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CONVECTIVE HEAT TRANSFER IN ROOMS WITH MIXED CONVECTION

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Introduction

Since 1970's, the insulation of buildings has been improved in order to reduce the space load in a room. As a consequence, the air supply has been reduced. Hence, buoyancy becomes one of the dominant factors to influence the indoor airflow. The airflow has been changed from forced convection into mixed convection. The convective heat transfer coefficients for mixed convection must be different from those for natural and forced convection. But until now, there is no sufficient information available for engineering applications in ventilated rooms.

On the other hand, numerical calculation of airflow and heat transfer in a room has been used extensively in recent years. The calculation method, which solves a set of partial differential equations for the turbulent flow and heat transfer, uses the standard wall function for solid boundary conditions [1]. The wall function can be applicable in fully developed conventional turbulent boundary layers of a room with a high overall Reynolds number. However, the overall Reynolds number in a ventilated room with mixed convection is considerably small and the boundary layers are not fully developed. Besides, the velocity and temperature profiles can be different from the conventional ones. Consequently, the standard wall function cannot present good results under low Reynolds numbers, because the convective heat transfer coefficients calculated from the wall function are always too small [2]. Moreover, the standard wall function cannot be applied for a complex boundary such as venetian blinds. Nevertheless many contributions have been published for the turbulent boundaries in a flat plate under low overall Reynolds numbers [3, 4, 5, 6]. But the foregoing studies showed that there is limited agreement about the nature of the turbulent boundary layer at low Reynolds numbers and these studies did not concern the specific airflow in a room. From the view point of easy cooperation with an airflow program, the wall function for temperature can be improved by giving new equations for convective heat transfer coefficients.

Therefore, it is necessary to provide new formulas of convective heat transfer coefficients for rooms with mixed convection. This paper will present an experimental approach for obtaining the convective heat transfer coefficients.

Experimental Setup

It is very common that the convective heat transfer coefficient h_c is defined as:

$$h_c = q_c / (T_{wall} - T_{room}) \qquad (1)$$

where q_c is the convective heat flux in an inside wall surface, T_{wall} is the surface temperature of the wall and T_{room} is the air temperature in the centre of the room. However, in most cases there is a vertical air temperature gradient in a room with mixed convection. This definition sometimes will present negative or infinite convective heat transfer coefficients. Because the thickness of boundary layers in a room with mixed convection is about 3 to 10 cm, we prefer to define the convective heat transfer coefficient as:

$$h_c = q_c / (T_{wall} - T_{10}) \qquad (2)$$

where T_{10} is the air temperature in the point 10 cm from the wall surface. The convective heat fluxes can be obtained from:

$$q_c = q_t - q_r \qquad (3)$$

where q_t is the total heat flux measured by a heat flux meter and q_r is the radiative heat flux. q_r in wall surface i is determined from:

$$q_r = \sum_{j=1}^{N} \epsilon_i \Phi_{i,j} (E_{b,i} - E_{b,j}) \qquad (4)$$

where N is the total surface number of the room, ϵ_i is the emissivity of the surface, $\Phi_{i,j}$ is the radiative heat exchange factor between surfaces i and j, $E_{b,i}$ and $E_{b,j}$ are the emissive power of black body of surfaces i and j respectively. The radiative heat exchange between enclosure surfaces is complicated. In practice, these enclosure surfaces are considered as grey bodies. Hoornstra [7] presented a formula to calculate the radiative heat exchange factor $\Phi_{i,j}$ in which the multiple reflection of the radiative heat exchange in a room was considered. If the view factor is defined as F, the $\Phi_{i,j}$ can be calculated from the following matrix equation:

$$[\Phi] = \{[I] - [F]\cdot \mathrm{diag}(1-\epsilon)\}^{-1}\cdot [F]\cdot \mathrm{diag}(\epsilon) \qquad (5)$$

where [I] is unit matrix.

Hence, experiments should concern the measurements of enclosure surface temperatures, the air temperature differences between the enclosure surfaces and the air points 10 cm from the surfaces and the total heat fluxes through these surfaces. Since the convective heat transfer coefficient is not only related to the Raleigh number but also to the Reynolds number, air velocities in the points 10 cm from the surfaces should also be measured.

In the Delft University of Technology, the experiments were carried out in a full scale climate room which is 5.6 m long, 3.0 m wide and 3.2 m high as shown in Fig. 1 [8]. The thermal properties of the room enclosure materials are presented in Table 1.

The venetian blinds near the window can be heated uniformly by electricity in order to simulate solar radiation. There is a cavity above the ceiling and one under the floor in which the temperature can be controlled to simulate the rooms above and below. The floor surface temperature of the space above the room was controlled to be the same as that of the room and the ceiling surface temperature of the space below the room to be the same as that of the room. When the room air temperature is higher than that of the inside surfaces of rear and side walls, these walls can be electrically heated to ensure the total heat exchange through these walls to be zero. The air temperature outside of the window can be set at a value

