data base from physical and numerical experiments and (2) analysis of the data base to extract functional dependencies of the surface heat fluxes on the variables which are expected to influence them. This approach assumes that the sensitivity of the heat flux to all important variables has been examined; if not, the correlation will not be generally applicable.

It is known that the convective heat flux at any given surface in an enclosure will depend on the temperature of that surface and temperature of the air adjacent to the surface:

$$q_i = h_i (T_i - T_{adj}) \qquad (1)$$

where h_i, q_i, and T_i are the convection coefficient, heat flux, and temperature, respectively, for surface i, and T_{adj} is the adjacent air temperature. The adjacent air temperature will in turn depend somehow on the temperatures of all surfaces in the enclosure and on the enclosure geometry and on internal heat sources or sinks; these variables determine the velocity and temperature distributions throughout the volume, in particular the air temperature adjacent to the surface in question. The correlation problem is therefore equivalent to developing a method for calculating T_{adj}. In most building energy calculations, the adjacent air temperature is assumed to be the mean room air temperature.

The experimental portions of the heat transfer data base developed at LBL and SERI during the past year include two configurations; this is insufficient to allow extraction of the geometric sensitivity of the convection coefficient. The correlations from the experiments therefore explore the sensitivity of the heat flux (typically expressed by the Nusselt number*) to the magnitude of the characteristic temperature difference in the enclosure (typically expressed by the Rayleigh number). Caution must be used when applying these correlations to other geometries. It is noted that the geometry variable is the most difficult to access experimentally, since it requires several experimental apparatuses or a highly reconfigurable model.

In order to supplement the experimental data base, the numerical analysis computer program has been used to perform sensitivity studies, especially in regard to geometric sensitivity. This portion of the heat transfer data base includes explicit dependence on the areas of subsurfaces for a particular two-dimensional enclosure. The data base has been analyzed to determine the sensitivity of the heat flux at the enclosure surfaces to both geometry and surface temperatures.

The parametric sensitivity studies were conducted for a two-dimensional room with a height of 2.44 m and a length of 3.66 m, and each of its four major surfaces have been divided into three subsurfaces (described in detail in [15]). The room is shown schematically in Fig. 1; the individual subsurfaces are identified numerically in this figure. 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.

For the first boundary condition configuration selected, the hottest and coldest surfaces are located on opposite vertical walls. Nine simulations were performed to study natural convection under the different temperature boundary conditions displayed in Table 2. The results from these nine simulations have been used to develop simplified correlations for convection coefficients for this particular configuration, as described in detail in reference [16] and briefly summarized here.

It was found that subsurfaces 1, 2, 7, and 8 contribute, in most cases, a very large portion of the total heat transfer into or out of the room air, typically on the order of 90%. The data interpretation has focused on these four subsurfaces, which have been labeled "active" subsurfaces. Successful development of correlations for predicting the heat transfer from the active subsurfaces would account for about 90% of the total heat transfer for this particular configuration. The remaining subsurfaces, which together contribute only about 10% of the total heat transfer between the bounding surfaces on the enclosed air, have not been considered in developing correlations; they have been labeled "inactive" subsurfaces.

When the convection coefficients for the various subsurfaces were calculated with reference to the mean

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*Nusselt number: $Nu = q \cdot H/\Delta T \cdot k$, where q = surface heat flux, H = height of enclosure, ΔT = characteristic temperature difference, and k = thermal conductivity.

