Inter-Zone convective heat transfer in buildings: a review

Barakat, S. A.

Publisher's version / Version de l'éditeur:

ASME Heat Transfer Division, 41, pp. 45-52, 1985

National Research Council Canada
Conseil national de recherches Canada

Division of Building Research
Division des recherches en bâtiment

Inter-Zone Convective Heat Transfer in Buildings: A Review

by S.A. Barakat

Reprinted from
Heat Transfer in Buildings and Structures—HTD-Vol. 41
p. 45-52
ASME-AIChE National Heat Transfer Conference
Denver, CO, August 4-7,1985
(DBR Paper No. 1323)

Price $2.00

NRCC 24953

DBR/DRB

Canada

RÉSUMÉ

On examine les travaux de recherche sur le transfert de chaleur de convection inter-zone dans les bâtiments et on traite des paramètres dont dépend ce processus. On présente une comparaison limitée entre les taux de transfert de la chaleur calculés à partir de corrélations existantes et ceux obtenus dans une installation d'essai en grandeur réelle. On a identifié certains secteurs qui nécessitent plus de recherche.

INTER–ZONE CONVECTIVE HEAT TRANSFER IN BUILDINGS: A REVIEW

S.A. Barakat
Division of Building Research
National Research Council of Canada
Ottawa, Ontario, Canada

ABSTRACT

Research work on inter-zone convective heat transfer in buildings is reviewed and the parameters that govern this process are discussed. A limited comparison between the heat transfer rates calculated by existing correlations and those obtained in a full-scale test facility is presented. A number of research areas where more work is needed are identified.

NOMENCLATURE

A_p	Aperture ratio (=D/H)
b	Exponent of Gr_D in Equation (19)
B	Room width
c_p	Specific heat of fluid
C	Coefficient defined in Equation (1)
D	Opening height
D_h	Hydraulic diameter of doorway
g	Acceleration due to gravity
h	Heat transfer coefficient
H	Room height
k	Thermal conductivity of fluid
L	Enclosure length (distance between hot and cold surfaces)
q	Total heat transfer rate through opening
q_c	Power consumption in cold room
q_h	Power consumption in hot room
q_r	Radiative heat transfer component through opening
t	Thickness of partition
T	Temperature
T_c	Cold surface (or room) temperature
T_h	Hot surface (or room) temperature
T_o	Outdoor temperature
ΔT	Temperature difference
ΔT_a	Temperature difference between top and bottom halves of the opening
ΔT_T	Average temperature difference between zones
$(UA)_c$	Loss coefficient of cold room
$(UA)_h$	Loss coefficient of hot room
W	Width of opening
μ	Dynamic viscosity
ν	Kinematic viscosity
ρ	Fluid density
$\bar{\rho}$	Average two-zone density
σ	Stefan-Boltzmann constant

Dimensionless groups

Gr_D	Grashof number [$= g\Delta\rho D^3/\bar{\rho}\nu^2$]
Nu_D	Nusselt number [= hD/k]
Pr	Prandtl number [$= c_p\mu/k$]
Re_D	Reynolds number [$= \bar{\rho}V_bD/\mu$]

INTRODUCTION

Inter-zone convective heat transfer, or the heat transfer between zones (or rooms) in a building, could occur solely due to a temperature difference between the zones (natural convection), or due to the movement of air by mechanical means (forced convection) or a combination of both. Forced convection could be generated by intentionally-used circulation fans or by an unbalance in the air supply or ventilation system.

Because of the fundamental role that inter-zone convection plays in both conventional and passive solar buildings, it has recently received increasing attention from the building research community. Convection is responsible (along with thermal radiation) for the distribution of heat within the occupied space and between spaces in a multi-zone building. This helps in achieving a more uniform temperature distribution in a building and reduces excessive overheating in the south zone of a direct-gain building.

Recent research results have emphasized the importance of the inter-zone convection process in a building and demonstrated the need for accurate accounting of convection in any attempt to predict the building thermal performance. Previous work has also pointed out the significant magnitude of natural convection across door openings. For example, heat flow rates of more than 1200 watts have been found to occur through a 0.9×2.05 m doorway as a result of a 4 K temperature difference between the spaces on either side of the door.[10] Convection through single doorways was also found to be the major mecnanism for distributing heat from a sunspace to the parent house.

Inter-zone convection is also an important factor affecting smoke movement and control in buildings as well as the transfer of pollutants between spaces. Other applications relate to the general movement of air within a building creating draughts, loss of heat at doorways of buildings where heavy pedestrian traffic exists, and heat gain through the doorways of cold storage rooms.

The inter-zone convective heat transfer process is highly dependent on both the geometric configuration of the structure (opening height and width, ceiling height, etc.) and on the boundary conditions that are encountered in the structure (surface temperatures and heat fluxes). This paper presents a detailed review of previous work on inter-zone convective heat transfer, particularly as related to buildings, and discusses the parameters that govern this process.

In the following presentation some changes are made in the terminology used by the original authors to increase clarity and permit comparisons. For example, subscripts are added to the Nusselt, Grashof, and Rayleigh dimensionless numbers to indicate the characteristic length used in their calculation. That is

$$Nu_D = hD/k$$

$$Gr_D = \frac{g\Delta\rho D^3}{\bar{\rho}\nu^2}$$

$$\text{and } Ra_D = Gr_D \times Pr$$

For explanations of all other terms, the reader is referred to the Nomenclature.