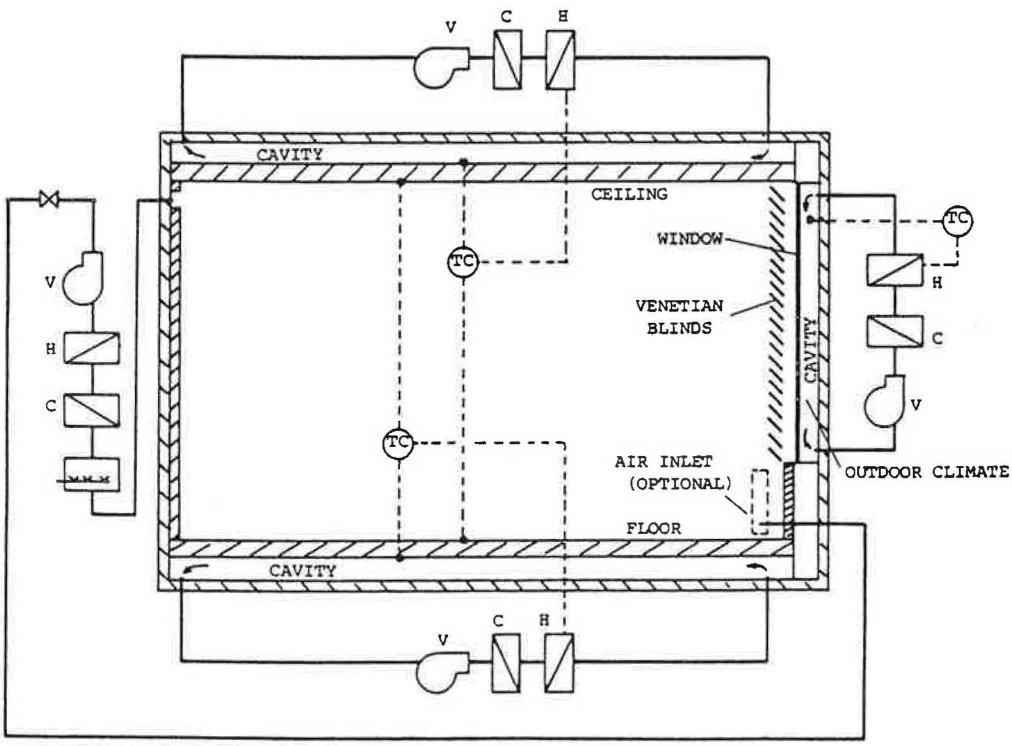


Fig. 1. The ventilated room.

Table 1. Room enclosure materials

Enclosure	Thickness m	Density kg/m3	Specific heat J/kg·K	Heat conductivity W/m·K
Ceiling	0.175	2300	840	1.9
Floor	0.175	2300	840	1.9
Rear wall	0.140	700	840	0.23
Side wall	0.140	700	840	0.23
Parapet	0.100	30	1470	0.03
Window	Outside glass: thickness 6 mm, absorption coefficient 0.018. Inside venetian blinds: slat angle 45°, slat width 55 mm, the height between two slats 50 mm, absorption coefficient 0.3.			

between 0.0 to 40.0°C. The sensor for the control of the room air temperature was placed in the middle of the occupied zone (x=2.8 m, y=1.5 m and z=0.9 m). The set point of the sensor can be a value between 10.0 to 35.0°C.

The measurements of convective heat transfer coefficients in walls were focused in the ceiling and the floor. The location of the measuring points for the air velocities, the surface temperatures, the temperature differences and the heat fluxes for the ceiling is shown in Fig. 2. The anemometers were positioned near the ceiling in the middle section of the room (y=1.5 m). A similar measuring method was applied for the floor.

The air velocities were measured through hot-wire anemometers, the temperatures by copper-constantan thermo-couples, the temperature differences by means of series connection of ten thermo-couples (Fig. 3A), and heat fluxes via TPD-TNO type heat flux meters (Fig. 3B). All of them were connected to a data logger. The air velocities in a room with mixed convection are quite small and a hot-wire probe can be used only for the places where the velocity is relative high. The measurement of air velocities below 0.05 m/s with the hot

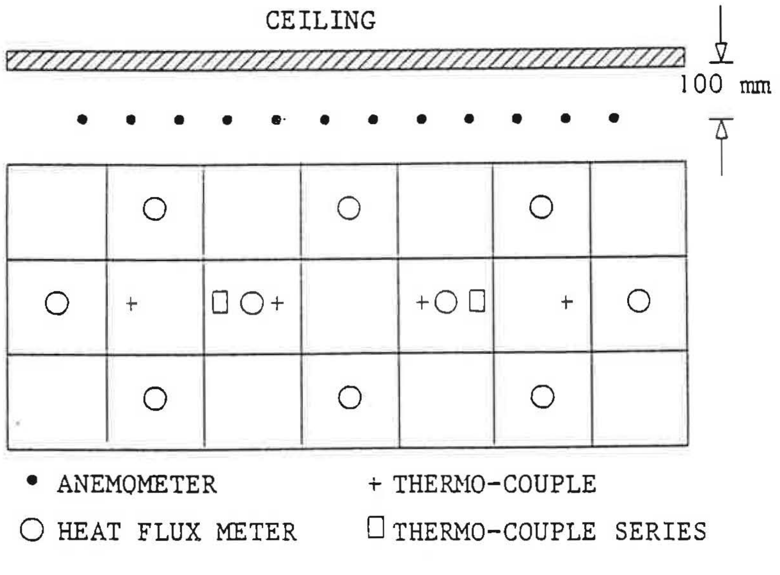
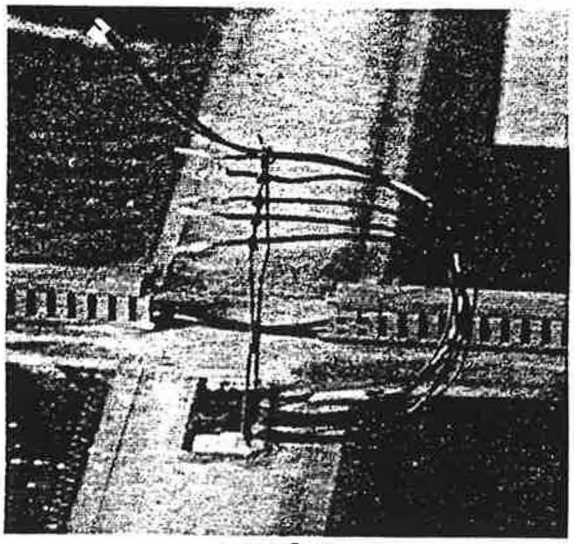
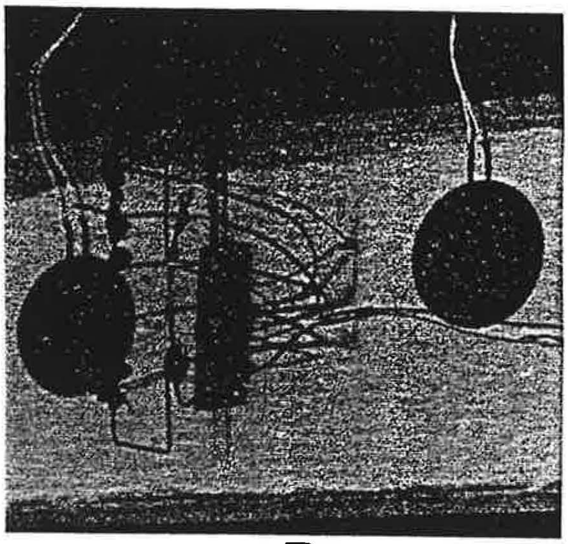