TABLE 2. Parametric Run Description

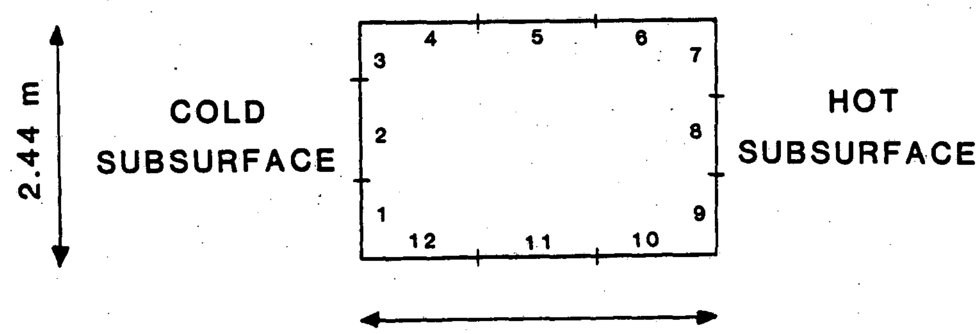
Run #	T_2		T_8	
	(°C)	(°F)	(°C)	(°F)
1	-6.67	20	21.11	70
2	-6.67	20	29.45	85
3	-6.67	20	37.78	100
4	4.45	40	21.11	70
5	4.45	40	29.45	85
6	4.45	40	37.78	100
7	15.56	60	21.11	70
8	15.56	60	29.45	85
9	15.56	60	37.78	100

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

FIG. 1

PARAMETRIC STUDY

ENCLOSURE DESCRIPTION



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air temperature in the enclosure, large variations were observed, particularly for subsurfaces other than 2 and 8. Even for these two subsurfaces the convection coefficients vary by nearly 50-60%. This variability results from the fact that the temperature of air immediately adjacent to a subsurface (T_{adj}) is often quite different from the mean room air temperature. The convection coefficients calculated with reference to the mean air temperature lacked an observable pattern, and would therefore be difficult to predict using a simple equation. Furthermore, convection coefficients calculated with reference to the mean air temperature can have nonphysical negative values. For example, the updraft of warm air leaving subsurface 8 will deposit heat into the cooler subsurface 7 in spite of the fact that the mean room air temperature is less than the temperature of subsurface 7.

When the convection coefficients were calculated with reference to the adjacent air temperature predicted by the numerical analysis, far more consistent results were obtained. For this reason the correlation efforts focused on attempts to obtain an expression for calculating T_i, the air temperature adjacent to subsurface i. The first attempt at obtaining such an expression is described below.

The analysis determined that the adjacent air temperature for all of the active subsurfaces could be calculated from

$$T_i' = K_1\left(T_H \frac{A_H}{A}\right) + K_2\left(T_C \frac{A_C}{A}\right) + K_3\left(T_I \frac{A_I}{A}\right) + K_4 T_H + K_5 T_C \qquad (2)$$

where K_1, K_2, K_3, K_4, and K_5 are constants, T and A are subsurface temperatures and areas, and subscripts H, C, and I refer to the hot, cold, and inactive subsurfaces, respectively. The values of the constants were determined by examining the individual dependence of the calculated adjacent temperature on each of the terms in the above equation. The preliminary results are shown in Table 3.

TABLE 3. Correlation Constants

Subsurface Number	K_1	K_2	K_3	K_4	K_5
2	1.49	1.38	0.89	0	0
8	1.49	1.38	0.89	0	0
1	0.76	0.70	0.45	0	0.49
7	0.76	0.70	0.45	0.49	0

The convective heat flux (q_i') for each active subsurface can be calculated from

$$q_i' = 2.42\ (T_i' - T_i) \quad . \qquad (3)*$$

RESULTS

Experiment

Data from the LBL experiments have been published elsewhere [12,17] and are not repeated here. Portions of the data have been used to validate the convection analysis computer program [12].

The more recent SERI results [8] have included both flow visualizations and heat transfer measurements for Rayleigh numbers between 0.3×10^{10} and 6.0×10^{10}. Flow visualizations have revealed the persistence of a relatively inactive central region of the enclosure for four combinations of heated and cooled vertical walls (the horizontal walls are adiabatic), which is qualitatively similar to the observations made in previous two-dimensional enclosure experiments. Fluid temperature measurements indicated that linear stratification profiles extended through the core region with deviations from linearity near the upper and lower surfaces where hot (top) or cold (bottom) fluid flowed in boundary layer type flows between the hot and cold vertical surfaces. These horizontal boundary layers dominate the heat transfer mechanisms between the temperature controlled surfaces; the core region plays only a small role at the Rayleigh numbers and boundary conditions explored to date. Finally, the beginnings of transition to turbulence in the boundary layers were observed at a Ra number of 3.6×10^{10}. These observations, too, are very consistent with those made previously for two-dimensional flow.