REVIEW OF PREVIOUS WORK

Almost all previous papers on inter-zone convective heat transfer have focussed on natural convection only in simple and complex, or divided, enclosures (see Figure 1). Ostrach[1] and Catton[2] presented two comprehensive reviews of natural convection in simple enclosures. This case, however, is outside the scope of this paper and will not be discussed further.

A number of studies have been performed for enclosures of different aspect ratios (H/L), with baffles or partitions mounted in the cavity to study their effect on reducing the convective transfer. For example, Probert and Ward[3] found that the insertion of comparatively short end-mounted vertical baffles (at floor and ceiling) reduces the total steady state heat transfer across the enclosure (including the radiative component) by 20 to 30%.

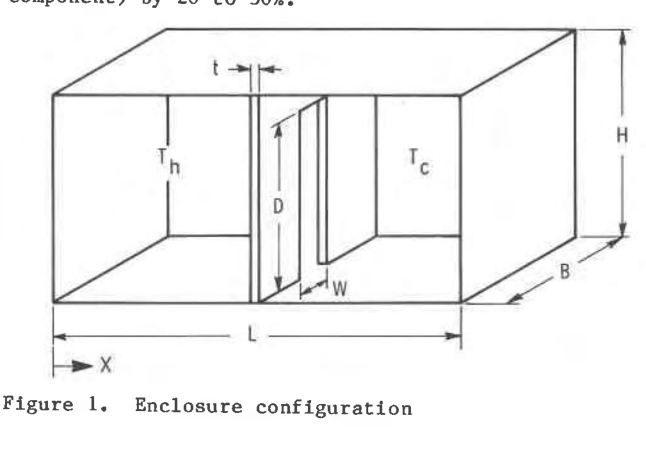

Figure 1. Enclosure configuration

In a Mach-Zender interferometric study, Janikowski et al[4] confirmed the above finding in a rectangular enclosure of high aspect ratio. They also indicated that the free convective heat transfer decreases as the baffle height is increased (opening height decreased).

Emery[5] reported that if vertical baffles are inserted centrally (i.e. leaving openings near the floor and ceiling of the enclosure), their presence does not produce any significant change in convective heat transfer from the unbaffled condition. This can be expected since this baffle arrangement does not alter the buoyancy-induced flow which occurs mainly close to the surfaces.

Chang et al[6] undertook a finite difference study of natural convection in 2-D enclosures, having an aspect ratio of 1, with a vertical partition mounted on either the ceiling or the floor. The vertical walls, parallel to the partition, were maintained at different uniform and constant temperatures while the rest of the surfaces were adiabatic. They showed that the effect of increasing the partition height (reducing the opening height), of increasing the partition thickness, and of moving the partition away from the centre of the enclosure is to reduce the heat transfer between the two sides of the enclosure. Changes in partition height had the most significant effect. Their conclusions on the effect of the partition thickness was based on results for an opening half the enclosure height only. No correlation was given for the heat transfer rates.

In relation to buildings, the first extensive study of natural convection through openings in partitions was carried out by Brown and Solvason in 1962.[7] They have theoretically described the process in terms of the Nusselt number, Nu_D, the Grashof number, Gr_D, and the Prandtl number, Pr, in the form

$$Nu_D = C\, Gr_D^{0.5}\, Pr \qquad (1)$$

where the coefficient C is in the range 0.2 to 0.33. The development of Equation (1) is given in Reference (7).

Brown and Solvason performed their experiments for the case of air flow between two large test chambers (each was 2.44 m square and 1.2 m long) through single rectangular openings at the centre of the partition between the chambers. Opening sizes of 7.6×7.6, 15.2×15.2, 15.2×30.5, 22.9×22.9, and 30.5×30.5 cm were examined for air temperature differences from 8.3 to 47.2 K. Various ratios of partition thickness to opening height, t/D, were also tested over the range 0.19 to 0.75. The air temperature difference between the warm and cold sides was used to determine the Nu and Gr numbers. The air temperature was measured as the average on a vertical grid (floor to ceiling) located 38 and 20 cm away from the partition wall on the warm and cold sides, respectively, and 61 cm from the centre of the opening parallel to the wall. The opening height, D, was used as the characteristic length.

Their experimental results indicated that the exponent of the Grashof number in Equation (1) is slightly greater than 0.5. The results also indicated a dependance of the heat transfer rate (or Nu) on t/D. At $10^7 < Gr_D < 10^8$, very little influence of t/D was noted for the range t/D = 0.19 to 0.38, however for $Gr_D < 10^7$ and t/D = 0.38 to 0.75, the effect of t/D was significant.

In another publication, Brown et al[8] presented equations and charts to calculate heat and moisture transfer due to natural convection through openings in vertical and horizontal partitions separating spaces at different conditions. For vertical partitions the following correlations were given:

$$Nu_D/Pr = 1.16\ (1.0 - 0.6\ t/D)\ Gr_D^{0.432} - 215 \qquad (2)$$

for $10^6 < Gr_D < 10^8$ and $0.19 < t/D < 0.75$, and

$$Nu_D/Pr = 0.343\ Gr_D^{0.5}(1 - 0.498\ t/D) \qquad (3)$$

for $Gr_D > 10^8$.

Brown[9] also examined the case of openings in a horizontal partition when the colder air is at the top. In addition, Brown and Solvason[7] performed limited experiments to examine the effect of forced air flowing horizontally parallel to the vertical partition on the natural convection rates. Such forced convection was found to have an effect (increase or decrease) on the net interchange across the opening depending on the forced air velocity. No correlation was given for these cases.