Fig. 2. The measuring points on the ceiling.





A B

Fig. 3. The measurements of temperature difference and heat fluxes.
(A) Series connection of ten thermo-couples; (B) heat flux meters.

wire anemometers is very difficult because of the calibration of the probes and the impact of free convection on the heat transfer from the probes. The error of the copper-constantan thermo-couples is about 0.3°C in temperature measurements. The series connection of the thermo-couples gives negligible errors for measuring the temperature differences. The accuracy of the calibration value for the heat flux meters is within 5%.

In order to simulate the airflow and heat transfer mostly occurring in an air-conditioned room, the measurements were carried out in the following two systems as shown in Fig. 4.

System 1: A displacement inlet with the dimension of 68 cm in height and 50 cm in width was on the floor near the rear wall. Two outlets were in the rear wall near the ceiling. A table, 175 cm long, 145 cm wide and 85 cm high was placed near the window.

System 2: The displacement inlet was installed on the

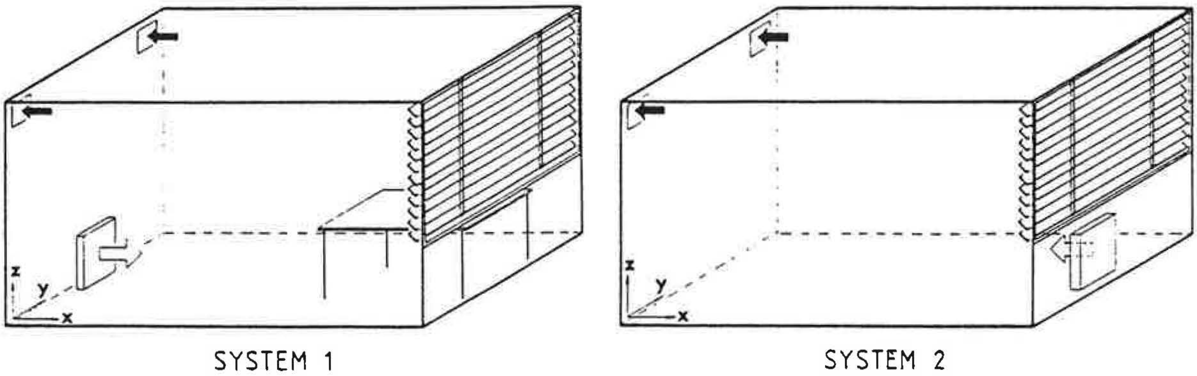


Fig. 4. The ventilating systems.

floor near the window and there were two outlets in the rear wall near the ceiling. In this system, the table cannot be placed near the window otherwise the inlet air is confined under the table.

The air supply was controlled to be 3, 5 or 7 times air exchange rate per hour (ach) for the room. The heat supply on the venetian blinds was controlled to be 300, 600 and 950 W for the simulation of solar radiation. All the measurements were done after the intended conditions were set up for more than 36 hours, because the room was constructed heavily and it needs very long time to reach a steady state.

Convective Heat Transfer Coefficients

Results on the Ceiling and the Floor

The convective heat transfer coefficients measured in the ceiling are illustrated in Fig. 5 and in the floor in Fig. 6. Figs. 5(A) and 6(A) show the convective heat transfer coefficients against the air velocities and Figs. 5(B) and 6 (B) against the air temperature differences between the wall surface and the air point 10 cm from the wall. The results are not encouraging for the values are widespread. The results in Fig. 5 have been divided into different groups by air velocities and temperature differences. The heat transfer coefficients seem still independent with these two parameters.