The heat transfer data from the SERI experiment has been analyzed and a correlation developed. Regression analysis of several hundred data points, representing all four hot and cold wall combinations, yielded an expression:

$$Nu = 0.620(Ra)^{0.250} \qquad (4)\#$$

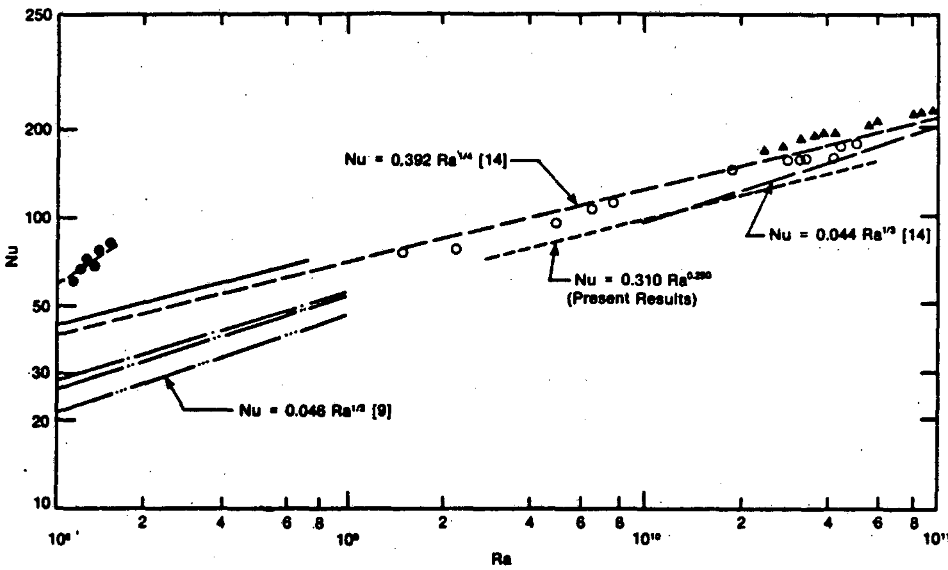
for Rayleigh numbers between 0.3×10^{10} and 6.0×10^{10} for a water filled enclosure. Experimental, numerical, and analytic results from various researchers for two-dimensional enclosure flows are compared with this correlation in Fig. 2. For comparison purposes, all the Nu values in Fig. 2 are based on the hot-to-cold wall temperature difference. Considering the differences between the SERI three-dimensional experiment and previous two-dimensional experiments, there is excellent agreement among the data. This is one extremely important result in that it lends credibility to the applicability of the past two-dimensional experiments and analyses to three-dimensional situations, but it is emphasized that considerably more three-dimensional work is necessary in order to fully understand this relationship.

*2.42 (W/m^2-°C) represents the heat flow weighted average of the convection coefficients on subsurfaces 2 and 8, calculated using the temperature difference between the mean room air and the subsurface for the nine simulations shown in Table 2.

#Nu in Eq. (4) is based on the difference between the bulk fluid temperature and the surface temperature.

FIG. 2

COMPILATION OF TWO-DIMENSIONAL HIGH RAYLEIGH NUMBER DATA AND COMPARISON WITH PRESENT RESULTS



(This figure taken in its entirety from [8], including reference numbers in the figure and the key.)