In 1971, Shaw[10] reported on a study of heat transfer by natural convection and combined natural and forced convection through large rectangular openings in a vertical partition. Since his work addressed the problem of contaminant exchange between hospital rooms, the experiments were actually performed in a hospital and measured air exchange between an isolation room and a vestibule. The rooms were ventilated by a central system with air supply and extraction openings in each room. For natural convection, each room had balanced air supply and extract volumes. For combined natural and forced convection, the cold room had air supply only.

The doorway opening between the rooms was 2.05 m high and ranged from 0.1 to 0.9 m wide. The temperature difference between the rooms was varied from 0 to 12 K, and t/D was constant at 0.05. The measured temperature difference between the top and bottom of the opening was used to calculate Nu and Gr, and the opening height was used as the characteristic length although the effect of this latter variable was not studied.

For the natural convection case and $10^8 < Gr_D < 10^{11}$, Shaw reported that the coefficient C of Equation (1) is a function of the magnitude of the temperature difference, ΔT, and therefore called it the coefficient of temperature. Between a temperature difference ΔT of 4 and 10 K, C had a value of 0.22 and increased for ΔT below 4 K. This increase was attributed to the air velocity or turbulence within the ventilated rooms, which was found by measurement to be of the same order of magnitude as the velocity through the opening. For ΔT above 10 K, C was observed to rise very slowly. Some of the results were affected by proximity of two doorways causing an interaction between the air flow through them when fully opened.

For the case of combined natural and forced convection, excess air was supplied to the isolation room (cold side) to counteract the air motion by natural convection. For this case, Shaw expressed the Nusselt number as a function of a dimensionless group, Sw, that included both the Reynolds and Grashof numbers. Shaw's theoretical development is presented in Reference (10). The final expression was given as

$$Nu_D = C'\ Sw\ Pr \qquad (4)$$

where $Sw = \dfrac{Re^3_{D_h}\ D^3}{Gr_D\ \ D^3_h}$

and D_h = hydraulic diameter of the opening = $\dfrac{2WD}{W + D}$

It can be noted that the term D^3/D^3_h would disappear if the same characteristic length, D, was used in both Re and Gr.

The coefficient C' in Equation (4) was found to be a product of the coefficient C (of the natural convection case) and a forced velocity coefficient C_V which is a function of the forced air flow.

In a later study, Shaw and Whyte[11] indicated that the temperature difference that led to the most accurate correlation of heat flow was the difference between the mean temperature of the air flowing out of the warm room and that flowing into the warm room measured on a grid at the opening. Next best was the temperature difference between the top and bottom of the opening. Since these were not readily available in practice, the third choice was picked, which was the difference between the room air temperatures at the centre of each room at a level half the door's height.

The choice of a different temperature difference from that used in the previous study led to a change in the calculated value of the coefficient C. That is, a C value of 0.27 was found (instead of 0.22) for ΔT between 1 and 10 K.

To indicate the significance of natural convection, Shaw and Whyte pointed out that a temperature difference of 4 K results in an air transfer rate, each way through a 0.9×2.05 m doorway, of 0.25 m^3/s (i.e. a heat transfer rate over 1200 watts).

In 1979, Wray and Weber[12] and Weber et al[13] reported the results of an inter-zone convection similarity study. The experiments were performed in a small two-zone scaled-down enclosure (L = 71 cm, H = 30 cm, B = 43 cm) with Freon as the working fluid. The temperature difference between zones was maintained by keeping the opposite end walls at different temperatures (one wall was heated and the other cooled). They presented their results in the form

$$Nu_H = K\ \ Gr_H^{0.46} \qquad (5)$$

where Nu_H and Gr_H are calculate using the enclosure height, H, as a characteristic length (and not the door height as in previous studies). As a result the value of K was found to be a function of the ratio of the door to ceiling heights (or aperture ratio A_p = D/H), with values of 0.19 and 0.29 for A_p of 0.82 and 1.0 respectively. (These values were reported erroneously in the original paper as 1.9 and 2.9; the new values are calculated from the presented data.) The temperature difference used to develop Equation (5) was taken as the difference between the average hot and cold side temperatures measured by a vertical grid of movable thermocouples inside the space.

If Equation (5) is changed to the form of Equation (1), that is

$$Nu_D/Pr = C\ Gr_D^{0.46} \qquad (6)$$

then coefficient C would have the value of 0.29 and 0.41 for A_p of 0.82 and 1.0, respectively.

In a follow-up study, Weber and Kearney[14] concluded that the opening height is more suitable as a characteristic length. When combined with an appropriate temperature difference, an expression of the form of Equation (1), independent of the room and doorway geometries, can be found.

Two temperature differences were used. The first, ΔT_a, was defined as the difference between the average temperature of the top and bottom halves of the doorway. The second, ΔT_r, is the difference between a volume-weighted average temperature calculated from vertically spaced thermocouples located to the side of the opening near the partition wall.

In addition to the model results, Weber and Kearney used the results obtained from two full-scale tests to determine the coefficient and the exponent of Equation (1) under variations of heat source and heat sink configurations.

They correlated their results in the form

$$Nu_D/Pr = 0.26\ Gr_D^{0.5} \quad \Delta T = \Delta T_a \tag{7}$$

and

$$Nu_D/Pr = 0.3\ \ Gr_D^{0.5} \quad \Delta T = \Delta T_r \tag{8}$$

Some differences observed between the correlation of the laboratory results and the full-scale results were attributed to different convection patterns that might have developed due to differences in configuration and boundary conditions.