From Equation (2), one has to realize that the smaller the temperature difference, the more unreliable the convective heat transfer coefficient. A slight change in the temperature difference between the wall surface and the air point 10 cm from the surface can cause a large difference of the heat transfer coefficient. Partly this is the reason for the widespread and occasionally extreme high values for small temperature differences in Fig. 6(B). Another remark can be drawn from the results is that there is indeed a tendency for increasing h_c with increasing air velocities.

Figs. 7 and 8 give the relation between the convective heat fluxes and the temperature differences (between the enclosure surfaces and the air points 10 cm from the surfaces). The graphs look much better although they present the same results as illustrated in Figs. 5 and 6. From Figs. 7 and 8, we may state that the relationship between the convective heat fluxes and the temperature differences is a linear one. The slope of the lines in the figures in fact is an "average" convective heat transfer coefficient. Table 2 shows the "average" convective heat transfer coefficients. The values on

Table 2. The "average" convective heat transfer coefficients

Enclosures	$T_{wall} - T_{10}$ K	Average h_c W/m2·K
Ceiling	<0 unstable	4.0
Floor	>0 unstable	4.7
Floor	<0 stable	4.3

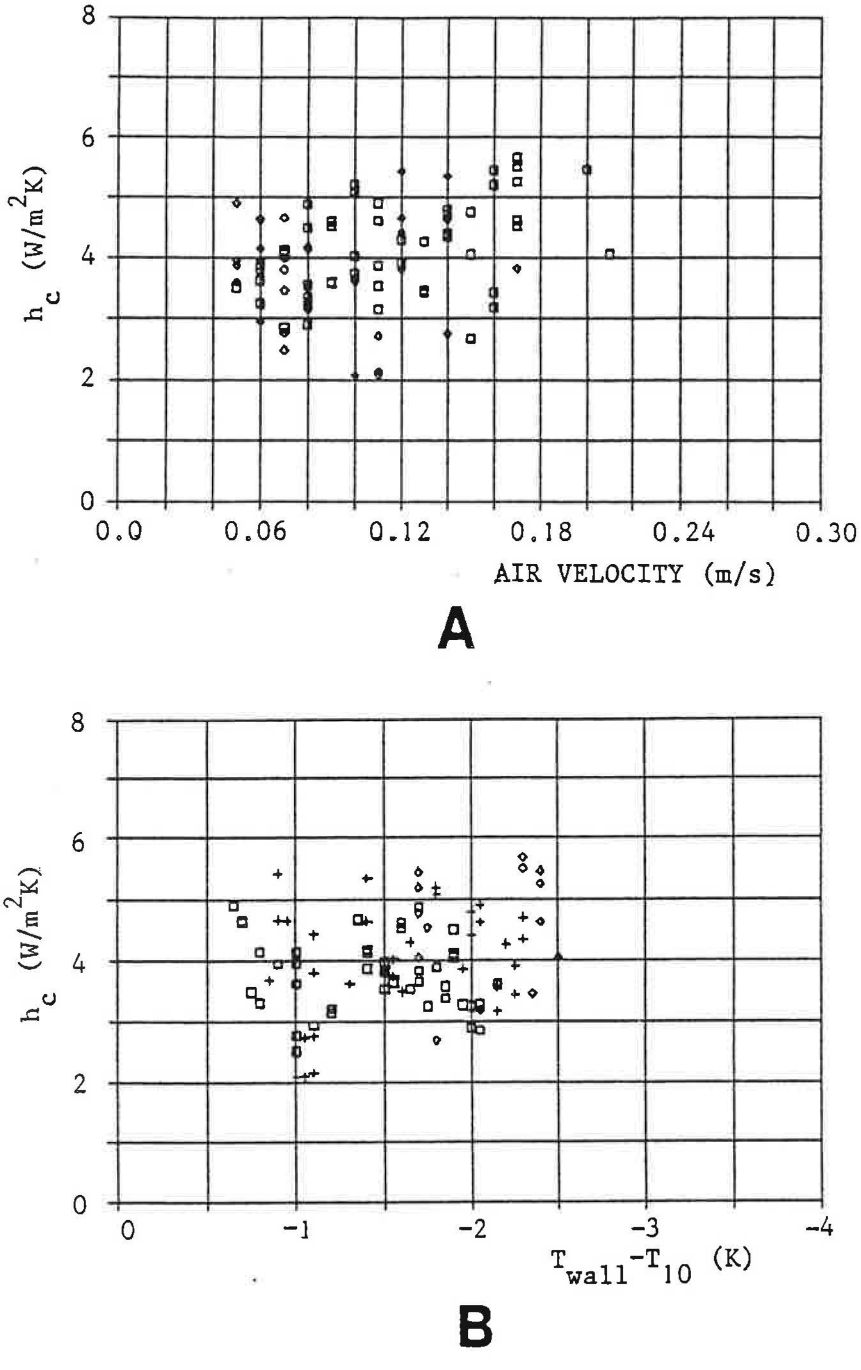


Fig. 5. Convective heat transfer coefficients on the ceiling. (A) Against the air velocities (□ $-0.5>T_{wall}-T_{10}>-1.5$ K, ◇ $-1.5>T_{wall}-T_{10}>-2.5$ K); (B) against the temperature differences (□ $v<0.1$ m/s, + $0.1<v<0.15$ m/s, ◇ $0.15<v<0.21$ m/s).

the floor are higher than those on the ceiling because the air supply inlet was placed on the floor and generated higher air velocities on the floor level. The higher air velocities result in larger convective heat transfer coefficients.