<u>KEY</u>

Raithby laminar b. l. theory [14], A=.5, Pr=3.5
Raithby turbulent b. l. theory [14], A=.5, Pr=3.5
Gadgil numerical analysis [26], A=1, Pr=.7
Gadgil numerical analysis [21], A=.2, Pr=1
Gadgil numerical analysis [21], A=.1, Pr=1
MacGregor and Emery [9]
○ Bauman, et al. [18], A=1, Pr=.7
▲ Nansteel and Greif [25], A=.5, Pr=3.5
● Burnay, et al. [18], A=1, Pr=.7

Analysis

The heat flux predictions for all four "active" subsurfaces calculated from Eq. (3) are compared against the results of the numerical simulations in Fig. 3. The solid 45° line represents the line of perfect agreement between the two calculation techniques. Calculations based on the present correlation are seen from this figure to compare well with the numerical predictions of the convection program for all four "active" subsurfaces. The points plotted on Fig. 3 have a root mean square deviation from the 45° line of only 2.5 W/m^2. The relative errors in heat flux predictions are observed to be large only in cases when the temperature differences between "inactive" and "active" subsurfaces are small.

Two methods* of calculating convective heat flux recommended by ASHRAE are compared with the numerically obtained results, in Figs. 4 and 5. Both methods relate the convection coefficient to ΔT_{as}, the difference in the mean enclosure air temperature and surface temperature. The convective heat fluxes in Fig. 4 are calculated using the ASHRAE constant convection coefficients according to the following equation:

$$q_{A1} = h \cdot \Delta T_{as} \tag{5}$$

where h = 3.08 (W/m^2-°C) for vertical surfaces

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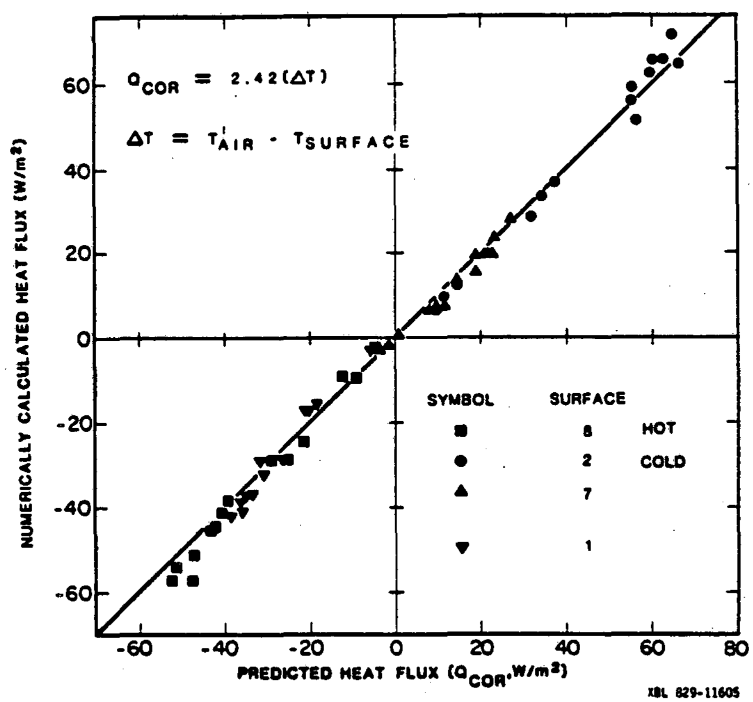
*The ASHRAE constant convection coefficient for a vertical surface is derived from Table 1, page 23.12, 1981 <u>Handbook of Fundamentals</u>, by subtracting out the radiative component of the total surface heat transfer coefficient. The radiative component is based on a 5.6°C (10°F) surface-to-surroundings temperature difference, which is not always typical for real buildings. In addition, it is extremely surprising to notice 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 ASHRAE 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, 1981 <u>Handbook of Fundamentals</u>.

FIG. 3

LBL HEAT FLUX CORRELATION

Comparison with Numerical Calculations



$= 4.04$ (W/m^2-°C) for horizontal surfaces with heat flow upwards

$= 0.95$ (W/m^2-°C) for horizontal surfaces with heat flow downwards.