Equation (8) was subsequently used by Balcomb[15] and Balcomb and Yamaguchi[16] to produce a formula that can be used as a design aid for determining the necessary door size for particular inter-zone and inside-outside temperature differences under steady state conditions. That is

$$W = UA(T_r - T_o)/\{63.5\ (H \cdot \Delta T_r)^{3/2}\} \tag{9}$$

From detailed simulations of a direct-gain house, Balcomb concluded that ΔT predicted by Equation (9) is quite close to the average room-to-room ΔT for cases of moderate temperature swings in the south zone (about 4 K). However, the equation tends to overpredict the average ΔT for large temperature swings.

Recently, extensive work in the general area of convective heat transfer in buildings was reported by investigators at the Lawrence Berkeley Laboratory. Their work relevant to inter-zone convection is summarized below.

In 1979, Gadgil[17] developed a numerical code for modeling natural convection in rectangular enclosures at Rayleigh numbers, Ra_L, up to 10^{10}. He presented a limited comparison between the numerical solution and actual measurements produced by Ruberg[18] for the case of convection from a heated floor.

Bauman et al[19] performed natural convection experiments in a small model (L = 31 cm, H = 17 cm, B = 84 cm) similar in configuration to the one used by Weber. The model was, however, built to approximate 2-dimensional flow and utilizes water to achieve high Ra. Their limited data on the two-zone configuration indicated the expected behaviour of decreasing convective heat transfer rates with decreasing opening height. In their presentation the enclosure length, L, and the temperature difference between the hot and cold surfaces were used in calculating the Nu and Ra. The choice of L as a characteristic length was not discussed. The height of the vertical hot and cold surfaces would be expected to be the dimension most relevant to the buoyancy-induced flow in this enclosure geometry.

Nansteel and Greif[20] carried out experiments in a small 2-D enclosure (similar to the one described above) with water ($3 < Pr < 4.3$) at Rayleigh numbers over the range $2.3\times10^{10} < Ra_L < 1.1\times10^{11}$ (enclosure length) and aperture ratios ($A_p = D/H$) of 1, 3/4, 1/2, and 1/4. Two partition wall materials were used, thermally conducting and non-conducting. Flow visualizations were also performed.

The results were correlated using the following relationships:

$$Nu_L = 0.748\ A_p^{0.256}\ Ra_L^{0.226} \tag{10}$$

for a conducting partition, and

$$Nu_L = 0.762\ A_p^{0.473}\ Ra_L^{0.226} \tag{11}$$

for a non-conducting partition.

The appearance of A_p in the equation and a different value of the Ra_L exponent (i.e. Gr and Pr exponents) from that in Equation (1) reflect the particular definition of ΔT and the characteristic length and the use of liquids with high Prandtl number. (Pr effect was not considered in previous studies using air.)

The flow visualization experiments revealed the existence of three relatively distinct regions at Ra_L approximating 10^{11}. As shown in Figure 2, they are: an inactive core region; a region of weak circulation in the upper corner on the warm side of the partition; and a peripheral boundary layer flow. The presence of the partition was found to markedly suppress the laminar-turbulent transition on the enclosure vertical surfaces to the extent that turbulent flow did not exist anywhere within the enclosure for Ra_L up to 10^{11}.

The authors pointed out the difference between the exponent of A_p in Equation (10) and that found for data produced in a similar study by Duxbury[21] for enclosures with conducting partitions. For the latter, an exponent of 0.41 was obtained for air at Ra_L in the range of 10^6 and $A_p = 5/8$. Nansteel and Greif questioned the Duxbury finding because of errors that might have occurred in his results due to the large heat loss from his test cell (15 to 40%).

In an extension to their work, Nansteel and Greif[22] carried out similar experiments using silicon oil ($620 < Pr < 920$) over the range $1.55\times10^9 < Ra_L < 5.86\times10^9$ and using a low conductivity partition. The following correlation was developed:

$$Nu_L = 0.280\ A_p^{0.271}\ \ Ra_L^{0.276} \tag{12}$$

The difference between the aperture ratio dependence in Equations (11) and (12) was attributed partially to the difference in Prandtl number but mainly to the differences in the conductance of the partition. They observed a reduction in recirculation strength in the upper warm quadrant of the enclosure as the partition conductance decreased causing reduced convection adjacent to the upper part of the hot wall. They concluded that insulating partitions represent an

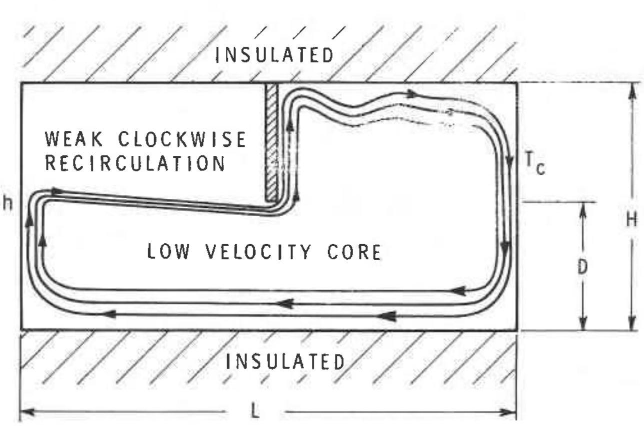

Figure 2. Flow pattern observed by Nansteel and Greif[20] for $10^{10} < Ra_L < 10^{11}$

increased resistance to the heat transfer between the hot and cold zones with a greater resulting sensitivity to the aperture ratio.