The measuring error of the convective heat transfer

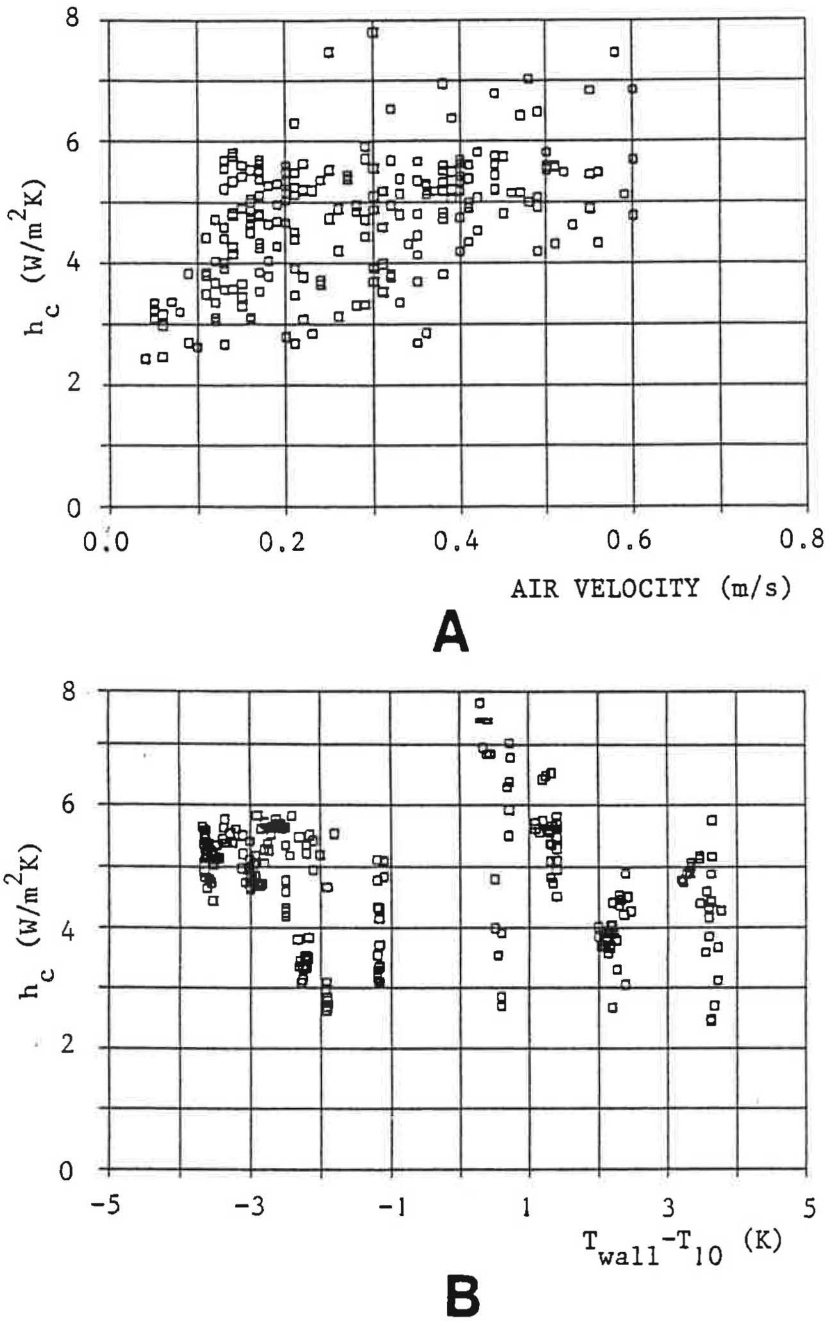


Fig. 6. Convective heat transfer coefficients on the floor. (A) Against the air velocities; (B) against the temperature differences.

coefficients, δh_c, can be written as:

$$\delta h_c=\delta q_t\cdot\frac{1}{T_{wall}-T_{10}}+\delta q_r\cdot\frac{1}{T_{wall}-T_{10}}+\delta(T_{wall}-T_{10})\cdot\frac{q_t-q_r}{(T_{wall}-T_{10})^2} \qquad (6)$$

The measuring error by the heat flux meters, δq_t, is less than 5% of q_t and it is rather small. The determination of the radiative heat exchange among the enclosures, in general, will not generate a large error if the enclosure surface temperatures measured are reliable. However, the surface

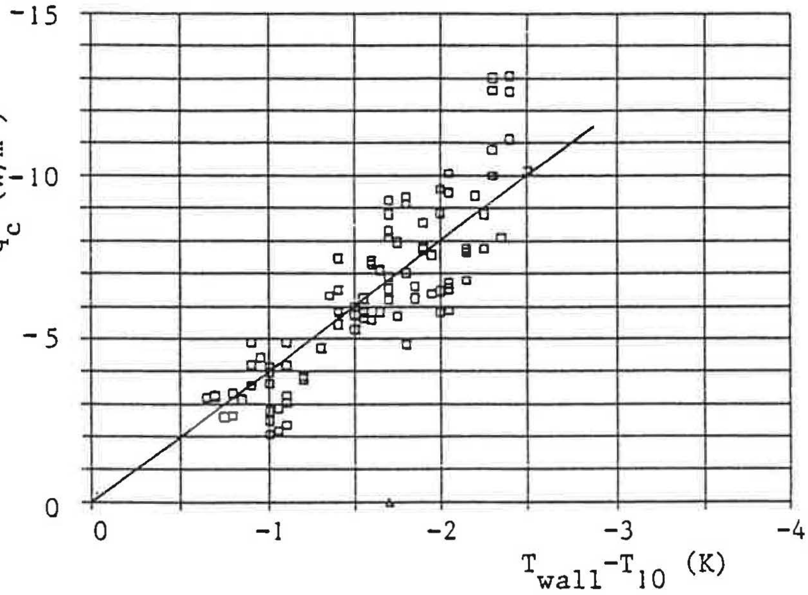


Fig. 7. Relationship between convective heat transfer and temperature difference on the ceiling.