The convective heat fluxes in Fig. 5 are calculated using the ASHRAE correlations for turbulent convection coefficients according to the following equation:

$$q_{A2} = h \cdot \Delta T_{as}$$

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

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

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

$$h = 0.51 \left(\frac{\Delta T_{as}}{L}\right)^{0.25}$$

where L is the characteristic length of the surface. Both of the ASHRAE correlations roughly approximate the heat flow to the room air from the cold and hot subsurfaces, 2 and 8, and have large errors for the other two active subsurfaces, 1 and 7. The ASHRAE correlations, in fact, often predict the wrong direction of heat flow for subsurface 7. The root mean square (RMS) deviations of the points in Figs. 4 and 5, from the 45° line are 13.3 W/m^2 and 15.8 W/m^2 respectively. These numbers are of the same order of magnitude as the heat fluxes on most subsurfaces. By comparing Figs. 3, 4 and 5, it is clear that the predictions from Eq. (3) show substantially better agreement than do the ASHRAE calculations.

For one of the nine numerical analyses the net imbalance in total convective flux from all surfaces has been calculated for each of the two ASHRAE recommendations for the correlations. Using Eq. (3), an imbalance of approximately 2% was observed. The two ASHRAE methods produced imbalances of -11% and +50% for the constant and temperature-dependent coefficients, respectively.

FIG. 4

ASHRAE CONSTANT COEFFICIENT CORRELATION
COMPARISON WITH NUMERICAL CALCULATIONS

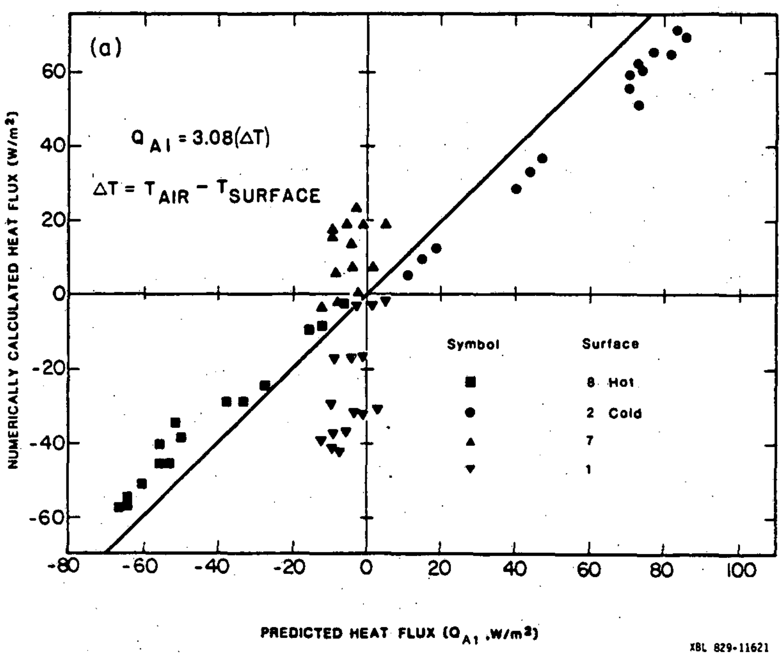
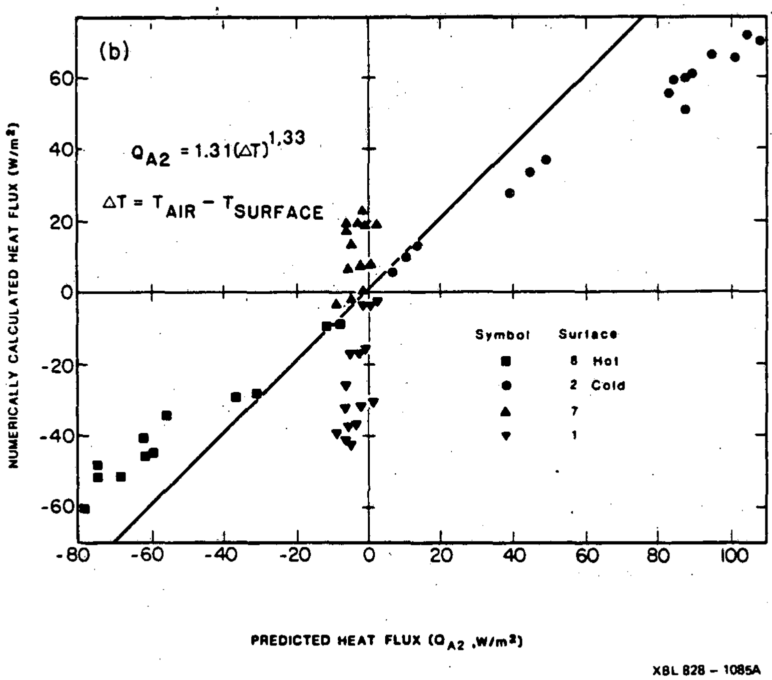