In a further development, Nansteel and Greif[23] examined the location of the partition on the inter-zone heat transfer. They also extended the work to cover 3-dimensional cases by dividing the enclosure into two zones using a vertical partition with a single opening at the centre. The opening width was maintained constant at 0.093 of the enclosure width, while the height was varied between 1/4, 1/2, 3/4, and 1 of the enclosure height. Measurements were carried over the range $2.4\times10^{10} < Ra_L < 1.1\times10^{11}$ using water ($3.0 < Pr < 4.3$).

Nansteel and Greif found that the basic flow pattern observed in the centrally-divided 2-D enclosure persists for other partition locations in the range examined (partition locations, x/L, between 0.25 and 0.75). Similarly, the temperature field remained qualitatively the same. The inter-zone heat transfer was shown to depend only slightly on the partition location.

The flow pattern in the 3-D enclosure (with $D < H$), however, was found to be very different from that in the 2-D case. For the 3-D configuration, the flow at the enclosure mid plane, W/2, resembled the flow in the 2-D case only when D/H = 1 (no soffit). The presence of a soffit had a substantial effect in the 3-D case, causing regions of localized turbulence, instability, boundary layer separation and reattachement, and a high degree of Rayleigh number dependence. The heat transfer rate was correlated using the equation

$$Nu_L = 1.19 \ A_P^{0.401} \ Ra_L^{0.207} \tag{13}$$

Little difference was observed in the heat transfer between the 2-D results from their earlier work (Equation (11)), and the 3-D results (Equation (13)) for the same opening height. This observation is surprising because of the substantial differences between the two flow patterns.

In a report by Bauman et al[24] summarizing recent LBL work, the authors used a method similar to that developed by Churchill and Usagi[25] to adjust the heat transfer equations to conditions in air (Pr = 0.7). They also reformatted the results and presented them in terms of Gr_H and Nu_H. The inter-zone heat transfer coefficient, h, in W/m • K was given as

$$h = 1.03 \ (D/H)^{0.47} (\Delta T/H)^{0.22} \tag{14}$$

for the 2-D case, where H is in metres and ΔT is in Kelvin, and

$$h = 1.95 \ (D/H)^{0.25} (\Delta T/H)^{0.22} \tag{15}$$

for the 3-D case. In both cases ΔT was based on the surface temperatures of the vertical end walls (hot and cold) and calculated as

$$\Delta T = (T_h - T_c)/2$$

The authors indicated that, for simplicity, an additional factor of $(1/H)^{0.12}$, calculated for H = 2.74 m, was absorbed into the constants of Equations (14) and (15). This causes a small error (less than 5%) when these equations are applied for room heights between 2 and 4 m. Taking this factor into consideration, the correlation (in dimensionless form and for air) takes the form

$$Nu_H = \text{const.} \ A_P^{a} \ Gr_H^{0.22} \tag{16}$$

where the exponent 'a' has the value of 0.47 and 0.25 for 2-D and 3-D cases, respectively.

It should be noted that Equations (14) and (15) exhibit a different functional dependence of h on the temperature difference compared to Equations (3), (7) and (8). This is due to the use of surface temperatures (and not room air temperatures) in developing the correlation. While the surface temperatures may be easier to measure experimentally, they are not well defined or available in most real situations. Room air temperature, however, is more practical to use for design as well as energy analysis purposes. For this reason, correlations such as those of Equations (14) and (15) may not be directly useful.

In a most recent development Yamaguchi[26] experimentally examined the effects of aperture height, width and vertical position on the inter-zone natural convective heat transfer. He used a one-fifth similitude model of a two-room building, filled with Freon, similar to that used by Weber.[14] For the seven different aperture configurations used, the resulting heat transfer correlation was found to be generally consistant with previous work by Brown and Solvason[7] and by Weber and Kearney.[14] The coefficient of Equation (1) was reported to range between 0.19 and 0.21 depending on the aperture configuration. The average temperatures 15 cm away from the vertical hot and cold surfaces were used to produce the correlation. Yamaguchi reported, however, that the results were much affected by the definition of the temperature difference.

<u>Summary of Previous Results</u>

The work described above on inter-zone convective heat transfer is summarized in Table 1. The main conclusions are:

1. It is apparent, both from theory and experiments, that the heat transfer across openings in partitions between two zones at different temperatures can be expressed by a correlation of the general form

$$Nu = \text{const.} \ A_p^{a} \ Gr^{b} \ Pr^{c} \tag{17}$$

2. The experimental results indicate that, for inter-zone convection in buildings (where air is the transfer medium) and when the opening height, D, is used as the characteristic length in both Nu and Gr, the dependence of the heat transfer coefficient on A_p disappears and Equation (17) reduces to

$$Nu_D/Pr = C \ Gr_D^{b} \tag{18}$$

3. The value of the coefficient C and the exponent b depend on the choice of the temperature difference. When ΔT is taken as the difference between the average room air temperatures, C is in the range 0.22 to 0.33 and the exponent b is about 0.5. There was also some indication that the value of C may be a function of the magnitude of ΔT.

4. All previous results indicate little effect of opening width on the heat transfer coefficient. Even with the large differences observed in flow patterns between the 2-D and 3-D cases (a full width opening and a standard doorway, respectively) there were surprisingly small observed differences in heat transfer coefficients.