temperature in a wall is not constant and only three or four thermo-couples were used in a wall for the measurements of surface temperatures. In fact, more measuring points are required. For getting an idea of the sensitivity of the calculation to small changes in surface temperatures, a number of computations have been done with several resonable variations. It has been found that the δq_r value could be as large as 2 W/m^2. The temperature difference between the enclosure surface and the air point 10 cm from the surface was measured by a series-connection of ten thermo-couples. The accuracy of the measurement is therefore very precise. Despite

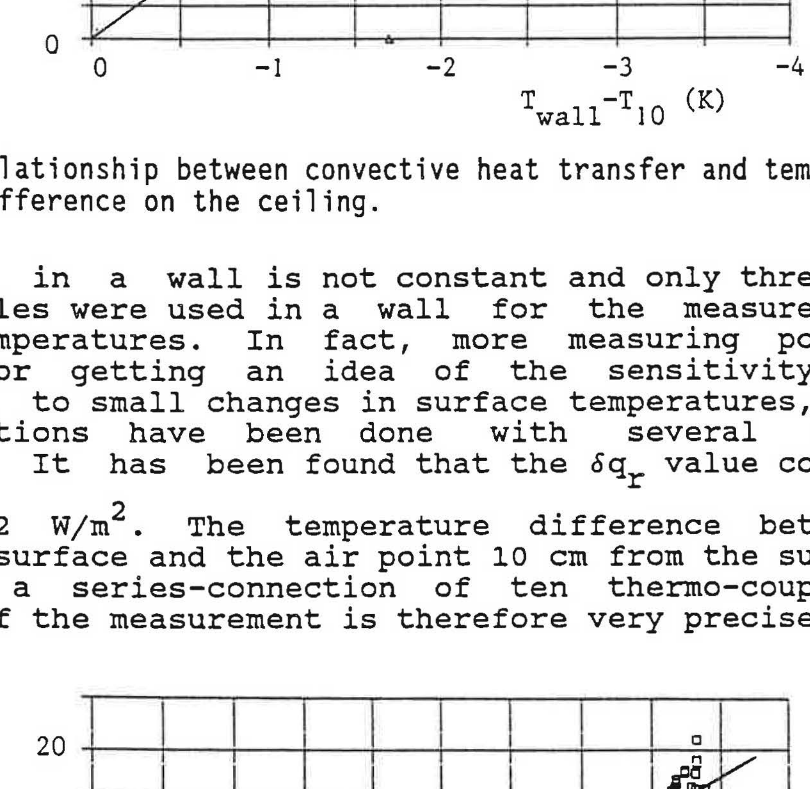


Fig. 8. Relationship between convective heat transfer and temperature difference on the floor.

this, due to there is a slight difference in the locations of the thermo-couples and the heat flux meters as shown in Fig. 2, a 0.3 K of the measuring error of the temperature differences is possible. With Equation (6), we obtain that the measuring errors of the convective heat transfer coefficient may be as large as 1.5 to 2.5 W/m^2K.

With such large possible errors, it is of course very difficult to find the relationships among those variables. For further measurements, we may recommend (1) to concentrate the measurements on only small particular areas; (2) to increase the number of measuring points for the temperatures on enclosure surfaces; (3) to place a heat flux meter and a thermo-couple series as close as possible; (4) to avoid complex geometry (for example, the venetian blinds are a disturbing factor in the determination of the radiative heat exchange).

Results on the Venetian Blinds

The heat transfer in a window with venetian blinds is rather complicated. It is very important in building energy calculations and the computations of indoor airflow patterns. Therefore, it will be discussed in this section.

The experiment on the heat transfer in a window with venetian blinds was carried out in the same time and under the same conditions as those described in the previous sections. Fig. 9 gives some additional information of measuring point locations.

The heat exchange in the window is illustrated in Fig. 10. The energy balance equation of the window can be expressed as:

$$Q_V = Q_{VC} + Q_{VF} + Q_{VW} + Q_{VE} \tag{7}$$

Q_V is the electricity supplied to the venetian blinds. Q_{VC} is the convective heat transfer between the blinds and the cavity air (between the venetian blinds and the glass). Q_{VC} can be written as:

$$Q_{VC} = h_{VC} (T_V - T_C) A_V \tag{8}$$

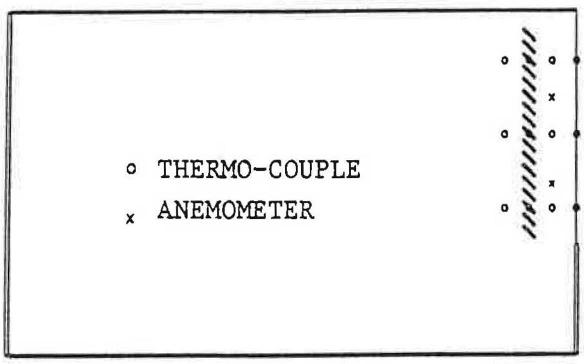


Fig. 9. Measuring points near the venetian blinds of the window.