FIG. 5

ASHRAE TURBULENT CORRELATION
COMPARISON WITH NUMERICAL CALCULATIONS



CORRELATION SENSITIVITY

A number of additional simulations have been performed in order to test the generality of the correlation by examining its ability to properly predict convection coefficients for configurations other than those on which it is based. For each sensitivity configuration the "active" subsurface heat fluxes were recalculated using the correlations and then compared with the results computed by the convection program. The parameters varied in the sensitivity studies were:

- o Area of hot subsurface, A_8.
- o Area of cold subsurface, A_2.
- o Temperature of "inactive" subsurfaces, T_I.

If the accuracy of the correlation is strongly affected by any of the above parameters, then caution must be exercised in extrapolating the present correlations to parameter values beyond the range of this study. Results from each of these sensitivity studies are presented below.

Figure 6 presents "active" subsurface heat fluxes for three different areas of the hot subsurface. The hot subsurface area was varied from 41% (1.01 m^2) of the total vertical wall area to 100% (2.44 m^2) of the total vertical wall area. The area and location of the hot subsurface for each sensitivity run is shown in the accompanying diagrams. Despite the wide range of hot subsurface areas, the results of the present correlation all agree quite well with the numerical predictions of the convection program. The root mean square (RMS) error for the correlation results shown in Fig. 6 was calculated to be only 2.30 (W/m^2).