SOURCE	EXPERIMENTAL SET-UP/ FLUID	ASPECT RATIO H/L	APERTURE RATIO, Ap	TEMPERATURE DIFFERENCE, LOCATION	GR	Pr	OTHER PARAMETERS	CHAR. LENGTH	CORRELATION EQUATION NO.	COEFFICIENT, C
BROWN & SOLVASON (7)	LARGE TEST CHAMBERS/AIR	0.44	0.062-0.125	AIR NEAR PARTITION	$10^6 - 10^8$	0.7	t/D = 0.19-0.75	D	(1)	0.2-0.33
BROWN ET AL (8)	LARGE TEST CHAMBERS/AIR	0.44	0.062-0.125	AIR NEAR PARTITION	$10^6 - 10^8$ $>10^8$	0.7	t/D = 0.19-0.75	D	(2) (3)	-- --
SHAW (10)	FULL SIZE ROOMS/AIR	-	0.84	AIR AT OPENING TOP & BOTTOM	$10^8 - 1.3 \times 10^{10}$	0.7	t/D = 0.05	D	(1)	ΔT DEPENDENT 0.22 $3 \le \Delta T \le 10$
SHAW & WHYTE (11)	FULL SIZE ROOMS/AIR	-	0.84	CENTRE-ROOM AIR	$10^8 - 1.3 \times 10^{10}$	0.7	t/D = 0.5	D	(1)	ΔT DEPENDENT 0.27 $1 \le \Delta T \le 10$
WEBER ET AL (13)	MODEL/FREON	0.3	0.82-1.0	AVERAGE AIR	$10^9 - 10^{10}$	0.7	-	H	(6)	0.29-0.41
WEBER & KEARNEY (14)	MODEL/FREON	0.3	0.46-1.0	1) AVG AT OPENING 2) AVG NEAR PARTITION	$10^{7.6} - 10^9$	0.7	-	D	(7) (8)	
BAUMAN ET AL (19)	2-D MODEL/ WATER	0.50	0.55-1.0	HOT & COLD SURFACES	$Ra_L = 1.6 \times 10^9 - 5.4 \times 10^{10}$	2.6-6.8	Ap	L		Ap DEPENDENT
NANSTEEL & GREIF (20)	2-D MODEL/ WATER	0.50	0.25-1.0	HOT & COLD SURFACES	$Ra_L = 2.3 \times 10^{10} - 1.1 \times 10^{11}$	3-4.3	PARTITION MATERIAL	L	(10) (11)	Ap DEPENDENT
NANSTEEL & GREIF (22)	2-D MODEL/ SILICON OIL	0.5	0.25-1.0	HOT & COLD SURFACES	$Ra_L = 1.55 \times 10^9 - 5.9 \times 10^9$	620-910	Ap	L	(12)	Ap DEPENDENT
NANSTEEL & GREIF (23)	3-D MODEL/ WATER	0.5	0.25-1.0	HOT & COLD SURFACES	$Ra_L = 2.4 \times 10^{10} - 1.1 \times 10^{11}$	3.0-4.3	Ap	L	(13)	Ap DEPENDENT
BAUMAN ET AL (24)	SUMMARY OF LBL WORK/WATER-AIR	0.5	0.25-1.0	HOT & COLD SURFACES	$Ra_H = 1.5 \times 10^9 - 6 \times 10^9$	0.7	Ap	H	(14) (15)	Ap DEPENDENT

TABLE 1
Summary of Inter-Zone Natural Convective Heat Transfer in Buildings

5. There is an indication that partition thickness has some effect on the heat transfer coefficient for small opening sizes as given by Equations (2) and (3). No experimental evidence exists to confirm such correlations for larger (door size) openings.

6. There is some indication that the natural convective heat transfer coefficient is affected by the partition conductance. This effect, however, was observed for 2-D cases with high Prandtl number fluids and was not tested for other cases with parameter ranges similar to those in real buildings.

7. Present inter-zone convective heat transfer correlations are limited to the boundary condition used in the experiments and are not tested for other boundary conditions. There is a large number of boundary conditions and building configurations potentially of interest; for example, a two-zone configuration with a warm floor in one of the zones, a condition that occurs in direct gain buildings. In general, the extent to which existing correlations can be applied to other configurations is unknown.

8. Most of the recent experiments were performed in small-scale test enclosures utilizing fluids other than air and, in some cases, without full hydrodynamic similarity. Further, little confirmation of the heat transfer correlations was produced using data from full-scale buildings under various boundary conditions.

9. Very few studies examined inter-zone heat transfer under combined natural and forced convection situations. Forced convection in a building could be generated by forced ventilation or unbalances in the forced air heating and ventilation systems.

Limited Comparison Between Natural Convection Correlations

The inter-zone heat flow calculated using previous correlations in the form of Equation (18) are compared to each other as well as to the results of a test obtained in a full-size building. The correlation developed by Nansteel (Equation (15)) is excluded since it is based on surface temperature differences and does not relate to the configuration examined.

Experimental results were obtained in a two-zone test unit of the passive solar test facility at the Division of Building Research, National Research Council of Canada.[27] The unit consists of two rooms 3.05×4.58×2.44 m each, connected by a 0.8×2.03 m doorway in the insulated partition separating the rooms.