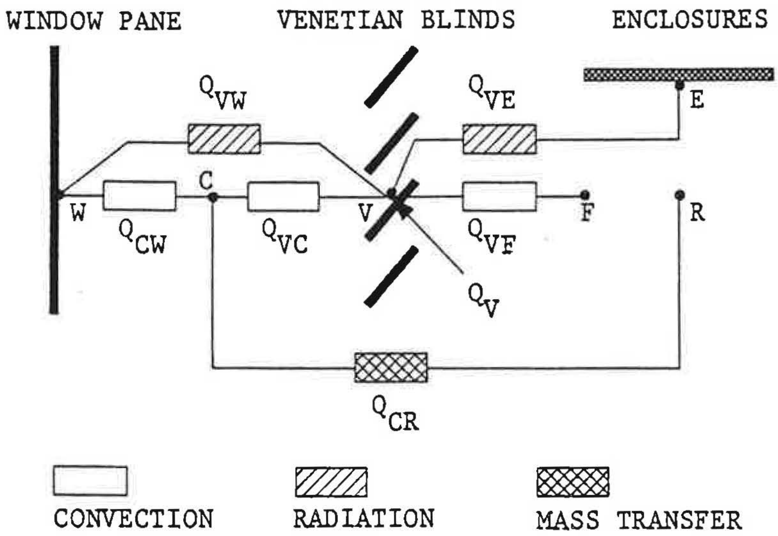


Fig. 10. Heat exchange in the venetian blinds.

where h_{VC} is the convective heat transfer coefficient for the surface, T_V and T_C are the temperatures of the blinds and the cavity air, and A_V is the area of the venetian blinds and is assumed to be equal to the window surface area. The real surface area of the slats of one side is approximately 20% higher. Q_{VF} is the convective heat transfer between the blinds and the room air where F is at the point 10 cm from the blinds:

$$Q_{VF} = h_{VF} (T_V - T_F) A_V \tag{9}$$

Q_{VW} is the radiative heat exchange between the blinds and the glass pane and Q_{VE} is the radiative heat exchange between the blinds and the enclosure surfaces of the room.

In order to make Equation (7) soluble, the energy exchange between the cavity air and the room air by mass transfer should be estimated. It is calculated from the following equation:

$$Q_{CR} = \rho \ c_p \ (T_C - T_R) \ v_C \ A_C \tag{10}$$

where ρ is air density, c_p is the specific heat of air, v_C is the average air velocity in the cavity and A_C is the section area of the cavity (the width of the window by the distance between the blinds and the glass pane). The v_C used in the equation is the mean value of v_{C1} and v_{C2} which are measured by the two anemometers in the cavity.

We have realized that the described method has a few flaws. A weak point, for example, is the assumption for obtaining Q_{CR} by Equation (10). Under certain circumstances, we might expect the presence of eddies in the cavity.

The measured results are presented in Table 3 and the corresponding convective heat transfer coefficients are indicated in Table 4.

The convective heat exchange coefficients in Table 4 show a reasonable consistency. Averaging of the results leads to $\bar{h}_{VC}$ = 4.5 [W/m^{2}K], $\bar{h}_{VF}$ = 10.7 [W/m^{2}K] and $\bar{h}_{VR}$ = 7.9 [W/m^{2}K]. Since the results discussed only present a rough indication on the heat transfer in a window with venetian blinds, more measurements under mixed convection are suggested.

An Application Example of the Measured Results

The measured convective heat transfer coefficients have been applied for the numerical simulation of three-dimensional indoor airflow and temperature distributions. The airflow program PHOENICS [9], which is based on the k-ϵ turbulence model, is used for the calculations of system 1 with 600 W of heat on the venetian blinds and an air supply of five-time air exchange rate per hour.

The computed velocity and temperature distributions of the

Table 3. Measured results from the venetian blinds*

Sys.	Air supply ach	Q_V W	T_W °C	T_C °C	T_V °C	T_F °C	T_R °C	v_{C1} m/s	v_{C2} m/s
1	7	950	25.0	27.0	31.1	25.4	24.0	0.07	0.38
1	7	600	24.1	25.0	28.3	24.2	23.3	0.06	0.34
1	7	300	23.6	24.3	25.8	24.1	23.6	0.08	0.27
1	5	600	24.5	25.7	29.6	25.3	23.5	0.04	0.25
1	3	600	24.6	25.8	30.0	25.8	23.5	0.04	0.08
2	7	950	25.3	26.5	32.2	26.7	24.7	0.13	0.45
2	7	500	24.0	25.1	28.1	24.9	23.7	0.11	0.29
2	5	600	24.6	25.1	29.2	25.2	23.1	0.04	0.21
2	3	300	23.7	24.4	26.3	23.5	23.0	0.04	0.09

* The values are the average ones.

Table 4. Convective heat transfer coefficient on the venetian blinds

Sys.	Air supply ach	Q_V W	Q_{VW} W	Q_{VE} W	Q_{CR} W	Q_{CW} W	Q_{VC} W	h_{VC} W/m2K	h_{VF} W/m2K	h_{VR}* W/m2K
1	7	950	145	105	200	56	256	8.9	11.1	8.9
1	7	600	100	65	101	25	126	5.5	10.8	8.8
1	7	300	60	40	36	20	56	5.3	12.1	9.4
1	5	600	115	90	95	34	129	4.7	8.8	6.2
1	3	600	125	85	41	34	75	2.6	10.7	6.9
2	7	950	155	125	155	34	189	4.7	12.5	9.2
2	7	500	90	70	83	31	114	5.4	10.1	7.3
2	5	600	110	75	74	14	88	3.1	11.7	7.7
2	3	300	60	35	27	20	47	3.5	8.1	6.8

* h_{VR} is defined as Q_{VF} / $(T_V - T_R)$ A_V.