FIG. 6

LBL HEAT FLUX CORRELATION

Hot Surface Area Sensitivity

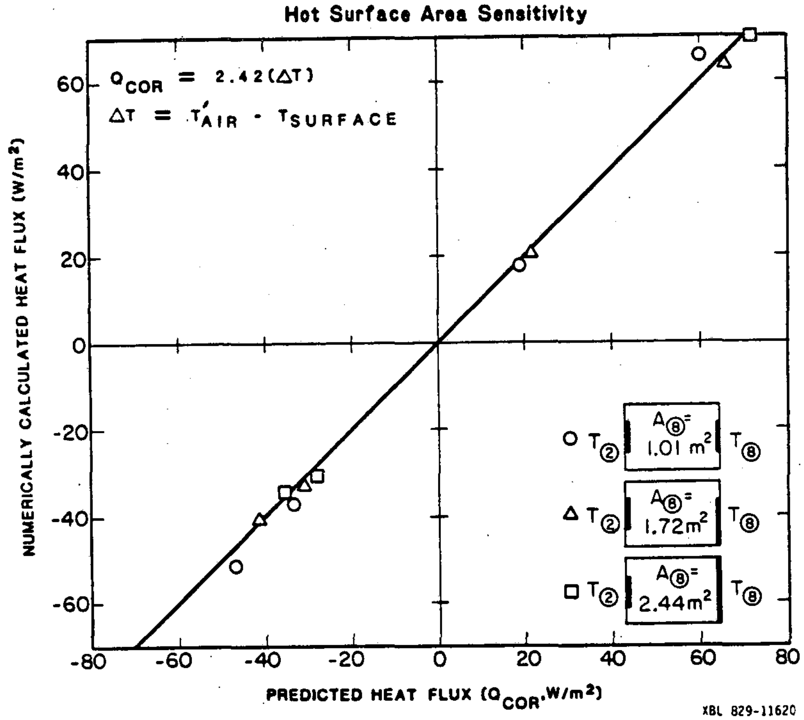


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Figure 7 presents "active" subsurface heat fluxes for three different areas of the cold subsurface, shown in the accompanying diagrams. For these sensitivity runs, the area of the cold subsurface was varied from 52% (1.26 m^2) of the total vertical wall area to 100% (2.44 m^2) of the total vertical wall area. Again the correlation results are reasonably insensitive to this parameter with the RMS error being 3.22 (W/m^2).

In Figure 8 the results are shown for three different values of "inactive" surface temperatures: the reference value of 20°C, and plus and minus 2.8°C from the reference value (22.8°C and 17.2°C). The accuracy of the present correlations over this range of "inactive" surface temperatures is seen to be of the same order as the other sensitivity runs, with the RMS error being 3.82 W/m^2. Part of this insensitivity may be due to the small magnitude of the "inactive" subsurface temperature variations (5.6°C) relative to the maximum temperature difference in the room (44.4°C).

The results for all three of the above sensitivity runs indicate that the present correlations can

FIG. 7

LBL HEAT FLUX CORRELATION

Cold Surface Area Sensitivity

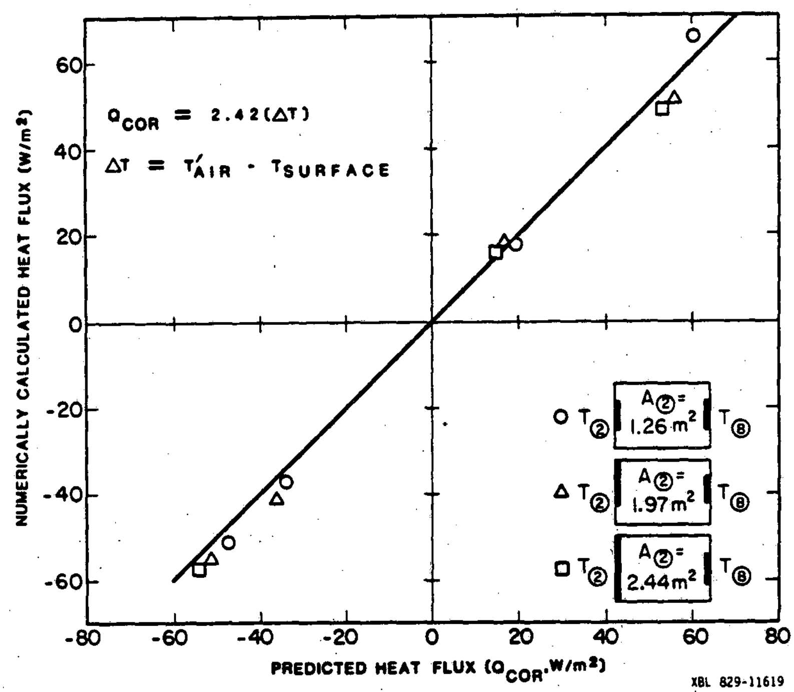
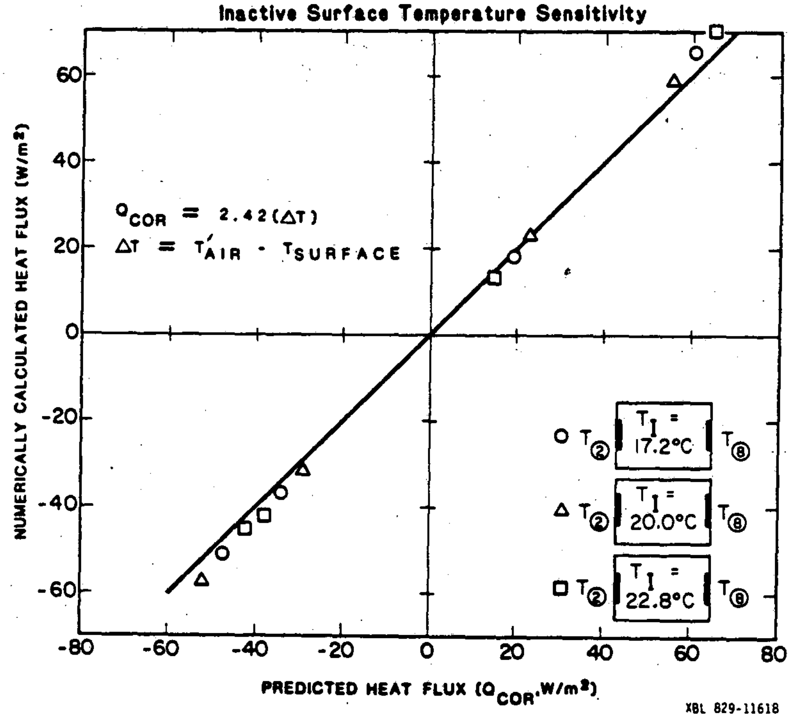