Since the heat loss characteristics of these rooms were determined previously,[28] it is possible, by performing a steady-state heat balance for each room, to calculate the amount of heat transferred through the doorway by convection and radiation as

$$q = q_h - (UA)_h (T_h - T_o) = (UA)_c (T_c - T_o) - q_c \quad (19)$$

where q_h and q_c are the heating power consumption in the warm and cold rooms, respectively, and T_o is the outdoor temperature.

The convective heat flow rate can then be calculated by subtracting the radiative component, q_r. Assuming that both room cavities behave like black bodies, the radiation heat transfer through the doorway is calculated as

$$q_r = W D \sigma (T_h^4 - T_c^4) \quad (20)$$

To approximate a steady-state condition, a period of 6 hours was chosen during a night when the outdoor temperature remained fairly constant at about 0.5°C. All windows were shaded before and during the test period to eliminate the dynamic effect associated with the storage of solar energy in the room. Rooms were kept at 21.6°C and 20.1°C (±0.1 K). Multi-point averaging thermopiles, with sensors placed at room mid-height, were used for temperature control and measurement.

Under the conditions described above, the total heat flow rate through the doorway (calculated from Equation (19)) was 322 watts (±10 W), of which 14 watts was estimated to be due to radiation.

The convective heat transfer rate obtained from measurements is compared with that predicted by correlations in Table 2. Values in the table correspond to $Gr_D = 1.9 \times 10^9$ and $Pr = 0.7$.

Although very limited, the comparison indicates that correlations in the form of Equation (1) would predict the inter-zone natural convective heat transfer rate provided that the temperature difference is properly defined and the corresponding coefficient C is determined. It should be noted that the temperature difference used to arrive at the experimental value was the difference between the average mid-height room air temperatures. This does not correspond to any of the temperature differences used in the correlations.

FUTURE RESEARCH REQUIREMENTS

Additional research is needed before reliable convection correlations and design guidelines can be made available to the building designer or energy analyst. This work should cover:

1. Examination of the applicability of recently developed correlations to full-scale conditions in actual buildings. Also modifying the correlations, if needed, to accommodate temperature differences that can be readily measured or calculated and are therefore of practical significance to designers.

Correlation	Natural convective rate, watts
Brown et al, Equation (3)	318
Shaw and Whyte, Equation (18) ($C = 0.27$, $b = 0.5$)	254
Weber and Kearney, Equation (8) ($C = 0.3$, $b = 0.5$)	285
Experimental value	308

TABLE 2
Comparison of Natural Convective Heat Transfer Rates

2. Examination of a wide variety of two-zone configurations and boundary conditions to test the generality of the correlations or develop more general correlations for zone coupling.

3. Study of inter-zone convection under combined natural and forced convection situations. For example, the interaction between natural convection, due to temperature differences between spaces, and forced ventilation or air distribution in the building.

4. Examination of the effect of partition thickness and conductance on inter-zone convective heat transfer with large opening sizes for cases where they may have significant impact; for example, natural convection between a sunspace and the parent building through multiple openings.

5. Study of the effect of the Prandtl number on the heat transfer coefficient to enable future use of small experimental models under similar conditions.

6. Implementation of developed correlations in energy analysis programs and assessing the ability of the improved models to predict energy consumption and indoor temperatures in multi-zone buildings.

SUMMARY

A review of previous research work on inter-zone convective heat transfer is presented. The pure natural convection case has received the most attention compared to other cases of natural convection combined with forced air movement between the spaces. Previous work has pointed out the significant magnitude of the natural convective heat flow through inter-zone openings. It indicated that for this case, the heat transfer can be correlated by an equation of the form

$$Nu_D/Pr = C\, Gr_D^{\,b}$$

where b is approximately 0.5 and C lies between 0.22 and 0.33 depending on the temperature difference used for the correlation. Work is still needed to define the most appropriate temperature difference and to determine the associated value of the coefficient C.

A need for additional work is identified to develop heat transfer correlations for other multi-zone configurations and boundary conditions, as well as for conditions of mixed natural and forced convection. The research should lead to methods and correlations that can be used by the building designer and energy analyst to enable them to take this significant heat transfer component into account.

ACKNOWLEDGEMENTS

This paper is a contribution from the Division of Building Research, National Research Council Canada.

REFERENCES

1. Ostrach, S., "Natural Convection in Enclosures", Advances in Heat Transfer, Vol. 8, pp. 161-227, Academic Press, New York, 1972.
2. Catton, I., "Natural Convection in Enclosures", Proc. 6th Int. Heat Transfer Conf., Toronto, Vol. 6, pp. 13-31, 1978.
3. Probert, S.D. and J. Ward, "Improvements in the Thermal Resistance of Vertical Air-Filled Enclosed

Cavities", Proc. 5th Int. Heat Transfer Conf., Tokyo, Vol. 3, pp. 124-128, 1974.

4. Janikowiski, H.E., J. Ward and S.D. Robert, "Free Convection in Vertical Air-Filled Rectangular Enclosures with Baffles", Proc. 6th Int. Heat Transfer Conf., Toronto, Vol. 2, pp. 257-263, 1978.

5. Emery, A.F., "Exploratory Studies of Free Convection Heat Transfer Through an Enclosed Vertical Liquid Layer with a Vertical Baffle", J. Heat Transfer, Vol. 91, pp. 163-164, 1969.

6. Chang, L.C., J.R. Lloyd, and K.T. Yang, "A Finite Difference Study of Natural Convection in Complex Enclosures", Proc. 7th Int. Heat Transfer Conf., Munich, Vol. 2, pp. 183-188, 1982.