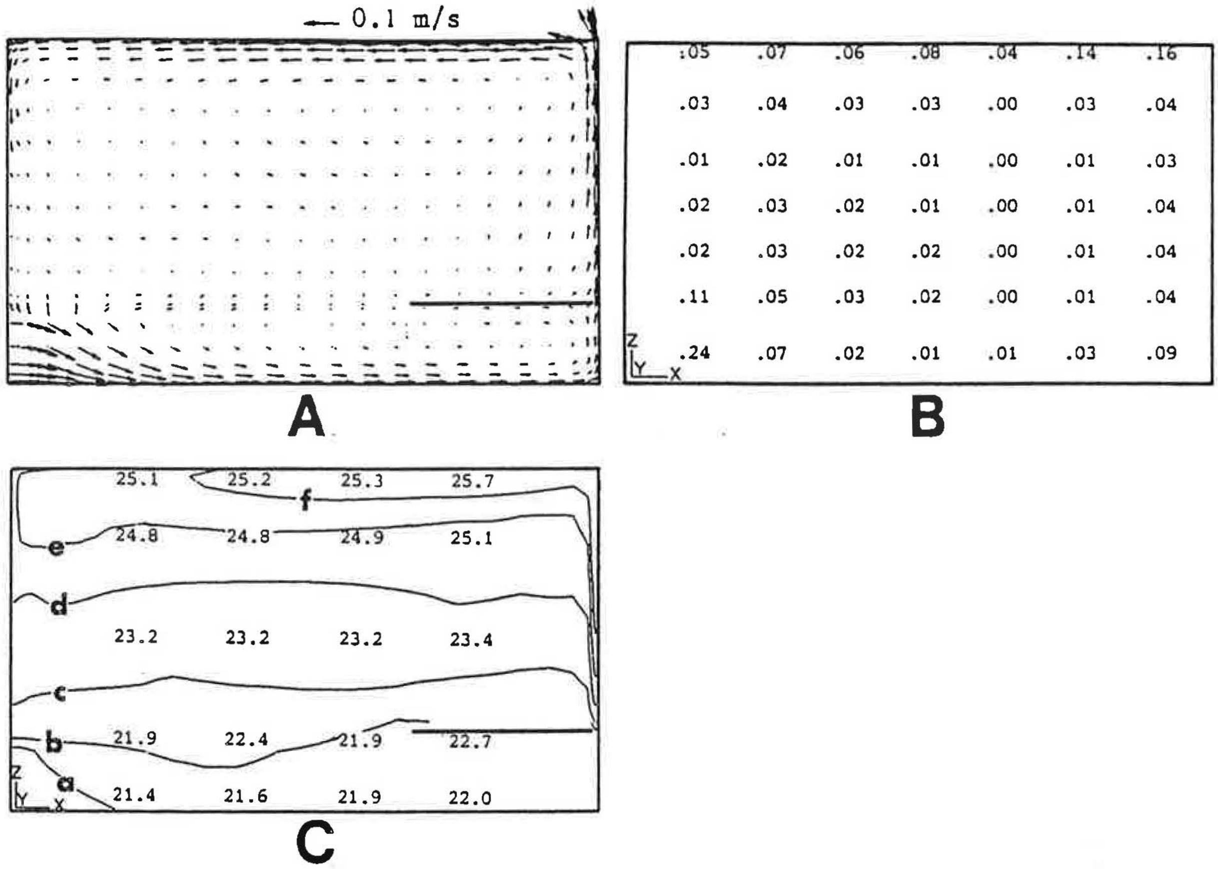


Fig. 11. The computed and measured results for system 1 in section y=1.5 m of the room. (A) Computed velocities (m/s); (B) measured velocities (m/s); (C) computed (contours) and measured (numerical values) temperature field (°C) (a-21 b-22 c-23 d-24 e-25 f-26).

room air in section y=1.5 m (middle section) are given in Fig. 11. The agreement between the computations and the measurements are acceptable for engineering utilizations although there are some discrepancies. The difference between the computations and the measurements in most cases occurs where the variation in gradient is large. This means that a slight difference in the measuring location can cause a significant error. More detailed information can be found in literature [2, 10].

Conclusions

Experiments have been carried out for obtaining the convective heat transfer coefficients in the enclosure surfaces of a room with mixed convection. It has been found that the convective heat transfer coefficient is about 4.0 W/m^2K for the ceiling and 4.7 for the floor with unstable flow and 4.3 for the floor with stable flow. Due to the measuring technical problems, the measuring errors are too large.

We may also conclude that the convective heat transfer coefficient is about 10.7 W/m^2K for the venetian blind surface facing to the room and about 4.5 for the venetian blind surface facing to the glass pane of the window. These results only present a rough indication on the heat transfer in a window with venetian blinds.

Those measured convective heat transfer coefficients have been applied in the numerical simulation of indoor airflow. This has achieved acceptable agreement between the computed and measured velocity and temperature distributions of room air.

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