FIG. 8

LBL HEAT FLUX CORRELATION

Inactive Surface Temperature Sensitivity



provide the same level of accuracy (10-15%) over a wider range of configurations than the particular one for which the correlation was originally developed. Future sensitivity analyses will concentrate on other parameters and configurations.

CONCLUSIONS

Convection coefficient correlations have been obtained by analyzing results from numerical simulations of the natural convection process in an enclosure described by a two-dimensional configuration with the hottest and coldest surfaces on opposite vertical walls. The correlations allow prediction of the dominant convective heat fluxes in configurations approximating the one that has been analyzed. The correlations presented apply for a variety of subsurface temperatures and areas. The correlations are a significant improvement over the standard techniques for calculating convective heat fluxes from room surfaces for this particular configuration. Of course, the validity of the present correlations is limited to the particular configuration for which the numerical simulations were performed. However, the methodology presented in this report is quite general and can be applied 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, one may not be able to identify one hottest and one coldest subsurface; all the subsurfaces may have temperatures close to one another. In this case, all the different applicable correlations will predict similar results. In other words, the correlations, if properly formulated, should (and must) give predictions that continuously and smoothly vary over all the possible variations of subsurface temperatures, even as one ranges from one configuration to another.

The numerical analysis technique used in this study has been compared with experimentally obtained flow patterns, temperature fields, and heat flows at the walls for enclosure geometries similar to the one considered in this report. These comparisons have shown agreement usually within a few percent, giving one confidence in using the computer program as a fast device for performing "numerical experiments" for a variety of conditions within the domain of this configuration. It would be desirable, 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.

Several cautions are in order relating to the use of the preliminary correlation reported here. In particular, it should be pointed out that experimental evidence exists showing steady, laminar flow in enclosures for the present configuration, even at the high Ra number values that we have considered. However, almost all the experiments in this regard have used fluids in the 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 needs to be carefully examined experimentally. 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 importance of the three dimensionality of the flow must be determined. This effect is known to exist, but has not been sufficiently studied and quantified in the literature to assert that the present correlations, obtained from two-dimensional simulations, are satisfactory approximations to the three-dimensional conditions that almost always exist in real buildings. Experimental results obtained to date imply that the two-dimensional results are valid in that the primary features of the convection process in enclosures at Rayleigh numbers of interest herein are consistent between two and three dimensions. This similarity persists for (1) the inactivity of the core region, (2) dominance of boundary layer flow in the heat transfer process, and (3) transition to turbulence.

The results presented in this report indicate that the problem of predicting convective heat transfer in rooms using simple correlations is not a hopeless task. The success achieved for the one case considered is encouraging, and leads one to expect that a similar approach will yield successful predictions in other cases as well.

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