7. Brown, W.G and K.R. Solvason, "Natural Convection Through Rectangular Openings in Partitions, Part I: Vertical Partitions", Int. J. Heat Mass Transfer, Vol. 5, pp. 859-868, 1962.

8. Brown, W.G., A.G. Wilson and K.R. Solvason, "Heat and Moisture Flow through Openings by Convection", ASHRAE Journal, Vol. 5, No. 9, pp. 49-54, 1963.

9. Brown, W.G, "Natural Convection Through Rectangular Openings in Partitions, Part II: Horizontal Partitions", Int. J. Heat Mass Transfer, Vol. 5, pp. 869-881, 1962.

10. Shaw, B.H., "Heat and Mass Transfer by Natural Convection and Combined Natural and Forced Air Flow through Large Rectangular Openings in a Vertical Partition", Proc. Int. Mech. Eng. Conf. on Heat and Mass Transfer by Combined Forced and Natural Convection, Manchester, Vol. 819, pp. 31-39, 1972.

11. Shaw, B.H. and W. Whyte, "Air Movement through Doorways: The Influence of Temperature and its Control by Forced Air Flow", Bldg. Serv. Engrg., Vol. 42, pp. 210-218, 1974.

12. Wray, W.O. and D.D. Weber, "LASL Similarity Studies: Part I Hot Zone/Cold Zone: A Quantitative Study of Natural Heat Distribution Mechanisms in Passive Solar Buildings", Proc. 4th Nat. Passive Solar Conf., Kansas City, Vol. 4, pp. 226-230, 1979.

13. Weber, D.D., W.O. Wray and R. Kearney, "LASL Similarity Studies: Part II Similitude Modeling of Inter-Zone Heat Transfer by Natural Convection", Proc. 4th Nat. Passive Solar Conf., Kansas City, Vol. 4, pp. 231-234, 1979.

14. Weber, D.D. and R. Kearney, "Natural Convection Heat Transfer through an Aperture in Passive Solar Heated Buildings", Proc. 5th Nat. Passive Solar Conf., Amherst, MA, pp. 1037-1041, 1980.

15. Balcomb, J.D., "Heat Storage and Distribution in Passive Solar Buildings", Los Alamos National Laboratory Report LA-9694-MS, 1983.

16. Balcomb, J.D. and K. Yamaguchi, "Heat Distribution by Natural Convection", Proc. 8th Nat. Passive Solar Conf., Santa Fe, New Mexico, 1983.

17. Gadgil, A., "On Convective Heat Transfer in Building Energy Analysis", Ph.D Thesis, Dept. of Physics, University of California Berkeley, (also issued as Lawrence Berkeley Laboratory Report LBL-10900), 1979.

18. Ruberg, K., "Heat Distribution by Natural Convection: A Modeling Procedure for Enclosed Spaces", M.Sc. Thesis, Dept. of Architecture, Massachusetts Institute of Technology, Cambridge, MA, 1978.

19. Bauman, F., A. Gadgil, R. Kammerud and R. Greif, "Buoyancy Driven Convection in a Rectangular Enclosure: Experimental Results and Numerical Calculations", ASME Paper No. 80-HT-66, Transfer Conf., Orlando, FL, 1980. (Also issued as LBL Report LBL-10257).

20. Nansteel, M.W. and R. Greif, "Natural Convection in Undivided and Partially Divided Rectangular Enclosures", Trans. ASME J. Heat Transfer, Vol. 103, pp. 623-629, 1981.

21. Duxbury, D., "An Interferometric Study of Natural Convection in Enclosed Plane Air Layer with Complete and Partial Central Vertical Divisions", Ph.D. Thesis, University of Sulford, UK, 1979.

22. Nansteel, M.W. and R. Greif, "Natural Convection Heat Transfer in Complex Enclosures at Large Prandtl Numbers", Trans. ASME J. Heat Transfer, Vol. 105, pp. 912-915, 1983.

23. Nansteel, M.W. and R. Greif, "An Investigation of Natural Convection in Enclosures with Two- and Three-Dimensional Partitions", Int. J. Heat Mass Transfer, Vol. 27, No. 4, pp. 561-571, 1984.

24. Bauman, F., A. Gadgil, R. Kammerud, E. Altmayer and M.W. Nansteel, "Convection Heat Transfer in Buildings: Recent Research Results", ASHRAE Trans., Vol. 89, 1983.

25. Churchill, S.W. and R. Usagi, "A General Expression for the Correlation of Rates of Transfer and Other Phenomena", AICHE Journal, Vol. 18, pp. 1211-1128, 1972.

26. Yamaguchi, Y., "Experimental Study of Natural Convection Heat Transfer Through an Aperture in Passive Solar Heated Buildings", Proc. 9th Nat. Passive Solar Conf., Columbus, Ohio, 1984.

27. Barakat, S.A., "NRCC Passive Solar Test Facility: Description and Data Reduction", Building Research Note No. 214, Division of Building Research, National Research Council of Canada, Ottawa, 1984.

28. Barakat, S.A., "NRCC Passive Solar Test Facility: Performance of Direct Gain Units", Building Research Note No. 215, Division of Building Research, National Research Council of Canada, Ottawa, 1984.

reprinted from

Heat Transfer in Buildings and Structures – HTD-Vol. 41
Editor: P. J. Bishop
(Book No. G00298)

published by

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 East 47th Street, New York, NY 10017
Printed in U.S.A.